Forsthoffer's

Best Practice Handbook for Rotating Machinery

William E. Forsthoffer

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Dedication

To my wife, children and extended family for their total faith in my endeavors

Contents

Preface

The objective of this book is to enable the implementation of cost effective best practices that will optimize plant safety, reliability and profits. The information is all presented in one book with easy to find best practices that have been demonstrated globally and have benchmarked success.

The format for each of the 219 best practices in this book is to present:

- The Best Practice in clear and concise terms
- The Lesson Learned that resulted in the Best Practice
- Benchmarks or 'references' to prove that the best practice results in increased safety, profit and reliability
- Supporting information to enable you to ensure timely management implementation of your recommendation.

This book is arranged to allow easy access to information, by beginning with project lessons, progressing to individual chapters on rotating equipment lessons (covering each equipment type in order from performance through to mechanical components) and concluding with construction, pre-commissioning, start-up and predictive maintenance details. I have also included a chapter on Communication Best Practices.

As is the case for my previous Machinery Handbook Series, this text is intended for everyone associated with rotating equipment, regardless of educational background (operators, millwrights and engineers).

In summary, it is my intention to present rotating machinery global best practices attained through my 45+ years in the industry, to anyone in any company that has the objective of achieving optimum rotating equipment safety and reliability. It is my belief that the information presented in this book can significantly increase the implementation rate of recommendations made to plant and corporate management.

Acknowledgements

The best practices contained in this book and the lessons learned that produced them are the result of interactions with many experienced and dedicated operators, millwrights, engineers and mentors from end users, contractors and suppliers. I would not be able to write this book if it were not for their unselfishness in sharing their knowledge.

I consider myself to be very fortunate to have been associated with the 'best of the best' mentors, business associates and dear friends throughout my career; most especially, in chronological order, Dick Salzmann, Bob Aimone, Merle Crane, Walt Neibel, the late Murray Rost, Mike Sweeney and Jimmy Trice. All have contributed to the supporting material in this book.

Special thanks go to all of the global machinery vendors who have allowed me to use their material in the 'Supporting Material' sections of this book.

Last but certainly not least; my career would not have been possible without the support, encouragement and assistance from my wife Doris and our children Jennifer, Brian, Eric, Michael and Dara. A special note of thanks to Michael who helped with the metric units for this publication and who has decided to pursue a career in rotating machinery.

About the Author

Over the last 48 years, in the position of a designer, end user and consultant, Bill has had the opportunity to be involved with all types of rotating equipment: pumps, compressors, gears, mixers, extruders, melt pumps, steam turbines, gas turbines and their associated auxiliary systems. Involvement consisted of: total component and system centrifugal compressor design, specification writing for a major oil company, selection of all types of rotating equipment from all major vendors, design audits, shop testing, start-up and troubleshooting for all major gas processing, chemical and refining companies world-wide. This background has resulted in a lot of 'Lessons Learned' and many of them coming the 'hard way'!

How to use this Book

The purpose of this book is to increase your rotating machinery recommendation implementation rate. The effective use of this book will enable you to meet this objective by presenting 219 best practices that will increase machinery safety, reliability and revenue.

To use this book in the most effective manner; go to the selected topic chapter and consult either the best practice or the lesson learned index. Each best practice and its corresponding lesson learned are arranged in numerical order.

The first page of each best practice is arranged in the following order:

- Best Practice – positively impacted safety and increased reliability and revenue
- Lesson Learned – negatively affected safety and reduced reliability and revenue
- Benchmark – where the best practice was used and its impact on safety, reliability and revenue
- Supporting Material – material that will help support the best practice recommendation to ensure rapid implementation by plant or corporate management.

Refer to the management implementation best practice in Chapter 12 for guidelines for the most effective method of presenting best practice recommendations to plant management.

List of Best Practices

Chapter 4 Gear and Coupling Best Practices

Chapter 5 Steam Turbine Best Practices

Chapter 6 Gas Turbine Best Practices

Chapter 7 Lube, Seal and Control Oil System Best Practices

Chapter 8 Pump Mechanical Seal Flush Best Practices

Chapter 9 Dry Gas Seal Best Practices

Chapter 10 The Post Shipment Phase: Installation, Pre-Commissioning, Commissioning and Start-up Best Practices

Chapter 11 Preventive and Predictive Maintenance Best Practices

Chapter 12 Implementation and Communication Best Practices

List of Lessons Learned

Chapter 1 Project Lessons Learned

1.1: Failure to incorporate 'Lessons Learned' during the pre-FEED phase of the project.
1.2: Failure to establish project team support early in the project.
1.3: Not screening vendors for experience for the project.
1.4: Failure to consider the proper selection and design of each item in the equipment train.
1.5: Excessive specs and incomplete data sheets reduces safety, reliability and plant revenue.
1.6: Absence or lack of ITB vendor instructions results in endless meetings and emails.
1.7: 80% of machinery failures are due to process changes not considered in the design phase.
1.8: Failure to have pre-bid meetings will reduce critical machinery safety and reliability.
1.9: Using the EP&C typical bid tabulation format extends the bid cycle time.
1.10: Poorly planned pre-award meetings will affect project schedule and result in cost adders.
1.11: Poorly planned VCMs will result in cost adders and delivery schedule extensions.
1.12: Failure to conduct timely design audits affects machinery safety and/or reliability.
1.13: Inefficient vendor and EP&C document reviews will cause delivery delays.
1.14: Failure to conduct an effective FAT will expose the end user to future reliability issues.

Chapter 2 Pump Lessons Learned

2.1: Not using lessons learned to select the pump will lead to lower pump reliability.
2.2: Letting the EP&C select pumps can lead to lower pump reliability.
2.3: Changes in operating conditions will cause lower pump reliability.
2.4: Leaving pump case design to the EP& C can lead to field safety and reliability issues.
2.5: Double suction pumps have the greatest amount of field reliability problems.
2.6: Operating an inducer pump outside its BEP by more than +/- 10% lowers reliability.
2.7: Failure to establish EROE limits will lead to low MTBF of centrifugal pumps.
2.8: Critical centrifugal pumps and 'bad actor' pumps require operator surveillance.
2.9: Centrifugal pump flow is determined by the head required by the process.
2.10: Failure to install discharge orifices in high speed pumps reduces MTBF.
2.11: Pump NPSH margins are traditionally set for only the rated operating point.
2.12: Selecting a pump for NPSH required without considering suction specific speed.
2.13: Pump bearings with a DN number above 350,000 are prone to low bearing MTBF.
2.14: Optimum MTBFs for high DN pumps is not possible without continuous lubrication.
2.15: Many pump failures have been caused by poorly designed temporary suction strainers.
2.16: EP&Cs and end users do not have guidelines for the use of min. flow bypass valves.
2.17: Low centrifugal pump flow results in pump damage and exposure to safety issues.
2.18: Internal type bypass valves can fail to open at the minimum flow set point.
2.19: Centrifugal pumps operating in parallel are prone to damage from liquid vaporization.
2.20: Excessive pump flange and foundation forces will lead to low bearing MTBF.
2.21: Many pumps have impellers that are too large for field operation parameters.
2.22: Lack of a plant spare changeover philosophy reduces pump MTBFs.
2.23: Impeller diameter increases have resulted in driver and/or coupling overloads.
2.24: Controlling centrifugal pumps with control valves is not energy efficient.
2.25: Operators need to understand the use of centrifugal pump test curves.

Chapter 3 Compressor Lessons Learned

3.1: Improper compressor selection will reduce safety and reliability.
3.2: Inaccurate process conditions cause project scope changes and increase delivery time.
3.3: Failure to use the proper case design and number of impellers reduces reliability.

3.4: Failure to limit piston speed in non-lube reciprocating applications results in low MTBFs.
3.5: Failure to address pulsation problems has resulted in safety and reliability issues.
3.6: Using high speed reciprocating applications has resulted in $MM in lost revenue and maintenance costs.
3.7: Do not use lubricated screw compressor types in sour gas services.
3.8: Horizontal split centrifugal compressors with top nozzles extend maintenance times.
3.9: Failure to review for impeller/blade experience reduces reliability and revenue.
3.10: Failure to limit head per impeller reduces safety, reliability and revenue.
3.11: Avoid the flat centrifugal (low head rise) compressor characteristic curves.
3.12: Avoid operating individual impellers more than 20% greater than their rated flow.
3.13: Require new curves if gas density changes more than +/- 20% from specified values.
3.14: When performance and phase angle are not monitored, fouling cannot be detected.
3.15: The use of low solidity diffusers (LSD) has resulted in severe production reductions.
3.16: Integral geared compressors have lower reliability than between bearing compressors.
3.17: Integral geared failures can result from intercooler performance deterioration.
3.18: Sub-sync vibration and critical speed issues occur when the shaft stiffness ratio > 10.
3.19: The use of lemon bore or offset sleeve type bearings has caused vibration issues.
3.20: Increase displacement alarm/ trip settings if the bearing pad temperature <108°C.
3.21: Check balance device conditions before changing thrust bearing assemblies.
3.22: Require bushing seals with 'flow through' feature in high pressure applications.
3.23: Dry gas seal reliability issues result from end user lack of proactiveness.
3.24: Low solidity diffusers (LSDs) can reduce product revenue.
3.25: Vacuum shop tests do not completely confirm centrifugal compressor acceptability.
3.26: Using the aftercooler for recycle gas heat removal reduces surge system response.
3.27: 80% of component failure root causes are contained in process changes.
3.28: Failure to consider sufficient drive system capability reduces revenue and reliability.

Chapter 4 Gear and Couplings Lessons Learned

4.1: Planetary gears require many additional gears and bearings which lowers reliability.
4.2: Damaging gearbox flooding can occur at pitch line velocities > 6,000 meters/min.
4.3: Avoid the use of three bearing systems in high torque reduction gears.
4.4: Gears can experience high vibration during start-up.
4.5: Integral gear compressor bearings are highly loaded and require centerline shaft position monitoring.
4.6: Oil lubricated gear couplings can require yearly PM checks if PDM is not practiced.
4.7: The majority of hydraulic coupling slip incidents are caused by liquid accumulation.
4.8: High vibration has occurred due to shaft end thermal axial growth calculation errors.
4.9: $^3/_4$" per foot tapers have caused catastrophic coupling failures.
4.10: Installing a hydraulic coupling with 'O' rings installed can cause coupling slippage.

Chapter 5 Steam Turbine Lessons Learned

5.1: Failure to properly specify steam conditions has led to low power output and erosion.
5.2: Fouling and steam conditions which are lower than anticipated can result in lost revenue.
5.3: The use of prototype blading has resulted in costly blade failures.
5.4: Failure to trend 'after first stage' pressure will result reduced power and vibration.
5.5: Failure to screen for shaft stiffness has resulted reduced field reliability.
5.6: Lemon bore or offset sleeve bearings have caused extended FAT test periods.
5.7: Many impulse steam turbines have encountered high thrust bearing pad temperatures.
5.8: Failure to monitor gland condenser vacuum has caused contamination of oil systems.
5.9: Most special purpose turbine gland sealing systems fail and cannot be repaired on-line.
5.10: Specific machinery unit field testing is not cost effective and should not be accepted.
5.11: Uncoupled overspeed trip checks expose personnel to injury.
5.12: Failure to exercise trip valves has resulted in catastrophic machinery failure.
5.13: Efficiency calculations will be erroneous when torquemeters are not installed.
5.14: Single stage turbine steam seal systems are ineffective and contaminate oil systems.
5.15: Single stage, turbine, lube oil pump drive governor linkage has caused unit tips.
5.16: Keeping hand valves closed in critical services has resulted in process unit ESDs.
5.17: Failure to exercise single valve steam turbines has led to critical train trips.

Chapter 6 Gas Turbine Lessons Learned

6.1: Aero-derivative types now have reliabilities equal to or greater than industrial types.

6.2: Failure to consider actual site conditions has resulted in power deficient gas turbines.
6.3: The majority of mechanical drive gas turbines produce insufficient site power.
6.4: Failure to use experienced gas turbine systems has led to significant start-up delays.
6.5: Lemon bore or offset sleeve bearings have extended FAT periods.
6.6: Modular change-out agreements can significantly reduce downtime for repair.
6.7: An inadequately sized inlet air filter will reduce turbine power.
6.8: Carbon steel lube systems extend oil flush times and expose bearings to failure.
6.9: Failure to trend performance does not allow maintenance cycles to be extended.
6.10: Efficiency calculations will be erroneous when torquemeters are not installed.

Chapter 7 Lube, Seal and Control Oil System Lessons Learned

7.1: Conditions and oil grades other than those specified will affect system reliability.
7.2: Critical trains with carbon steel systems have suffered un-scheduled shutdowns.
7.3: Consoles that do not allow easy personnel access reduce system reliability.
7.4: Not filling console baseplates with cement will affect component reliability.
7.5: Failure to design audit oil systems has caused many start-up delays and trips.
7.6: Failure to prevent infiltration of oil into refrigeration systems will result in large decreases in plant capacity.
7.7: Not using centrifugal pumps in oil systems reduces system reliability.
7.8: Failure to detect rotary pump strainer blockage has resulted in excessive pump wear.
7.9: Steam turbine main pump drivers have the lowest reliability of any oil pump driver.
7.10: Steam turbine main oil pump drivers are responsible for most oil system trips.
7.11: Many unit trips have been traced back to improper setting of relief valves.
7.12: Critical (un-spared) unit trips have been caused by friction-bound relief valves.
7.13: Pneumatic actuated valves response time is less than direct acting control valves.
7.14: Lube oil systems installed without accumulators will eventually cause unit trips.
7.15: Filters do experience bypassing and expose critical machines to bearing failures.
7.16: Failure to check all oil system functions during the FAT will result in delayed start-up.
7.17: Oil systems not provided with TMR shutdown logic experience lower reliability.
7.18: Poor oil system field flushes result in significantly extended start-up times.
7.19: Failure to condition monitor oil reservoirs and rundown tanks causes failures.
7.20: Cool oil in the non-operating cooler or filter can cause unit trips on low oil pressure.
7.21: Excessive unit trips lower the reliability of critical machines.
7.22: Plants frequently experience unit trips caused by control valve issues.
7.23: There have been explosions and fires from excessive gas in the oil reservoir.
7.24: Many plants have low machine reliability due to old instrument malfunction.
7.25: Failure to mark and monitor control valve stem position has led to many surprises.
7.26: Oil system functional checks during shutdown do not confirm transient functions.
7.27: Always evaluate oil system reliability issues in terms of lost revenue costs.
7.28: Improper sizing of interconnecting oil piping has caused reliability and safety issues.
7.29: Not tracing oil system piping in the field has led to many oil system failures.
7.30: Failure to detect initial condition change has caused long shutdown periods.
7.31: Failure to check the accumulator condition has resulted in many unit trips.
7.32: Malfunctioning solenoid valve trip systems have resulted in turbine failures.
7.33: Failure to periodically exercise turbine trip valves has caused catastrophic damage.
7.34: Insufficient oil seal atmospheric bushing cooling has resulted in low seal MTBFs.
7.35: The MTBF of dynamic bushing inner oil seals is less than straight bushing seals.
7.36: Failure to use a sweet and clean buffer gas has resulted in low seal MTBFs.
7.37: Failure to install sight glasses in the atmospheric drain lines has led to gas releases.
7.38: Introducing gas into a bushing seal prior to seal oil operation will cause a gas release.
7.39: An atmospheric bushing operating on low reference gas pressure will overheat.
7.40: Many oil seal failures are caused by debris from dirty overhead seal oil tanks.
7.41: Internal float type drainers have been responsible for plant safety issues.
7.42: Seal oil entry into the compressor can be the result of too small drainer orifices.
7.43: Degassing tanks do not eliminate the entrance of process gas into the oil reservoir.

Chapter 8 Pump Mechanical Seal Flush Lessons Learned

8.1: Incorrect process conditions on the seal data sheet will lower seal MTBFs.
8.2: The use of strainers or cyclones in the flush system will lower seal MTBFs.

8.3: Using cartridge seal assemblies has resulted in significantly higher seal MTBFs.
8.4: All cartridge seals should be statically pressure tested prior to installation.
8.5: Use of un-pressurized dual seals has exposed plant and personnel to potential losses.
8.6: Failure to monitor seal chamber pressure has caused low mechanical seal MTBFs.
8.7: Checks all seal system components every time the seal is replaced to optimize MTBF.
8.8: Low seal MTBFs have been caused by incompatible external flushes.
8.9: Use external flushes instead of flush line strainers and cyclone separators.
8.10: Failure to monitor the operation of seal quench systems has caused low seal MTBFs.
8.11: Failure to monitor performance is the major contributor to pump component failure.
8.12: Process fluid will leak from dual un-pressurized seal systems if the vent line is closed.
8.13: Lack of a mechanical seal installation procedure has caused low seal MTBFs.

Chapter 9 Dry Gas Seal Lessons Learned

9.1: Failure to consider plant operating conditions has resulted in low dry gas seal MTBFs.
9.2: Double dry gas seal systems have fewer components than tandem seal systems.
9.3: Failure to monitor secondary seals has exposed plant and personnel to safety issues.
9.4: Labyrinth separation seal nitrogen consumption is too great to justify their usage.
9.5: The use of 'bone dry' N_2 (dew points below −30°C) has resulted in low seal MTBFs.
9.6: Seal gas conditioning does not ensure reliability in saturated process gas applications.
9.7: Many failures occur where the flare header can exceed the seal gas supply pressure.
9.8: Dry gas seal separation systems do not prevent carryover of oil into the seal chamber.
9.9: Failure to include a seal gas conditioning unit has resulted in low seal reliability.
9.10: Tandem seals with a seal chamber pressure above 0.5 barg have higher MTBFs.
9.11: A primary orifice bypass device in high suction pressure services increases reliability.
9.12: Tripping on high primary vent conditions only can cause excessive plant shutdowns.
9.13: Never connect bearing housing vents if dry gas seals are used in the compressor.
9.14: Not using a delta pressure controller for the separation seal has caused gas releases.
9.15: Lack of personnel awareness of DGS system function reduces seal MTBFs.

Chapter 10 The Post Shipment Phase Installation, Pre-Commissioning, Commissioning and Start-up Lessons Learned

10.1: Plant life safety and reliability is directly connected to the project construction phase.
10.2: Delayed machinery start-ups are often caused by insufficient preservation programs.
10.3: Vendor site visits are not effective unless detailed site details are sent in advance.
10.4: Installing machinery without a proven foundation procedure will reduce MTBF.
10.5: Failure to use epoxy grout for all machinery installations will reduce MTBFs.
10.6: Improper process piping attachment will significantly reduce machine MTBFs.
10.7: Most oil flush procedures take too long and are not effective.
10.8: Failure to check operation of auxiliary systems prior to start results in start-up delays.
10.9: Failure to perform an inert gas run exposes the plant to reduced safety and reliability.
10.10: Not performing commissioning driver solo runs exposes the plant to revenue losses.
10.11: Failure to obtain detailed component baseline conditions reduces unit reliability.
10.12: Lack of operator machinery awareness will reduce machinery safety and reliability.
10.13: Lack of an effective site predictive maintenance (PDM) program reduces MTBF.

Chapter 11 Preventive and Predictive Maintenance Lessons Learned

11.1: PM (time based) programs do not increase machinery reliability and are costly.
11.2: Short pump unit bearing bracket oil change intervals (13−26) weeks reduce MTBFs.
11.3: Lack of a pump changeover program results in significantly lower MTBFs.
11.4: Control valve replacement during plant operation exposes the plant to shutdowns.
11.5: Yearly oil lubricated coupling PMs expose the plant to unnecessary shutdowns.
11.6: Oil system transient tests performed during a turnaround are not accurate.
11.7: Failure to trend machinery operating points reduces machinery safety and reliability.
11.8: PM-heavy maintenance programs produce lower machinery MTBFs.
11.9: Reliability groups not using operations and process input produce lower MTBFs.
11.10: Operations and process groups need increased machinery functional awareness.

11.11: Plants not sending personnel to machinery seminars suffer lower MTBFs.
11.12: Plants that do not have a strong vendor bond are subject to varying vendor support.

Chapter 12 Implementation and Communication Lessons Learned

12.1: A low implementation rate is usually the result of poor management presentation techniques.
12.2: Failure to continuously improve best practices will result in reduced reliability.
12.3: Failure to convert plant lessons learned into best practices results in lower MTBFs.
12.4: Plants that do not offer site specific machinery workshops experience lower MTBFs.
12.5: Incomplete and inaccurate emails can be the root causes of reliability issues.
12.6: Including email recipients not directly concerned with the issue delays action.
12.7: Meetings conducted without a detailed agenda sent in advance are ineffective.
12.8: Failure to follow meeting best practice guidelines has resulted in ineffective meetings.
12.9: Failure to issue detailed technical reports in a timely fashion affects implementation.

1 Project Best Practices

Introduction

Plant machinery safety and reliability begins here – during the project phase. Input from lessons learned and incorporation of the resulting best practices into the earliest phase of the project will ensure a plant that enjoys optimum safety and reliability for its usable life. Use the best practices contained in this chapter to convince and support management to make your project and the resulting plant the "best of the best".

Best Practice 1.1

Input machinery lessons learned to the project team during the project pre-FEED phase.

Obtain plant, company and industry (from seminar attendance and publications) lessons learned and incorporate them into the project scope during the pre-feed phase, to ensure that the 'cost of incorporation' will be included in the project scope.

Define the associated best practice for each lesson learned and note these B.P.s on the appropriate machinery data sheet on a special page, to ensure they are included in all quoting suppliers' scope and costs. This action will ensure that all supplier content will be equal.

Do not accept exceptions to any of the required best practices. If a certain supplier refuses to incorporate any or all of the best practices, remove them from the bidder's list for this particular project, and explain why they were removed.

Confirm that all required best practices are included in the supplier's final bids and the purchase order to eliminate any project schedule delays and cost adders.

Lessons Learned

Failure to incorporate 'lessons learned' during the pre-FEED phase of the project will result in lower plant safety, reliability, and revenue and/or extended project schedule and can result in supplier cost adders.

Not having specific lessons learned defined as best practices in the invitation to bid has resulted in the following issues during the project:

- Significant differences in supplier scope
- Significant difference in supplier costs resulting in frequent low cost bid (and most usually lower scope) acceptance
- Possible required supplier scope changes during the project resulting in schedule delays and/or high cost adders

Note that industry specifications (API) and company specifications are not written for specific plant conditions and locations, and will not include all the necessary best practices.

Benchmarks

This best practice has been used since 1990, and has been incorporated into the following projects with the benefits noted above:

- Mega ethylene plant
- Mega butyl rubber plant
- Methanol plants
- Refinery hydrocracker recycle compressor
- Oil and gas booster compressor trains
- Small (modular) LNG plants

This practice has resulted in minimum bid decision time for large compression trains (10 weeks from issue of ITB). Life cycle cost savings exceed $5,000,000 per year for the present process unit size.

B.P. 1.1. Supporting Material

As someone who has been involved with projects as a rotating equipment vendor, end user and consultant since 1970, I have had the opportunity to see custom-designed rotating equipment projects from all industry viewpoints. Regardless of your position, you will face the challenges of company profit optimization, depleted workforce experience levels and time constraints.

My initial involvement with rotating equipment projects began in 1970 as a project engineer for a centrifugal compressor vendor, where I was responsible for the project management of all process compressor applications. This interesting and busy portion of my career taught me many valuable lessons, and the challenges and associated action required to survive this experience. 'Vendor lessons learned' are detailed in Figure 1.1.1.

It was interesting to note that in my next industry position, as a corporate rotating equipment specialist for a major oil, gas and chemical company, I observed that the characteristics noted above were present in all equipment companies regardless of global location or final product. However, in my new position there were also many challenges as noted in Figure 1.1.2.

Review Figures 1.1.1 and 1.1.2 and observe the similarities; all imposed by time and budget constraints. Also, observe how

- Time constraints forced the acceptance of what was on the data sheet
- The tendency was to think inside the flanges of the compressor only and not consider the process
- Questions to the end user/contractor were minimal based upon competitive pressures and time constraints
- Copying from past jobs 'cut and paste' was a necessity to minimize engineering hours and today (21st century) is electronic cut and paste
- Contractor/end user questions diminished valuable engineering time. There was little time or money for visits to client plants unless there were significant design problems

Fig 1.1.1 • Vendor lessons learned

- Time constraints forced acceptance of what was on the process data sheet without time to question the basis for the stated conditions
- The tendency initially was to think inside the machinery flanges, but eventually it was understood that all equipment is directly influenced by the process
- Contact with the client (plant where the equipment will be installed) was minimal based on project team pressures for schedule milestones
- Company specification contents were increasing rapidly since all company divisions and plants were required to review specifications and therefore naturally contribute something
- There was limited project budget for visits to client plants unless there were equipment design problems.

Fig 1.1.2 • End user lessons learned

the individuals involved seldom have the opportunity to observe how their client operates and what their objectives are.

Since 1990, as a rotating equipment consultant engaged in troubleshooting, machinery selection and revamps and site specific operator, maintenance and engineering training, I have faced other challenges, but the similarities are striking and the challenges are the same. These facts are noted in Figure 1.1.3.

- Both vendors and clients have limited experience bases
- Decisions are made quickly, often without benefit of all the pertinent facts
- Most projects are run on the basis of minimum capital investment and not life cycle cost
- Implementation of action plans is slow
- Vendor and end users interface infrequently – usually only during field failures

Fig 1.1.3 • Contractor/consultant lessons learned

Based on my experience, I have learned, most of the time the hard way, that all three of these groups (vendors, contractors and end users) have the same objective but different means of obtaining that objective. Figure 1.1.4 presents these facts.

Everyone has the objective of maximum profits but the means to accomplish this end is different:

- Vendor – designs for minimum cost
- Contractor – engineers and installs for minimum cost
- End user – must operate the custom designed equipment 24/7 for 30 years. Therefore, the end user's objectives can be directly opposed to the vendor's and contractor's!

Fig 1.1.4 • The objective – maximum profits

It is important to remember these facts at all times during the entire project. The information contained in the following figure should be the basis for convincing the project team that all decisions regarding equipment purchase should be made on the basis of process unit life cycle cost and not capital cost and/or schedule considerations. The specific objectives of the end user are presented in Figure 1.1.5.

- Maximum machine reliability
- Minimum operating cost
- Minimum time to repair

These objectives result in............................maximum up time
which will yield... maximum revenue
and... maximum profits
For the entire life cycle of the process unit!

Fig 1.1.5 • End user specific objectives for maximum profit

The most important factor in life cycle cost considerations is daily revenue, and obtaining this figure should be the number one priority in the early stages of the project. It will be a key fact in obtaining management support for your project action plans. Figure 1.1.6 presents these facts.

- Is the amount of revenue obtained in 24 hours of operation
- Trip of an un-spared item = exposure to revenue loss
- Daily revenue values can range from 1MM$ to 5MM$+
- Always justify project scope requirements on the basis of daily revenue loss
- Assign an actual daily revenue loss amount to each proposed best practice if it is not implemented

Fig 1.1.6 • Daily revenue

Therefore, the company life cycle revenue and profit potential will be a result of incorporating all of your project best practice requirements into the project action plan at the first opportunity, before the first project budget estimate is prepared. Figure 1.1.7 shows the advantages of incorporating this philosophy as early as possible into the project.

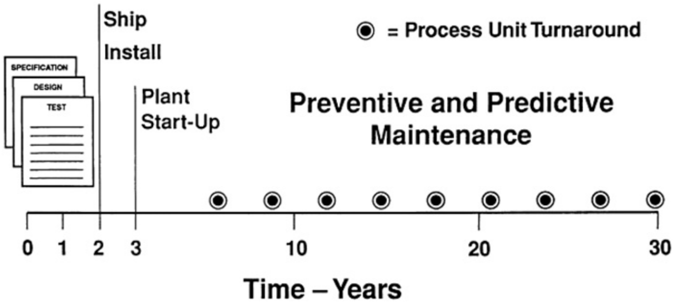


Fig 1.1.7 • The life span of rotating equipment

This action should be taken when the project is first announced, and the project team is assembled. The approach taken during the first 3–6 months after the initial project kick-off will determine the level of reliability and life cycle cost savings for the entire life of the process unit (over 30 years). Most important is the necessity of establishing immediate creditability with the project team so that your ideas will be implemented.

Hopefully, the above information will be of use in your project involvement in terms of lessons learned. The resulting best practices should be developed into a project philosophy that will eliminate all the issues noted above, and will obtain and maintain your management's support throughout the entire project; from the pre-feed phase to field operation.

Note that while this book is concerned with rotating equipment best practices, many of the principles are equally applicable to all assets included in a project.

Best Practice 1.2

Establish project management team trust, communication and support in the pre-FEED phase of the project.

Corporate project culture should require immediate project team selection upon project announcement.

Meet with all project team members, and present machinery lessons learned and best practices to project management as soon as the project is announced (pre-FEED phase).

Be sure to include only those best practices that increase safety and show the revenue savings (cost of unavailability) achieved by increased reliability.

Be sure to benchmark all best practices and cooperate fully with the project team, with timely updates and communication.

Lessons Learned

Failure to establish project team support early in the project will result in lack of implementation of proposed action and/or possible schedule impact and cost adders.

Not having project management support from the beginning of the project has resulted in the following issues during the project:

- Lack of implementation of company best practices in the project machinery scope.
- Acceptance of machinery scope by project management without the knowledge of the machinery project specialist.
- Possible required supplier scope changes during the project resulting in schedule delays and/or high cost adders.

Benchmarks

This best practice has been used since the 1970s, and has been incorporated into all major projects that the author has been associated with in the following industries:

- Upstream oil and gas
- LNG
- Refining
- Chemical
- Co-generation

Good machinery specialist/project management relationships have resulted in minimum project time for machinery selection, minimum schedule delays and smooth start-up.

Centrifugal compressor reliabilities have exceeded 99.5% when this approach is followed for the entire project (pre-FEED, FEED, test, construction, commissioning and start-up).

B.P. 1.2. Supporting Material

The action taken during the pre-FEED phase (Front End Engineering Design) relative to rotating equipment will set the stage for its availability and profit improvement for the entire life of the process unit. However, the company must take the initiative to assemble and brief the project team members immediately upon project inception. Corporate responsibilities are outlined in Figure 1.2.1.

The corporate action outlined in Figure 1.2.1 will enable the specialist to acquire the project information that she or he needs to determine the degree of risk involved for the purchase and manufacture of the critical (custom designed) equipment needed in the project. In addition, the specialist can prepare the machinery best practice list from company and plant lessons learned, being sure to only select those best practices that affect safety, and produce significant revenue increases. The manner in which project recommendations are presented will have a great impact on project team trust and support. Figure 1.2.2 presents these facts.

- Assemble the entire project team immediately upon project announcement
- The team should include existing plant maintenance, operations personnel or experienced personnel if the plant is a grass roots (new) installation
- Brief specialists regarding details of process, size of equipment special details etc.
- Require specialist input immediately regarding equipment special project requirements (Best Practices)

Fig 1.2.1 • Corporate project responsibilities

- Present best practices based on lessons learned
- Benchmark each best practice in terms of safety and revenue increase
- Present envisioned equipment overview based on input data, estimation calculations, prior vendor discussions and best practices
- Define risk – safety and cost of unavailability
- Recommend action plan – include: audit, application best practice and special test requirements
- Define cost and additional schedule time based on recommended action plan
- Define company savings and increased profit over life of process unit
- Request decision to proceed with the proposed plan

Fig 1.2.2 • Guidelines for project plan presentation

The format of this presentation can range from a discussion with the project engineering manager and project manager to a formal PowerPoint presentation. Regardless of project team trust and type of presentation, time is of the essence and the presentation must detail in clear and concise terms, the specific requirements, schedule time and life cycle cost savings for the proposed plan. An example of turning acquired pre-FEED information into an action plan is presented in the case history in Figure 1.2.3.

The above action was possible because the project team provided early information to the specialists, allowed prescreening meetings to audit vendor experience and took the specialist's recommendations seriously.

This action took place before the budget estimate. The example outlined in Figure 1.2.3 will become more important in the future, as the size of projects increase, and the exposure to loss of daily process unit profit can easily exceed millions of dollars.

- Input from project team – propylene refrigeration duty data sheet
- Calculations and vendor discussions showed that duty required a prototype machine in regards to rotor bearing span and shaft diameter (shaft stiffness)
- Risk class was determined as multiple component inexperience
- Vendors were invited to pre-screening design review meetings to determine action
- Based on meeting reviews with three vendors, it was determined that bearing span had to be reduced and that two compressor cases, in series, were required for proven reliability
- Costs of second case were assembled along with supporting data and cost figures for exposure to reduced availability and benchmarks of problems experienced with the one case option (this 'lesson learned' information was obtained from experienced plant maintenance and operations personnel)
- The management presentation was successful and additional $5mm was approved for purchase and installation of the second compressor casing

Fig 1.2.3 • Refrigeration compressor selection case history

Best Practice 1.3

Screen preferred vendors list by corporate and plant experience for the project requirements.

Review potential vendors, experience against specific process conditions for the project to confirm that they have experience for this project.

Obtain past project and field experience details for each potential vendor from corporate and plant machinery specialists.

Do not hesitate to eliminate a potential vendor for this project based on lack of experience.

If a vendor is eliminated from quoting, discuss the reasons openly with them, and ensure the vendor that this elimination will not affect future project proposals and is in their interest in terms of proposal cost savings.

Lessons Learned

Not screening vendors for experience for the project at hand will result in schedule delays in the selection process and lower unit reliability.

Not properly screening vendors for experience for the project specific conditions will result in:

- Excessive time spent in reviewing vendor bids only to disqualify them based on experience
- A large exception to specifications list resulting in additional review time
- The potential of accepting an unproven machine for the application, based on a lower quoted price
- Reduced reliability during operation, resulting in large revenue losses.

Benchmarks

This best practice has been used since the 1970s, and ensures minimum machinery review time, optimum safety and reliability, and maximum revenue over the life of the process. It has been incorporated globally in all upstream and downstream projects. It has resulted in minimum machinery design times, minimum change orders, smooth FATs (Factory Acceptance Tests) and trouble free start-ups. These facts have resulted in reduced project costs and schedules and minimum start-up times, resulting in early start-ups and increased product revenue.

B.P. 1.3. Supporting Material

Key screening factors for vendor experience for the specific project requirements are noted in Figure 1.3.1.

Once these facts are obtained, a list of acceptable vendors for this specific project can be prepared. It is important to note that each specific project will have different requirements and some vendors will not be in the same position as their competition in terms of design experience and/or manufacturing capability.

- Past project experience – design errors, manufacturing problems, delays, etc.
- Past field experience – availability, maintainability, field support, etc.
- Vendor's reference lists for the specific project application detailing flow, head, efficiency, etc.
- Networking with industry peers (API, symposiums, etc.)

Fig 1.3.1 • Determine potential vendor capabilities

The decision not to use a certain vendor may be difficult, because of past associations, but it is in the end user's interest to only select the vendors that have the most design and manufacturing experience for a specific application. Vendors not having the required level of experience for this application should be informed immediately, to save them the high costs of quoting. It should be explained that the decision taken is not an expression of the quality of their design and manufacturing, but of their relative experience level for this specific application, and this will definitely not impact future business opportunities.

After the potential vendors are determined, the degree of risk for this application must be defined to determine when and if audits are required. Figure 1.3.2 presents these considerations for critical un-spared machinery.

Risk classification	Action
Prototype	Design and manufacturing pre-screening in pre-FEED phase
Multi-component inexperience	Design audit in bid phase**
Single-component inexperience	Design audit at coordination meeting**

** = Manufacturing audit may be required based on design

Fig 1.3.2 • Risk classifications and action

Best Practice 1.4

Select and design each equipment item in the train for optimum safety and reliability.

The safety and reliability of the equipment train is directly affected by each item in the train (unit).

Consider the proper design and selection of each train item. Devote the same attention to design and experience review for each train item.

The equipment items contained in any train are:

- The driver
- The driven equipment
- The transmission devices (couplings, gears and clutches)
- The auxiliary systems (lube, control, seal, cooling, etc.)

The majority of factors that reduce reliability are contained in the auxiliary systems. They should be carefully reviewed in the pre-bid phase and design audited for proper selection and sizing early in the design phase.

Lessons Learned

Failure to consider the proper selection and design of each item in the equipment train will lower train (unit) safety, reliability and revenue.

Not considering the proper selection and design of each train component can reduce safety and reliability and revenue by:

- Driven equipment experience, mis-application and design issues
- Driver experience, mis-application, insufficient power and design issues
- Coupling and/or gear experience, insufficient torque capability, and design issues
- Auxiliary system experience, improper component selection and sizing

Benchmarks

This best practice has been used, especially for critical (un-spared) trains since the 1970s, and ensures optimum safety and reliability and maximum revenue over the life of the process. It has been incorporated globally in all upstream and downstream projects.

Optimum machinery train reliability has resulted from this Best Practice, saving upwards of $2m (minimum) per year for plants with daily revenues of $1m.

B.P. 1.4. Supporting Material

Classifications of rotating equipment

There are four basic function classifications of rotating equipment. Refer to Figure 1.4.1 for a definition.

- Driven
- Drivers or prime movers (provide power)
- Transmission devices
- Auxiliary equipment

Fig 1.4.1 • Classifications of rotating equipment

Each machinery train or unit is made up of all of the four classifications. The safety and reliability of the train is directly related to the proper selection and design of each of these classifications. Failure to consider the proper experience, selection and design of each train component will result in lower train safety and reliability. Table 1.4.1 is a partial listing of some rotating equipment types, grouped according to their classification (function).

Site equipment examples

Shown below are examples of typical site rotating equipment.

Figures 1.4.2 to 1.4.5 show examples of each rotating equipment classification.

Table 1.4.1 Major types of rotating equipment: Site equipment examples

Major types of rotating equipment

I. Driven equipment		**II. Drivers – prime movers**
A. Compressors		A. Steam turbines
1. Dynamic	2. Positive displacement	B. Gas turbines
Centrifugal	Screw	C. Motors
Axial	Rotary lobe	Induction
Integral gear	Reciprocating	Synchronous
	Diaphragm	Vari-speed
	Liquid ring	D. Engines
B. Pumps		Internal combustion
1. Dynamic	2. Positive displacement	Diesel
Centrifugal	Plunger	Gas
Axial	Diaphragm	
Slurry	Gear	
Integral gear	Screw	
	Progressive cavity	
C. Extruders		
D. Mixers		
E. Fans		
III. Transmission devices		**IV. Auxiliary equipment**
A. Gears		A. Lube and seal systems
Helical		B. Buffer gas systems
Double helical		C. Cooling systems
B. Clutches		
C. Couplings		

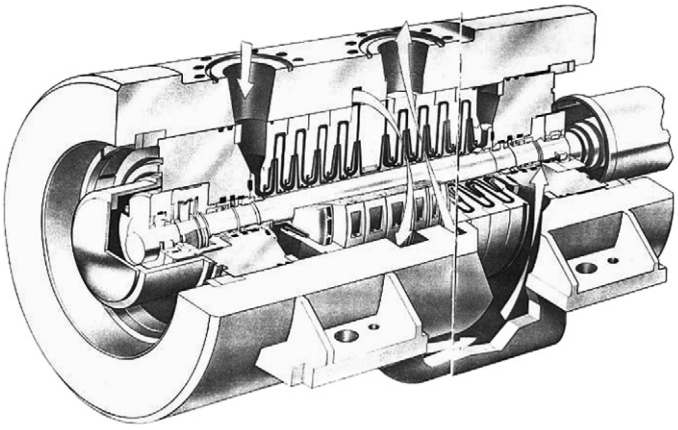


Fig 1.4.2 • High pressure centrifugal compressor (Courtesy of Dresser Rand)

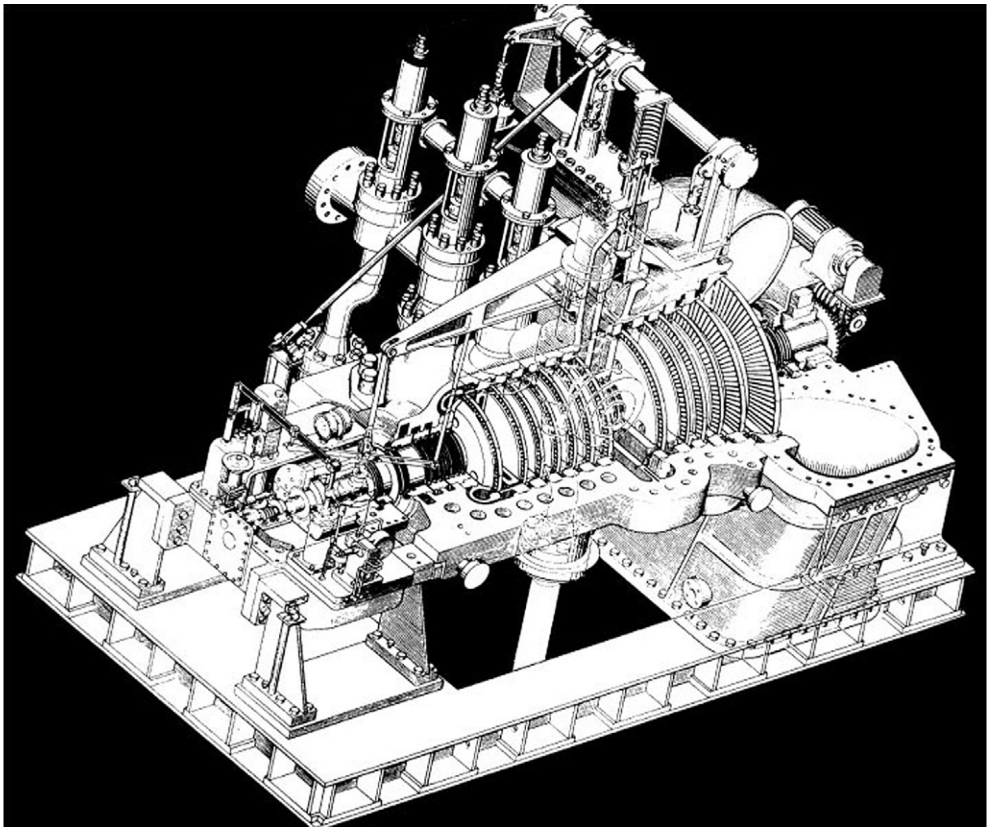


Fig 1.4.3 • Extraction – condensing steam turbine (Courtesy of MHI)

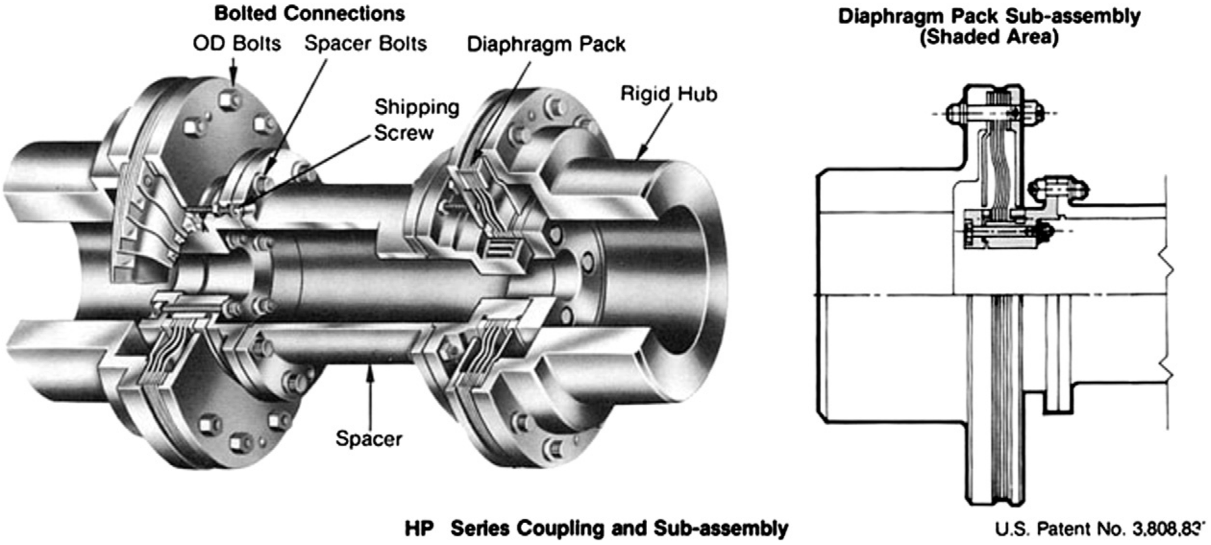


Fig 1.4.4 • Multiple, convoluted diaphragm-spacer coupling (Courtesy of Zurn Industries)

Fig 1.4.5 • Horizontal oil console arrangement (Courtesy of Oltechnique)

Best Practice 1.5

Use proven specification formats for minimum vendor/contractor/end user review time.

Minimize the amount of specifications and use established industry specifications (API, ANSI, etc.).

Use industry data sheets that are completed to contain all details of scope of supply.

Do not leave details up to the vendor if a 'lesson learned' has shown that certain design details are required (e.g. provide a P&ID of the dry gas seal based on plant lessons learned that will be required to be supplied without exception).

Attach a best practice list applicable specifically to this project.

Note in the ITB (Invitation to Bid) that strict compliance with all requirements is mandatory.

Lessons Learned

Excessive specifications, incomplete data sheets and lack of specific best practice requirements extend the project schedule and reduce safety, reliability and plant revenue.

Failure to provide a minimum amount of effective specifications, detailed data sheets and accepted project best practices will result in the following issues:

- Exceeding scheduled time for bid review and vendor recommendations
- Unequal technical content in vendor bids
- Specification interpretation error
- Excessive amount of exceptions to specifications, data sheets and best practices

Benchmarks

This best practice has been used, especially for critical (un-spared) equipment, since 1995 when specifications had grown to enormous sizes and extended project schedules by months! It has been incorporated globally in all upstream and downstream projects since 1995 by Forsthoffer Associates.

Using this 'streamlined' approach has reduced the typical machinery procurement cycle time by 4 weeks or more.

B.P. 1.5. Supporting Material

At this point, all the information required for preparing the project specifications should be available. The challenge will now be to format these specifications such as to ensure complete compliance by all quoting parties, and to minimize the schedule time. The selected format of the specifications will significantly affect the project schedule, in terms of preparation time, vendor response time and contractor/end user/vendor time to review specification exceptions for approval. The effects of the specification format on reliability and project schedule are shown in Figure 1.5.1.

The selected specification format can produce the following effects on equipment reliability and project schedule:

- Interpretation error – due to complexity of specs = reliability risk and delay
- Additional preparation time – for contractor
- Additional review time – for vendors
- Additional meeting and/or document exchange time for contractor/end user review of vendor exceptions to specifications

Fig 1.5.1 • Specification format effects

Let's face it; specifications have become very complex and extensive. It is my experience that this is because most end users send specification drafts out to their plants and affiliates for review prior to publication, and the result is that every participant has to contribute or it does not look good! The end product is a thick specification. Are specifications of this type really necessary? Another consideration regarding complex specifications is that the experience level in all phases of the industry is decreasing and the time required and accuracy of specification reviews is being affected.

Is there a viable alternative? Over the last 15 years, I have been involved with a number of medium to small companies that are beginning to build new facilities, and have the flexibility to decide on their own specification format. Considering that their budget for specification preparation is limited, but they clearly recognize the importance of a sound specification based on industry standards and global best practices, they have generally adopted a strategy similar to that shown in Figure 1.5.2.

- Use an established industry specifications (e.g., API or ANSI)
- Include industry data sheets that are completed to outline all details of scope of supply and company lessons learned
- Attach a best practice list applicable for the specific project only
- Use a global consultant firm to review and comment
- Note in the cover letter to the specification package that strict compliance with all requirements is mandatory – options are not acceptable

Fig 1.5.2 • The streamlined specification strategy

The approach noted in Figure 1.5.2 will only be possible if the credibility of the specialist with project management has been secured and maintained. If there is a reluctance to depart from the established specification format, a recommendation would be to contact other associates who you have met at industry conferences, or search the web for consultants who have experience in this regard and can support your effort. Believe me, there are many benefits to this approach for all parties (vendors, contractors and end users). Benefits to the streamlined specification approach are shown in Figure 1.5.3.

The streamlined specification approach shown in Figure 1.5.3 is really a win for all participants, but for larger, established companies, it definitely is a culture change and will have its price in terms of learning curve and initial expended revenue.

- End user – low probability of misinterpretation, shorter vendor bidding cycle time and reduced review time for vendor exceptions; resulting in lower cost, reduced project schedule and manpower
- Contractor – less preparation and review time; resulting in lower cost, reduced project schedule and manpower
- Vendor – less review and exception preparation time; resulting in lower cost, reduced project schedule and manpower

Fig 1.5.3 • Streamlined specification format – benefits

Any new or different approach will meet initial resistance (a 'paradigm shift') and will require specific 'instructions to the bidder' in brief, clear and concise terms. Figure 1.5.4 presents some guidelines. It has been my experience that the guidelines presented above for a streamlined approach, if endorsed by the project management team and implemented as noted, can result in considerable schedule savings and increased equipment reliability.

- The consideration given to your bid will be based on your compliance with the requirement to list the exceptions applicable to this project only that are necessary due to manufacturing, not cost constraints
- Include the added cost to comply with requirements in your bid and not as an option
- Blanket exceptions to industry specifications (API, ANSI etc.) are not acceptable

Fig 1.5.4 • Important ITB message to all vendors

Best Practice 1.6

Carefully state instructions to quoting vendors in the Invitation to Bid (ITB) document.

The ITB is the only communication sent to quoting vendors prior to discussions and/or meetings.

Clear instructions are essential to minimize vendor/contractor/end user discussions, and will greatly reduce the time needed to determine the successful bidder.

Be sure to include final accepted specifications, data sheets, best practices accepted for the project and especially pre-bid (bid clarification) meeting instructions.

Lessons Learned

Absence or lack of ITB vendor instructions results in endless meetings, discussions and emails.

Not accurately stating vendor instructions in the ITB will result in the following issues:

- Exceeding scheduled time for bid review and vendor recommendations
- Unequal technical content in vendor bids
- Potential reliability issues resulting from lack of a detailed bid review (bid clarification meetings)
- Excessive amount of exceptions to specifications, data sheets and best practices

Benchmarks

This best practice has been used since the 1970s, especially for critical (un-spared) equipment, and ensures optimum safety and reliability and maximum revenue over the life of the process. It has been incorporated globally in all upstream and downstream projects.

This approach has resulted in a 'teamwork' spirit between EP&Cs, vendors and end users, because all parties know the rules and work together towards the stated objectives in the ITB Instructions.

B.P. 1.6. Supporting Material

- Incorporate all project team accepted items (design audit, best practices, pre-bid meetings, test requirements, etc.)
- Include pre-bid meeting instructions (when, where and who attends)
- Include design audit details (when, where and who attends)
- Define discipline and experience requirements for all participants in all scheduled meetings
- Note penalty for non-compliance (e.g., bid not accepted)

Fig 1.6.1 • ITB instructions to vendors

Best Practice 1.7

Machinery specialists must understand the processes in which the machinery will operate.

Review the Process Flow Diagram (PFD) and Process and Instrument Diagram (P&ID) for each process on the project.

Discuss the PFD and P&ID for each process with experienced process engineers from the process licensors, project team and/or plant process engineers and operators.

Define the potential upset and unusual process conditions with the experienced process engineers and operators.

Be sure that all upset and unusual process conditions are defined on the appropriate equipment data sheets.

Require that an experienced process engineer and operator be part of the machinery project team, and that they attend all planning sessions, specification and data sheet reviews, as well as all vendor meetings during the pre-selection, design and test phases.

Lessons Learned

It is the writer's experience that approximately 80% of machinery failure root causes are due to process variations that are not anticipated in the design phase.

Detailed RCFAs (Root Cause Failure Analyses) show that the majority of root causes of failure lie in unanticipated process condition changes. Failure to incorporate these upset and unusual conditions on the appropriate equipment data sheets can:

- Extend project schedule by requiring machinery re-design
- Result in significant cost adders for machinery re-design
- Reduce machinery safety and reliability
- Extend plant start-up

Benchmarks

Since the late 1990s, due to increased process unit size and potential daily revenue losses, we have required detailed process reviews between machinery specialists, process engineers and operators prior to the issue of specifications and data sheets. This action has been implemented for all projects large and small.

Results from this best practice have been:

- Elimination of changes after issue of the purchase order
- Reduced machinery design time and early machinery delivery
- No surprises during plant start-up

Best Practice 1.8

Require that critical machinery pre-bid meetings be properly defined and carried out.

Pre-bid meetings require that all technical details are discussed, with appropriate design changes, before a price is quoted.

These meetings are not the same as bid clarification meetings, which review the vendor's bid after a price is quoted.

Pre-bid meetings, held separately with each quoting vendor, ensure that each vendor's scope, design experience and exceptions to specifications will be similar.

Vendors must be advised in the ITB of the requirements, time and agenda for these meetings in order to minimize project schedule.

Lessons Learned

Failure to have pre-bid meetings can reduce critical machinery safety and reliability, and significantly increase bid evaluation time.

Clients that have renounced pre-bid meetings, and have opted for bid clarification discussions (endless emails and conference calls) or meetings (numerous meetings) have had the following issues occur during the later stages of the project, and during field operation:

- Delayed bid selection – adding weeks or months to the schedule
- Cost adders for items that were not defined or exceptions to specifications that were not resolved during bid clarification
- Delays in the manufacturing schedule for incorporation of items not resolved during the bid phase
- Lower field reliability and possible safety issues resulting from selection of the low cost bidder

Benchmarks

The pre-bid meeting approach has been used since 1975 in all critical equipment selection since 1975. This has produced machinery of optimum safety and reliability (compressor trains = 99.5%+). Pre-bid meetings have been used in all of the following industries globally:

- Upstream oil and gas – offshore and onshore
- Refining
- Chemical
- Co-generation

B.P. 1.8. Supporting Material

Once the ITB has been released to the quoting vendors, the work begins. The first order of business is to prepare for the audits required at this stage. If the equipment in question is a prototype, design and manufacturing audits have already been initiated and will be ongoing. If the equipment contains multiple major components that do not have field experience, audits are required in this phase. These facts are shown in Figure 1.8.1.

- Finalize audit results and prepare project team recommendations for prototype class equipment
- Interview quoting vendors and determine requirement for design and/or manufacturing audits during pre-bid phase for major equipment with multiple component inexperience

Fig 1.8.1 • Design and/or manufacturing audit requirements in the pre-bid phase

The concept of pre-bidding is very powerful and rewarding to all three parties in the bid process – vendors, contractors and end users. Pre-bidding requires that all technical details are discussed, and that appropriate changes for optimum safety and reliability are made before a price is quoted. The advantages of this approach are presented in Figure 1.8.2.

Eliminates competitive pressures on the vendor by:
- Allowing technical review before price
- Assuring the same scope for each supplier
- Assuring offering of the highest safety and reliability
- Being performed regardless of risk classification

Fig 1.8.2 • Technical discussions before $$$$$

The pre-bid meeting is frequently called a bid clarification meeting. This title can be misleading and may not have the same advantages as a pre-bid meeting. The significant differences are noted in Figure 1.8.3.

- Pre-bid meetings – are conducted before a price is quoted and allow for modifications to a technical offering
- Bid clarification meetings – are conducted after a price is quoted and may not allow for modification to a technical offering

Fig 1.8.3 • Bid clarification vs. pre-bid meeting differences

It is most important to confirm the requirements and details of the pre-bid meeting with the contractor at the beginning of the project. Figure 1.8.4 presents the benefits of conducting a true pre-bid meeting, and not a bid clarification meeting.

- The highest equipment reliability
- The lowest life cycle cost
- Equal scope of supply for each vendor
- Shortest bid evaluation cycle time

Fig 1.8.4 • The pre-bid meeting ensures

A pre-bid procedure fact summary and a typical agenda for a compressor train are contained at the end of this chapter. It is recommended that this information be used to justify these meetings with the project management team as early as possible in the project, preferably in the pre-FEED phase.

Due to competitive pressures, past union agreements and high in-house manufacturing costs, vendors have been forced to use numerous sub-suppliers for major component and auxiliary system manufacture and in some cases, design. This approach exposes the end user to potential delivery delays due to sub-supplier manufacturing, quality and schedule issues. These important facts are presented in Figures 1.8.5 and 1.8.6.

Vendors frequently use sub-suppliers for:
- Reduced component costs
- Reduced vendor machine shop investment
- Greater schedule flexibility
- Reduced in-house shop load

Fig 1.8.5 • Who really manufactures it?

- Component scrap due to inexperience
- Component scrap due to improper machine tools
- Component scrap due to improper handling
- Poor or non-existent inspection
- Delay in shipment

Fig 1.8.6 • Potential sub-supplier issues

Based on the potential sub-supplier problems noted above, when should they be audited? The suggested action is noted in Figure 1.8.7.

Audit sub-suppliers when:
- Experience for similar components is low
- Equipment risk class is high
- End user 'lessons learned' warrant

Fig 1.8.7 • Audit sub-suppliers

The recommendation therefore is to always have vendors define their major sub-suppliers and their experience during the pre-bid phase. Please refer to Figure 1.8.8.

- Casing
- Impellers and/or blades
- Diaphragms
- Shaft
- Baseplate
- Auxiliary systems
- Control panels

Fig 1.8.8 • Always require definition of sub-vendor and experience for the above

At this point, all details concerning vendor experience, scope, exceptions and sub-supplier experience have been identified. If the objectives of the pre-bid phase and any required audits have been met, the bid evaluation phase will be short and easy, since there will be a true 'apples to apples' comparison, and the lowest price vendor can be selected without any additional meetings or discussions.

Suggested vendor pre-bid meeting details letter

The following are suggested letter contents. Comments for consideration are noted in **bold**.

Please be advised that you will be asked to attend a pre-bid meeting at ___________ (Your or Contractor's or Company … decision required) offices.

Note: If possible, the meeting should be held at the vendor's offices. This is advisable since more experienced specialists are immediately available to answer any questions that may arise. This decision should also be influenced by the machinery risk classification. The higher the risk, the more important the vendor office meeting is.

The pre-bid meeting will take place approximately _______

(**2–4 weeks after receipt of bid and must be coordinated with the project team … note this decision will also be influenced by the machinery risk class**).

Only technical details will be discussed. Please bring the technical, un-priced proposal for the equipment that you will quote (trains include compressor, gear (if applicable) turbine and auxiliary systems). Your representatives at the meeting must include an experienced application instrument engineer and any other personnel you require.

The meeting objective will be to qualify your bid technically, based on component experience, and to fully define scope of supply and approve exceptions to specifications. If necessary, the technical aspects for your equipment may change as a result of the meeting discussions. In addition to our (contractor) equipment specialist the end user specialist ________ **(or other assigned engineer**) will participate in these meetings. We emphasize that it is in your interest to bring the most qualified personnel to the meeting, since this will be the only technical meeting prior to the final bid.

The following additional audits may be required as a result of your bid details and the use of major sub-suppliers.

____

(The end user to identify sub-suppliers for manufacturing, handling and shipping audits based on vendor bid details)

At the conclusion of the meeting, all details will be summarized and you will be asked to submit your priced proposal, in accordance with the technical details, scope of supply and approved exceptions to specifications agreed to in the meeting. Your proposal will be required in ________ weeks after the meeting.

(Normally 2 weeks but may be longer based on complexity of equipment offering and machinery risk classification.)

The pre-bid meeting agenda is attached.

Pre-bid procedure fact summary

The following is a brief summary of the salient procedure facts:

1. **Required personnel experience** – experienced rotating equipment specialists from contractor, supplier(s) and client are required to participate in the pre-bid meetings. Note: end user 'in-house' specialists are required.
2. **Individual supplier meetings** – individual meetings are held with each supplier, using notes from previous meetings, to ensure equal supplier experience, scope and exceptions to specifications.
3. **Meeting duration** – Anticipated to be 1–2 days per major equipment train, depending on machinery risk classification. Note: this includes compressors, drivers and auxiliaries. Please refer to the typical agenda below.
4. **Typical pre-bid meeting activity** – Technical details are reviewed using agenda requirements, to ensure proven component experience (with impellers, diffusers, rotor response, bearing, seal, and auxiliary system etc.), and to scope compliance and acceptable exceptions to specifications. Modifications are made, as necessary, to ensure that each vendor is offering proven components within acceptable design limits.
   Manufacturing capabilities are confirmed and sub-suppliers for all major components and auxiliary systems are identified, and their experience is confirmed for similar component manufacture.

At the conclusion of the meeting, notes are reviewed and each vendor is instructed to submit a final priced proposal, in full accordance with meeting notes that will be used for the bid evaluation.

Depending upon machinery risk classification, some additional end user in-house and/or independent third party design checks may be required. In addition, separate vendor and sub-supplier machining, handling and shipping capability audits may be required.

Typical compressor train pre-bid meeting agenda

Please note that the following agenda will be followed for each of the compressor trains being offered. Note: 1–2 days will be required for the meeting to review all details based on unit risk classification.

1. Compressor experience review (vendor to include necessary reference charts, tables etc.)
   - Casing experience and review of compressor layout drawing
   - Impeller experience (flow and head coefficient)
   - Individual impeller curve (location of rated point to impeller best efficiency point)
   - Impeller stress
   - Rotor response
   - Stability analysis (if applicable)
   - Bearings – surface speed, load and experience
   - Thrust balance
   - Seals – surface speed, balance forces and experience
   - Surge control and process control system
2. Steam turbine or motor experience review (vendor to include necessary reference charts, tables, etc.)
   - Turbine casing experience and review of layout drawing
   - Stage nozzle and blade experience (profile, velocity ratio, BTU/stage)
   - Blade attachment method and blade stresses
   - Campbell and Goodman diagram review
   - Rotor response

- Bearings – surface speed, load and experience
- Thrust balance (reaction and hybrid types)
- Shaft seals
- Transient torsional response experience review (synchronous motors)
- Control and protection system

3. Gear experience (if applicable) (vendor to include necessary reference charts, tables etc.)

- Gear box experience review and review of layout drawing
- Review of gear data sheet
- Gear calculation review (in accordance with API 613)
- Bearings – surface speed, load and experience
- Thrust loading – single helical gears
- Pitch line velocity review

4. Auxiliary system experience (lube, dry gas seal and control oil system)

- Review of P&IDs
- Review of API 614 data sheets
- Review of typical arrangement drawings
- List of experienced system sub-suppliers
- Review of proposed dry gas seal supplier information

5. Scope of supply for compressor train (all components and auxiliaries) review

6. Compressor train (all components) exceptions to specification

7. Meeting summary and action required

Note: Based on machinery risk, the following 'design checks' may be required:

- Aero-dynamic
- Thermodynamic
- Rotor response
- Stability analysis
- Seal balance
- Thrust balance
- Bearing loading
- Control system simulations
- System layout maintenance accessibility

Best Practice 1.9

Require from the EP&C (contractor) effective bid tabulations that minimize project time and assure machinery selection based on life cycle cost.

Input required bid tabulation details to project team management for review and transmittal to the EP&C early in the FEED phase (Front End Engineering Design).

Require the bid tabs to be minimized in size, but include the critical items necessary for evaluating bids on a life cycle cost basis.

Use the 'typical bid tab' format contained in this section as a guide.

Lessons Learned

Use of the EP&C typical bid tabulation format extends the bid cycle and evaluates only on a capital cost and not life cycle cost basis.

Using this format will result in the following issues:

- Extension of the bid cycle time by 2 or more weeks
- Evaluation based on technical items but on their capital cost only
- A vendor selection that will not result in the highest level of plant safety, reliability and revenue for the life of the process unit

Benchmarks

I have used this approach since 2007 in all critical equipment bid tabulations, and have gained approval from project teams for its use. As the size of plants (mega) and daily revenue has increased significantly in recent years, end user clients have seen the advantages in this approach over the "standard industry bid tabulations" that were used in the past. This approach has been used in the following projects since 2007:

- Upstream – gas plant booster compressor project
- LNG plant feed gas compressor project
- LNG plant mixed refrigeration gas compressor project
- Proposed Bio – Fuels compression equipment project

One of the projects mentioned above allowed the selection of a machine with a capital cost of +25% based on yearly revenue, power cost and maintenance savings equal to the capital cost increment.

B.P. 1.9. Supporting Material

Bid evaluations

It is an established fact that the contractor will prepare the bid tabulation and present it to the end user for acceptance. However, it has been my experience that the contractor bid tabulation is usually nothing more than a scope of supply list, and not a true technical comparison of the offered equipment.

As noted in B.P. 1.8, if the pre-bid meetings are conducted correctly, the scope and the vendor exceptions to specifications will be essentially the same. Based on my experience, I require

Table 1.9.1 Bid tabulation key fact list

- The end user should review the contractor proposed bid tab format
- A technical check list should be included to quickly identify advantages
- Carefully consider the weight given to power costs and their tendency to influence selection of equipment of lower availability
- Detail exceptions to specifications and require vendors to list only exceptions specific to the project
- Encourage contractor to minimize size of the bid tabulation since scope should be equal based on pre-bid meeting results

Table 1.9.2 data sheets from orig doc

SR	ITEM	TUTORIAL DESCRIPTION	%	EXPLANATION	Vendor #1	Total Rating	Vendor #2	Total Rating	Vendor #3	Total Rating	Vendor #4	Total Rating
	ROTATING MACHINERY EVALUATION				SUMMARY EVALAUTIONS							
	COMPRESSOR											
1	Model	Compressor Vendor Model with definition of designation EG: 6 B 44 = 44' crossover diameter, Barrel, 6 impellers										
2	Rated inlet flow	Compressor Vendor to list rated case inlet flow										
3	Rated Head	Compressor Vendor to list rated polytropic head										
4	Rated Comp. Power	Compressor Vendor to list rated compressor power including all compressor mechanical losses										
5	Rated Efficiency	Compressor Vendor to list rated compressor efficiency (Polytropic)										
6	Rated Speed	Compressor Vendor to state rated speed										
7	Surge Margin	Compressor Vendor to state % of surge flow from rated flow at rated speed minus 100%	1	5 = largest %, 1 = lowest %								
8	Head Rise to Surge	Compressor Vendor to state % of head at surge divided by head at design flow at rated speed	1	5 = largest %, 1 = lowest %								
9	Case Experience	Compressor Vendor to list references for units installed and operating 2 years of same model (case + impeller #)	1	5 = max. ref., 1 = min. ref.								
10	Impeller Head/stage	Compressor Vendor to list Polytropic head for each stage (impeller) at rated conditions (Isentropic head not acceptable)	2	5 = lowest hd/stg., 1 = highest								
11	Head Coefficient	Compressor Vendor to define head coefficient and list head coefficient for each Impeller										
12	Flow Coefficient	Compressor Vendor to define flow coefficient and list flow coefficient for each impeller	1	5 = lowest flow coeff., 1 = high								
13	Head Coeff experience	Vendor to plot head coefficient for each stage and show experience points for 2 year operating experience for same coeff.	1	5 = most exp., 1 = least								
14	Flow Coeff experience	Vendor to plot flow coefficient for each stage on plot with 2 year operating experience for some coeff.	1	5 = most exp., 1 = least								
15	Position of BEP	Vendor to show % of rated Impeller operating point to impeller BEP - each impeller	2	5 = closest to BEP, 1 = farthest								
16	Impeller max stress	Vendor to list maximum impeller stress, state which impeller and list operating stress for this Impeller	2	5 = greatest margin, 1 = least								
17	NC1 & NC2	Vendor to list first and second critical speeds										
18	% NC2 > Rated Spd.	Vendor to list % of NC2 (based on oil film stiffness and damping) above rated speed	2	5 = greatest margin, 1 = least								
19	Shaft Stiffness ratio	Vendor to plot offered shall stiffness (brg. Span/ dia. @ impellers) vs. 2 year operating similar ratio's	1	5 = most experience, 1 = least								
20	J Bearing experience	Vendor to list offered J bearing type, diameter and show 2 year operating experience for some rubbing speed and area	1	5 = most experience, 1 = least								
21	Journal brg. Load	Vendor to list % of journal brg. Load to maximum allowable load	1	5 = lowest load, 1 = most load								
22	T Brg experience	Vendor to list offered T bearing type, diameter and show 2 year operating experience for same rub spd and area	1	5 = most experience, 1 = least								
23	Thrust brg. Load	Vendor to list % of thrust bearing Load to maximum allowable load	1	5 = lowest load, 1 = most load								
24	Total tube oil flow	Vendor to list total tube oil flow to compressor, gear and turbine	1	5 = highest flow, 1 = lowest								
25	DGS experience	Vendor to list offered DGS Mfg., type, diameter and show 2 year operating experience in offered case	2	5 = most experience, 1 = least								
	GEAR											
26	Model	Gear Vendor Model with definition of designation										
27	Centerline experience	Gear Vendor to list references for units installed and operating 2 years of same model	1	5 = most experience, 1 = least								
28	Maximum power	Gear Vendor to list maximum power transmission capability	1	5 = highest, 1 = lowest								
29	Gear Efficiency	Gear Vendor to list Gear Efficiency	1	5 = highest, 1 = lowest								
30	Power Loss	Gear Vendor to list gear power loss	1	5 = lowest, 1 = highest								
31	Service Factor	Vendor to list service factor in accordance with API 613	2	5 = highest, 1 = lowest								
32	Torque experience	Vendor to list gear torque transmission experience	1	5 = greatest #6, 1= lowest								
33	Pinion Brg. Type	Vendor to list offered pinion bearing type end provide 2 year field operating list for similar bearing	1	5 = most experience, 1 = least								
34	Powers @Turb. Flg.	Vendor to list power for rated condition at gas turbine flange	2	5 = lowest power, 1 = highest								
	GAS TURBINE											
35	Model	Vendor Model with definition of designation										
36	Power @ site cond	Vendor to state output shaft power net (after all power take offs)	2	5= highest power, 1 = lowest								
37	Heat Rate @ site	Vendor to state site Heat rate	2	5= lowest heat rate, 1 = high								
38	Speed	Vendor to state rated shaft speed and show experience	1	5= most experience, 1=lowest								
39	Model experience	Vendor to state model operating experience for 2 years minimum at similar site conditions for mech. Drive experience	2	5= most experience, 1= lowest								
40	Model References	Vendor to list two model references with contact number for similar site conditions	1	5= good report, 1=worse report								
41	HGP Interval	Vendor to state hot gas path part change out interval	2	5 = longest, 1 = shortest								
42	HGP Cost	Vendor to list present total cost (parts + maintenance time) for HGP replacement	2	5= lowest cost, 1 = highest								
43	Gas useage/yr.@R	Vendor to list total gas usage per year for operation at rated flow and gas turbine site conditions	2	5= lowest usage, 1 = highest								
44	Sched.down time/yr.	Vendor to list scheduled downtime per year and also downtime per year in which there will be a HGP Change out	2	5= minimum hours, 1 = max.								
45	Unsched. DT/yr.	Vendor to list unscheduled downtime per year based on reliabilities of compressor, gear, turbine and support systems	2	5= minimum hours, 1 = max.								
46	Life Cycle OP Cost	Capital Cost +Gas Usage cost* + HGP cost* + Scheduled Downtime* + Unscheduled Downtime* *= based on 15 yr. interval	20	5= lowest LCC, 1 = max.								
47	**SCOPE**	List required scope items that were are not supplied	14	5= max. scope, 1 = min.								
48	**EXCEPTIONS**	List Vendor Exceptions taken that are not acceptable	10	5= minimum excep., 1 = max.								
49	**NEQS**	List vendor Selected machine Emissions	2									
50	**Maintenance**	Vendor to provide compliance to Maintenance Contract requirements. Also give brief details on possibility of site maintenance of machines.	4									

100% RATING CRITERIA TOTAL=500 PTS.

NOTES:

(1) WHERE INFORMATION IS NOT PROVIDED BY THE VENDORS, THE LOWEST RATING FOR THAT ITEM (1) WILL BE USED
(2) EVALUATION RATING (5-1 SCALE) WILL BE PER PROJECT TECHNICAL EVALUATION DOCUMENT
(3) THE REQUIRED NUMBER OF POINTS OUT OF 500 FOR TECHNICAL ACCEPTANCE WILL BE DECIDED BY TEAM AFTER REVIEW OF ALL TECHNICAL BIDS
(4) HEADS PER STAGE, 11,000 FT-LBF/LBM ARE NOT ACCEPTABLE WITHOUT TEAM APPROVAL
(5) NO EXPERIENCE IN ANY CATEGORY WILL BE CAUSE FOR REJECTION OF TECHNICAL BID AFTER DISCUSSION WITH VENDOR
(6) ROWS 42 & 46 WILL BE DETERMINED AFTER OPENING OF COMMERCIAL BIDS
(7) ALL EQUIPMENT DATA SHEETS SHALL BE COMPLETED AS MUCH AS POSSIBLE

the contractor to be sure to include a 'technical check list' section in the bid tabulation, for a quick confirmation that the best technical alternative, that will produce the highest availability and therefore highest profits, is selected. I have included an example of a technical check list for a compressor train in the back of this chapter.

Based on my experience, I have included a bid tabulation key fact list in Table 1.9.1 and an acrual Bid tabulation in Table 1.9.2.

Best Practice 1.10

Conduct effective pre-award meetings to confirm that vendor final proposal details will be incorporated into the purchase order.

Have EP&C issue a detailed agenda for vendor approval a minimum of 2 weeks prior to the meeting.

Confirm that the scope and all exceptions to the specifications are accepted prior to issue of the purchase order.

The meeting can be held in the EP&C offices, but should include the vendor personnel involved in the proposal stage, as well as the assigned vendor project manager as a minimum.

Establish the project schedule, coordination meeting dates and document transmittal schedule.

Lessons Learned

Poor planning and absence of key individuals in the pre-award meeting will affect the project schedule, and can result in cost adders and delivery schedule extensions.

Experience shows that there are always pre-award meetings, but if they are not conducted properly, the following issues will arise:

- Project and possible manufacturing schedule extensions to handle differences between the agreed terms and vendor accepted purchase order terms
- Change in vendor exceptions to specifications, resulting in cost adders and possible delivery delays
- Lack of 'trust' between vendor, EP&C and end user, causing a long difficult project for all parties

Benchmarks

The writer has used the above approach in all critical equipment projects since 1990, resulting in 'smooth, issue free projects', without significant cost adders and schedule delays. Pre-award meetings have been used in all projects handled by FAI since 1990:

- Upstream oil and gas – offshore and onshore
- Refining
- Chemical
- Co-generation

B.P. 1.10. Supporting Material

Pre-award meeting

After completion of the bid tabulation and approval of the selected vendor, confirmation of the approved vendor's proposal details is required prior to the award of an order. In my experience, there have been many times when the vendor's marketing and engineering departments have had significant differences of opinion in regard to 'what was actually sold'. The purpose of the pre-award meeting, therefore, is to confirm the content of the order before a contract, in order to eliminate additional costs and delays during the equipment engineering and manufacturing phases. Key facts regarding the pre-award meeting are presented in Figure 1.10.1.

A suggested outline for a pre-award meeting is given in Figure 1.10.2.

- Purpose – to ensure agreed compliance
- With who? – the recommended vendor
- When? – ASAP after the bid tab is approved
- Where? – depends on complexity and risk class
- Confirm – marketing to engineering continuity

Fig 1.10.1 • Pre-award meeting 'key facts'

- Assure the attendance of vendor marketing and project engineer
- Agenda to be prepared by contractor/end user
- Agenda contents:
  - Scope of supply confirmation
  - Clarification and agreement of all exceptions to specifications
  - Resolution of pending design audit issues
  - Confirmation of price and delivery schedule
  - Agreement of minutes and action points

Fig 1.10.2 • The pre-award meeting agenda

Best Practice 1.11

Conduct effective vendor coordination meetings (VCMs) to confirm scope and design are in complete accordance with the purchase order.

Have the EP&C prepare the agenda, pass it by the end user team for review and issue to the vendor a minimum of 2 weeks prior to the meeting.

Conduct the meeting in the vendor's works and have it in the early stages of the design engineering phase (approximately 6 weeks after order placement).

Be sure that the end user machinery specialist, an experienced operator and maintenance engineer attend as a minimum.

Lessons Learned

Poorly planned vendor coordination meetings (VCMs) and absence of key individuals will result in cost adders and delivery schedule extensions.

If the VCM is not conducted properly, the following issues could arise:

- Project schedule extensions resulting from engineering changes by the EP&C (piping changes, foundation changes, etc.)
- Project schedule extensions resulting from engineering changes by the vendor (machine changes due to process condition changes, mechanical component changes arising from differences in interpretation of exceptions to specifications, etc.)
- Cost adders
- Lack of trust between EP&C and vendor

Benchmarks

Use of the above approach in all Critical Equipment Projects since 1970 has resulted in 'smooth, issue free projects' without significant cost adders or schedule delays. VCM's conducted in this best practice format have been used in all projects that I have handled for the following projects:

- Upstream oil and gas – offshore and onshore
- Refining
- Chemical
- Co-generation

B.P. 1.11. Supporting Material

The coordination meeting

After the order is placed, the coordination meeting is the first contact between the contractor, the end user and the vendor. This meeting is usually held approximately 6 weeks after the order placement and should be held at the vendor's shop.

In my experience, the effectiveness of this meeting is significantly increased if the end user's rotating equipment specialist and/or consultant, senior operator and maintenance engineer are in attendance. Depending on the project management team, it may be necessary to 'campaign' for the attendance of these valuable people. There can be the suspicion that incorporating these individuals will add additional cost to the job. It is my strong opinion that the addition of these individuals will reduce significantly the life cycle cost of the job by incorporating lessons learned and best practices into the job. Refer back to the pre-FEED phase of the job and note that the same individuals were asked to contribute input to the job in this phase.

My 'best practice' is to include a senior operator and maintenance man (millwright) in all phases of the project from pre-FEED up to and including the test phase. This ensures that all company 'lessons learned' are turned into 'best practices' for the project.

The key facts for the VCM (vendor coordination meeting) are presented in Figure 1.11.1.

Vendor coordination meeting key facts are shown in Figure 1.11.2.

A VCM checklist is included in Figure 1.11.3 for your use, to ensure that all important facts are covered. Depending upon the risk class of the equipment, this may very well be the last chance for vendor engineering contact prior to the shop test phase.

- Purpose – to confirm scope and design
- Design confirmation amount is proportional to the risk class
- If there is any component inexperience, details must be finalized now!
- Location – vendor's shop
- Attendance by: vendor specialists, sub-supplier specialists, contractor specialists, end user specialists
- Timing – approximately 6 weeks after order
- Duration – 2–4 days based on complexity and risk class

Fig 1.11.1 • The VCM – key facts

- Agenda by vendor approved by contractor/end user
- Agenda to be issued for review 2 weeks before meeting
- Assure that all required design reviews are included
- Inform project team in advance of required attendance
- End user should take detailed minutes
- Review all minutes and acceptance required by all parties, with action point responsibilities and required dates noted prior to adjournment

Fig 1.11.2 • VCM agenda – key facts

- Review and confirm process conditions
- Review and confirm aero, thermo and mechanical design
- Conduct any required design and/or manufacturing audits
- Confirm all major connection locations
- Review machine and auxiliary layouts for maintenance accessibility
- Review preliminary test agenda
- Resolve any outstanding specification issues
- Review vendor and sub-supplier QC procedures (there may be a separate meeting for this activity)

Fig 1.11.3 • VCM agenda checklist

Best Practice 1.12

Determine in the pre-bid project phase if and when design audit meetings are required.

Determine the critical machinery risk class as soon as the project is announced, in the pre-feed phase, using the guidelines contained in the supporting material section of this B.P. It may be necessary to use design audits in the pre-feed phase as a means of vendor acceptability screening.

Determine if vendor sub-supplier audits are required (carefully review 'new international country sources' – China, Middle East, etc.). The machinery industry has recently experienced poor quality sub-supplier work from 'new countries' entering into sub-supplier work.

Lessons Learned

Failure to conduct design audits at the proper time in a project can lead to significant start-up delays, and life-long machinery safety and/or reliability problems.

Since 1990 FAI has been involved with many field troubleshooting assignments (RCFAs). These were required because properly conducted design audits were either not performed or conducted too late in the project.

Benchmarks

The above approach has been used in critical equipment projects since 1990, and particularly since 2000 when MEGA projects became common in the industry. This approach has resulted in projects free of safety and reliability issues, without significant cost adders or schedule delays. Design audits were performed in compressor trains for the following recent projects:

- Refinery hydrocracker recycle compressor
- LNG mixed refrigerant compressor
- Methanol MAC and BAC air separation train
- Ethylene refrigeration compressor

The anticipated revenue savings from properly timed and conducted design audits can easily reach $60m, or much greater if based on a $1m per-day-plant. Design problems that are not determined during an audit can easily cause field delays in excess of two months.

B.P. 1.12. Supporting Material

Design and manufacturing audits

The need for design and manufacturing audits, as previously stated, depends on the equipment risk class, vendor and sub-supplier design, and manufacturing experience level. These audits can be conducted at any phase of the project, but the sooner the better. Prototype equipment requires that audits be conducted during the pre-FEED or FEED phase of the project. Today, most projects are defined as MEGA projects, since the process units are the largest size ever built and most probably will incorporate single equipment trains that are prototype in nature. Therefore, many projects require that design audits (pre-screening) should be conducted as soon as the project starts. Planning and conducting effective supplier design and manufacturing audits requires pre-planning and a significant amount of work, but it is certainly worth the effort in terms of increased profits and reduced project schedule. These salient points are noted in Figure 1.12.1.

When supplier or sub-supplier manufacturing audits are required, suggested action is shown in Figure 1.12.2.

In Table 1.12.1 I have presented a suggested list of what the design audit should include, depending on the risk classification.

- Detailed agenda, well in advance
- Design audit at vendor's offices with follow-up at end user's offices
- Manufacturing audit at vendor's and/or sub-supplier's plants
- End user specialists must participate
- Conduct preliminary end user in-house checks prior to the design audit if possible

Fig 1.12.1 • Vendor audit requirements

- Machining capabilities (max, size capability)
- Balancing capabilities (Low speed and/or high speed, max rotor size)
- Size of assembly area
- Shop load status
- Testing capabilities (gas test, full load test, power limits)
- Handling capabilities (max lift, laydown area)
- Shipping capabilities

Fig 1.12.2 • Manufacturing audit guidelines

Table 1.12.1 Suggested design audit activity

Suggested design audit activity

1. Risk type	1	2	3	4
2. Design checks				
■ Aero-dynamic	X	X	?	*
■ Thermodynamic	X	X	?	*
■ Rotor response	X	X	?	*
■ Stability analysis (if applicable)	X	X	?	*
■ Seal balance	X	X	?	*
■ Thrust balance	X	X	?	*
■ Bearing loading	X	X	?	*
■ Train lateral analysis	X	X	?	*
■ Torsional analysis (if applicable)	X	X	?	*
■ Transient torsional (if req'd)	X	X	?	*
■ Control system simulations	X	X	X	X
■ System layout – accessibility	X	X	X	X

Key
1 = Prototype; X = Required
2 = Multiple component inexperience; ? = Optional
3 = Single component inexperience; * = Not required
4 = Proven experience for all components.

After conducting the appropriate audits, prompt follow-up regarding any action items is required to confirm the acceptance of the supplier and/or sub-suppliers, and to maintain the project schedule. Figure 1.12.3 presents these facts.

- Prepare an executive summary of conclusions
- Immediately present to the project team for approval
- Inform vendors of results
- Prepare vendor follow up meeting agenda

Note action required and follow up as required to maintain project schedule

Fig 1.12.3 • Design audit summary and follow up action

After completion of the required audits, regardless of the project phase in which they were conducted, follow-up document review is essential to confirm that all stated design and manufacturing requirements are met.

In Figure 1.12.4, I have presented a typical design audit meeting agenda to resolve a long term problem with a compressor seal oil system. Always remember to send agendas well in advance to allow the vendor sufficient time to prepare the required material.

Lube/seal oil system

1. Introductions
2. Purpose of meeting
   2.1 Review study results, past modifications/failures and recommendations to assist client in resolving seal oil delta pressure trips on the subject compressor.
      2.1.1 Supply client with recommendations, modifications required and cost and delivery.
3. Results of studies performed in the field
   3.1 Client field reliability study
   3.2 Vendor engineering study
4. Review seal design and requirements
   4.1 Review seal components and function
      4.1.1 Upgrades?
      4.1.2 Modifications?
5. Review seal oil system component design
   5.1 Sizing of components (pumps, coolers, filters, reservoir, etc.)
   5.2 Review valve selection and sizing (including Cv)
   5.3 Review control system
      5.3.1 Upgrades?
      5.3.2 Modifications?
6. Review comments to seal oil system component sizing study
   6.1 List recommendations
      6.1.1 Feasibility and reliability issues
7. Review final recommendations and feasibility
   7.1 Assign tasks and schedules
   7.2 Create final timeline up to delivery of parts and installation
8. Conclusion

Fig 1.12.4 • Design audit agenda

Best Practice 1.13

Timely vendor and EP&C document review.

Finalize document review times in the pre-feed phase of the project, and ensure they are realistic, and will enable comments from plant site personnel to be incorporated.

Review all documents completely, and watch out for vendor and/or EP&C 'cut and paste' documents that are from other projects in error or are generic in nature.

Follow up by reviewing the next issue of the document to ensure that your comments have been incorporated.

I have been involved in projects where the scheduled plant personnel document review response time has been very unrealistic. This has caused important plant review comments to not be incorporated in the machinery design, which has resulted in reliability issues in the field that delayed plant start-up and could have been avoided.

Lessons Learned

Inefficient vendor and EP&C document reviews will cause delivery delays and possible field start-up delays.

Benchmarks

I have used the above approach in all critical equipment projects since 1975. This approach has resulted in safety and reliability 'issue free' projects, without significant cost adders or schedule delays.

B.P. 1.13. Supporting Material

Document review

It goes without saying that document review should definitely be timely, within the project schedule, and accurate. However, in addition there are other pertinent points, which are presented in Figure 1.13.1.

- Assure that required review time frames are realistic and then meet them!
- Thoroughly review all items
- Question all required items and follow-up
- Be especially careful in the final phases of the project to ensure that all required vendor changes have been made

Fig 1.13.1 • Effective document review considerations

Best Practice 1.14

Assure a FAT (Factory Acceptance Test) of maximum effectiveness.

Establish FAT test requirements in the job specifications and confirm during the pre-award and coordination meetings.

Require the FAT agenda to be submitted for approval by the end user a minimum of 2 months prior to the FAT date.

Require a pre-test meeting the day before the first FAT with each vendor.

Assure that the FAT team consists of experienced machinery, operations and maintenance personnel as a minimum.

Require a post-test meeting to determine final acceptance and confirm the format of the format test report.

Lessons Learned

The FAT is the last chance to ensure that the equipment was designed correctly and operates with optimum safety and reliability!

Failure to conduct an effective FAT, from the end user's perspective, will expose the end user to future safety and reliability issues.

My company has been involved with numerous site machinery reliability issues that have resulted from ineffective FATs.

Benchmarks

The above approach has been used in critical equipment projects since the early 1990s, and has resulted in projects free of safety and reliability issues, and without significant cost adders or schedule delays.

B.P. 1.14. Supporting Material

Testing phase

The testing phase is the last phase in terms of vendor and subsupplier design and manufacturing involvement in the project; and it is the last chance to ensure the optimum availability of the finished product.

Remember that all of the equipment addressed in this section is most likely custom designed, and no matter how much accrued design and manufacturing experience is present, the possibility of some abnormality, hopefully a minor one, is high. Therefore, it is imperative that this phase be carefully observed and witnessed by the end user team. Figure 1.14.1 lists some important facts surrounding this phase of the project.

The shop test is an opportunity to:

- Confirm vendor proper design and manufacture
- To match field conditions
- To witness assembly and disassembly using job special tools
- To have plant personnel observe test, assembly, etc. and take pictures for purposes of emergency field maintenance excellence
- Review the instruction book
- Have the assigned vendor field service engineer observe the equipment he will install
- Review all vendor field procedures

Fig 1.14.1 • The shop test

I have included a shop test checklist at the end of this section, which will be useful in planning and executing the shop test phase. Yes, there certainly are many opportunities to ensure equipment reliability during shop test, but there are also a lot of potential lost opportunities if they are not justified to the project team early, during the pre-FEED phase of the project. The potential lost shop test phase opportunities are noted in Figure 1.14.2.

The following opportunities will be lost if they are not justified at project inception:

- Possible full load test
- Unproven component tests
- Attendance at test by plant personnel
- Use of special tools
- Vendor permission for pictures
- Agreement that assigned vendor field service specialists will be present for tests
- Agreement that the instruction book is reviewed
- Agreement for formal field construction meeting to clearly define all vendor procedures from receipt of equipment on site to initial run in of equipment

Fig 1.14.2 • Potential lost test opportunities

The success of the shop test depends on a good test plan that is reviewed by the end user and contractor and modified as requested well in advance of the test. Figure 1.14.3 presents these facts.

I actually began my career in rotating equipment on the test floor. I can still remember how we would see the witnesses come in with the intention of participating completely in the entire test, only to leave for a long 'test lunch' an hour or so later. Why did this occur? Usually because the concerned end user and contractor witnesses did not have the opportunity to review

- The agenda is issued for review two months prior to test
- It incorporates agreed VCM scope
- Compressor performance test conditions are per ASME PTC–10 requirements
- A sample of test calculations and report format is included
- Vendor concurs with all end user and contractor comments prior to test

Fig 1.14.3 • Shop test agenda review – key facts

the test set-up and the procedure prior to the test. As a result, I have always been a proponent of a pre-test meeting.

Is it always required? I think it is, but the detail and timing of the meeting depends on certain factors, noted in Figure 1.14.4.

- If the equipment is prototype
- If the equipment is complex
- If a full load test is required
- If the test facility is new

Fig 1.14.4 • When is a pre-test meeting required?

If it is decided to conduct a pre-test meeting, the key facts are noted in Figure 1.14.5.

- Conduct the meeting prior to the test day
- Send the agenda to the vendor well in advance
- A typical agenda outline:
  - Confirm test agenda requirements
  - Confirm all test parameter acceptance limits
  - Confirm instrument calibration
  - Review test set up or concept drawing
  - Review data reduction methods
  - Confirm all test program agreements

Fig 1.14.5 • Pre-test meeting – key facts

Figures 1.14.6 to 1.14.8 define recommended test activity for the mechanical, auxiliary equipment and performance shop tests respectively.

- Per API and project requirements
- Confirm all components are installed
- Confirm all accessories are installed
- Monitor progress of test, look for leaks, etc.
- Do not accept test until all requirements are met

Fig 1.14.6 • Mechanical test – key facts

At the conclusion of all test activities, there is still important work to be performed. These items are defined in Figure 1.14.9.

What happens if the test is not successful? Approximately 50% of the tests that I have either run, or participated in, over my career have not been successful with respect to one component or more not meeting test requirements. Possible rejection test action is noted in Figure 1.14.10.

- Must be per API and project requirements
- Confirm that the test agenda is followed
- Confirm all components are installed
- Confirm that all required instruments are installed
- Monitor the progress of the test – look for leaks, etc
- Do not accept until all requirements are met

Fig 1.14.7 • Auxiliary system test – key facts

- Per ASME PTC-10 requirements
- Reconfirm test speed is per PTC-10
- Confirm all instruments are calibrated and installed
- Confirm test gas purity
- Agree that conditions are stable prior to each test point
- Confirm vendor's calculations for each test point
- Do not accept until all test requirements are met

Fig 1.14.8 • Compressor performance test – key facts

- Confirm performance results, corrected to field conditions
- Confirm mechanical test acceptance
- Confirm auxiliary system test acceptance
- Inspect components and confirm acceptance
- Agree to any corrective action in writing
- Accept or reject test – any corrective action requires a retest!

Fig 1.14.9 • Post test – key facts

- Immediately provide details to the project team
- Confirm if field conditions can handle the abnormality
- Determine if the `as tested' machine will meet all reliability requirements
- If the decision is to reject, inform the vendor and detail the reasons
- Do not accept unrealistic delivery delays

Fig 1.14.10 • Rejection test action

Finally, do not forget the importance of test report requirements. The test report is a most important document that represents the 'baseline performance of the unit', and will be a benchmark for field operation acceptability. Test report key facts are noted in Figure 1.14.11.

- The shop test is the field baseline!
- The test report must be detailed and complete
- Review the preliminary contents of the report before leaving the test floor
- Obtain the actual test results
- When the final report is received, check the results obtained at test against the final report
- Immediately contact the vendor if there are any differences

Fig 1.14.11 • Test report – key facts

Shop test checklist

1. Scope

Appropriate industry specs included (ANSI, API, NEMA, etc.)
In-house and/or E&C specs included
Project specific requirements

☐ Performance test	☐ All rotors	☐ One rotor
☐ Test (equivalent) conditions		
☐ Field (actual) conditions		
☐ Mechanical test	☐ All rotors	☐ One rotor
☐ Test (equivalent) conditions		
☐ Field (actual) conditions		
☐ Unit test of all equipment (string test)		
☐ No load	☐ Includes auxiliary systems	
☐ Full load	☐ Does not include auxiliary systems	
☐ Use of job couplings and coupling guards		
☐ Testing of instrumentation, control and protection devices		
☐ Auxiliary system test		
☐ Lure oil	☐ Test press	☐ Full press
☐ Control oil	☐ Test press	☐ Full press
☐ Seal oil	☐ Test press	☐ Full press
☐ Seal gas	☐ Test press	☐ Full press
☐ Fuel	☐ Test press	☐ Full press
☐ Flow measurement required		
☐ Time base recording of transient events required		
☐ Use of all special tools during test (rotor, removal, coupling, etc.)		
☐ Shop test attendance (includes assembly and disassembly)		
☐ Site reliability	☐ Site maintenance	☐ Site operations
☐ Review of instruction book during shop test visit		
☐ Test agenda requirements		
☐ Mutually agreed limits for each measured parameter		
☐ Issue for approval 2 months prior to contract test date		

2. Pre-test meeting agenda

Meet with test department prior to test to:

☐ Confirm test agenda requirements
☐ Confirm all test parameters have mutually agreed established limits
☐ Review all instrument calibration procedures
☐ Review test set-up drawing
☐ Review data calculation (data reduction) methods
☐ Define work scopes for site personnel (assembly and disassembly witness, video or still frame pictures, etc.)
☐ Confirm assigned vendor service engineers will be in attendance for:
☐ Assembly
☐ Disassembly
☐ Test

3. Shop test activity

Review and understand test agenda prior to test
Immediately prior to test meet with assigned test engineer to:
☐ Review schedule of events
☐ Designate a team leader
☐ Confirm test team leader will be notified prior to each event
☐ 'Walk' test set-up to identify each instrumented point
☐ Confirm calibration of each test instrument
☐ Obtain documents for data reduction check – if applicable (flow meter equations, gas data, etc.)
During test
Note: coordinate with test personnel to avoid interference
☐ Review 'as measured' raw data for consistency
☐ 'Walk' equipment – look for leaks, contract instrument, piping and baseplate vibration, etc.
☐ Use test team effectively – assign a station to each individual
☐ Ask all questions now, not later, while an opportunity exists to correct the problem
☐ Check vendor's data reduction for rated point – if applicable
After test
☐ Inspect all components as required by the test agenda (bearings, seals, labyrinths, RTD wires, etc.)
☐ Review data reduction of performance data corrected to guarantee conditions
☐ Review – all mechanical test data
☐ Generate list of action (if applicable) prior to acceptance of test
☐ Approve or reject

2

Pump Best Practices

Introduction

The most prolific machinery items in any plant are the pumps. They are spared and are not usually considered as critical equipment items except, of course, when one of the units is down for maintenance and the other item suffers a component failure. When one considers that the majority of pump types are dynamic (centrifugal types) and are 90% or more of the pump asset population, proper attention is not usually given to monitoring the flow rate which changes with process conditions. Operating centrifugal pumps outside of the proper operating flow range will significantly reduce pump and component MTBFs.

This chapter addresses the best practices that will optimize site pump reliability and will minimize repair time and maintenance costs.

Best Practice 2.1

Use plant, company and industry lessons learned to select the proper type of pump.

First, accurately determine all of the operating conditions to be accomodated. Process conditions not anticipated in the selection phase account for 80% or more of pump reliability problems. Then use company guidelines for selection of the pump type if they are available.

Whether company guidelines are available or not, determine the pump type by site and industry lessons learned to avoid plant safety and reliability issues.

Finally, indicate the selected pump type on the data sheet with all pump best practices dictated by plant, company and industry lessons learned noted.

Lessons Learned

Not using lessons learned from past problems to select the type of pump required will lead to lower pump reliability.

The writer has continuously experienced pump reliability issues that were present in similar pumps in the plant but were not corrected in new projects or revamps. It is most important to list all pump reliability issues and make them known to plant management and project teams as early as possible in the project phase (pre-FEED phase).

Benchmarks

Since the 1990s this best practice has been used for duplicate plants in the ammonia, ethylene, LNG, and methanol refineries. All past pump reliability issues were evaluated to produce a best practice pump type selection list for each application, along with other best practice guidelines to be included on the pump data sheet. Typical pump MTBFs using this best practice exceeded 80 months.

Best Practice 2.2

Pre-select centrifugal pumps (before the EP&C does) and modify the pump data sheet to include all details.

Select the pump for the noted performance conditions, and determine the pump selections using the acceptable vendor's pump data books or website information.

Select the pump to operate as closely as possible to its best efficiency point.

Notify the EP&C of the selections for each acceptable vendor for the project, noting specific component requirements and all best practice requirements on the data sheet.

Pre-selection of all centrifugal pumps assures that:

- All pump operating points will be in the Equipment Reliability Operating Envelope (EROE) resulting in optimum pump safety reliability and performance.
- Specific component best practice requirements will be met (impeller type, journal bearings, thrust bearings, seal and auxiliary systems).
- All company best practices will be incorporated into the pump design prior to sending the out the ITB (invitation to bid) to vendors.

Lessons Learned

Letting the EP&C select pumps can lead to lower pump reliability and most likely a narrower range of operation.

Many centrifugal pump field reliability issues can be traced back to improper selection of the pump. We have found that pumps which suffer damage from operation at high or low flows frequently were selected such that the specified rated point was either in the overload region (selecting too small a pump) or close to the minimum flow point (selecting too large a pump).

Benchmarks

Since the mid 1970s a standard practice has been to pre-select centrifugal pumps for the stated operating conditions, and note all best practice requirements on the pump data sheets. This practice has resulted in trouble free pump start-up reliability of the highest level (MTBFs greater than 80 months).

B.P. 2.2. Supporting Material

Start with a data sheet to completely define requirements

One of the single most important factors in selecting a pump to meet the requirements of a process system is to completely and accurately state all the requirements on a data sheet. A centrifugal pump data sheet, courtesy of the American Petroleum Industry (API 610) is supplied at the end of this supporting material.

Regardless of the source, all pump data sheets should contain the categories of information shown in Figure 2.2.1.

(P) (U)	■ Pump application and operating mode (single or parallel)
(P) (U)	■ Detailed operating conditions
(P) (U)	■ Accurate site and utility requirements
(M)	■ Pump performance
(P) (M)	■ Pump construction and experience
(P)	■ Spare parts required
(P) (M)	■ Driver details
(P)	■ QA inspection and test requirements

Note: to be completed by:
P = Purchaser
M = Manufacturer (vendor)
U = User

Fig 2.2.1 • Minimum data sheet requirements

Completely define the operating conditions

Correctly stated operating conditions are essential for proper definition, and for subsequent selection of a specific type and configuration of pump.

Once it is decided to install a pumping system, a sketch should be drawn to define all of the components that are required to be included into that system. Some of the factors which need to be considered in completing the sketch and system design are as follows:

Flow rate – All flow rates, including minimum, normal and rated should be listed in the data sheet. Normal flow is usually that required to achieve a specific process operation. The rated flow is normally a set percentage increase over the normal flow, and it usually includes consideration for pump wear, and the type of operation within the process system. It can be up to ten (10) percent, depending upon specific company practice. The minimum flow is important to identify in order to establish if a minimum flow bypass line is required for process or mechanical design considerations.

Head required – The required head that the pump must develop is based on the static pressure difference between the discharge terminal point and the suction source, the elevation difference and the friction losses through system components including suction and discharge side piping, pressure drop through heat exchangers, furnaces, control valves and other equipment. It is represented by the equation in Figure 2.2.2.

$$H = \frac{10.2 \times \Delta P \text{ (at pump flanges)}}{S.G.}$$

ΔP = Total pressure difference between the discharge system and suction system, measured at the pump flanges in barg
S.G. = Specific gravity of the liquid at pump temperature
H = Pump required head in meter-kg force/kg mass
Notes: If pressure is measured in kPa, constant = 0.102
If pressure is measured in kg/cm^2, constant = 10.003
If pressure is measured in PSIG, constant = 2.311 and head required is measured in ft-lb force/lb mass

Fig 2.2.2 • Required head equation

Liquid properties – Viscosity, vapor pressure and specific gravity each play an important role in achieving the required level of pump reliability within the operating system. **Viscosity** can impact pump performance to the extent that it may not be justified to even use a centrifugal pump when the viscosity values are greater than 7.5 centistokes (50 SSU). The hydraulic institute has published curves which can be used to calculate the performance effects resulting from pumping viscous liquid.

Vapor pressure and **specific gravity** influence the type of pump to use and its mechanical design configuration. **Vapor pressure** is an important property when determining whether there is adequate net energy available from the pump suction to avoid vaporization of the liquid – which can lead to performance deterioration and possible lower life expectancy of the pump.

Specific gravity is the liquid property used to calculate the amount of head a pump has to produce to overcome the resistance of the suction and discharge systems. It is also used as a guideline to determine whether a pump casing design should be of the vertical (radial) split or horizontal (axial) configuration (refer to Figure 2.2.3 for some guidelines).

Use radial split casing for:
- S.G. $\leq$ 0.7 @ pumping temperature
- Pumping temperature $\geq$ 200°C (400°F)
- Flammable or toxic liquids at rated discharge pressures above 6,896 Kpa (1000 psig)

Fig 2.2.3 • Casing configuration guidelines

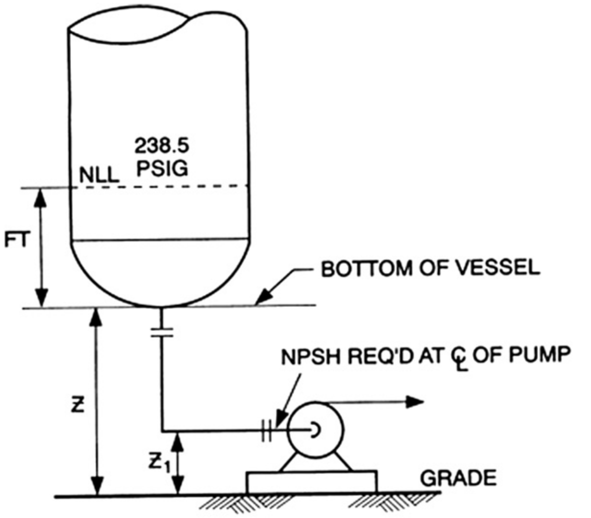
NPSH available – Net positive suction head available is a characteristic of the process suction system. It is the energy above the vapor pressure of the liquid, measured at the suction flange of the pump, which is required to maintain the fluid in a liquid state. In a centrifugal pump it is usually measured in feet of liquid (refer to Figure 2.2.4 for a typical method for calculating NPSH available). It is important to note that the pressure at the suction source cannot be considered equal to the NPSH. In Figure 2.2.4, it can be seen that the source pressure is the same as the vapor pressure, indicating that the liquid is at its boiling point. When the vapor pressure is subtracted from the suction pressure, the resulting NPSH available is 2.1 psi or ten (10) feet. When calculating NPSH available, it is prudent to incorporate a margin of safety to protect the pump from potential cavitation damage resulting from unexpected upsets. The actual margin will vary from company to company. Some use the normal liquid level as the datum point, while others use the vessel tangent or the bottom of the vessel. Typical suggested margins are two (2) feet for hydrocarbon liquids (including low S.G.), and ten (10) feet for boiling water.

Defining the pump rated point for efficient operation

Since centrifugal pumps are not normally custom designed items of equipment, it is important to ensure that each vendor will quote similar pump configurations for the specific operating conditions set forth on each application data sheet. When establishing which pump characteristic and impeller pattern to select for a specific application, certain guidelines should be followed (refer to Figure 2.2.5).

Carefully define critical component requirements

The reliability of a pump can be improved through proper specification, selection and operation of components such as bearings, mechanical seals and drivers. Industry standards such as API Standard 610 for centrifugal pumps, and Standard 682 for mechanical seals, contain minimum requirements which, if implemented, should result in improved reliability and extended on-stream operating time. Some salient points about each of these components are highlighted in Figures 2.2.8–2.2.10.



$NPSH_a$ = AVAILABLE NET POSITIVE SUCTION HEAD
P = PRESSURE ABOVE LIQUID INTERFACE, PSIA
P_a = ATMOSPHERIC PRESSURE, PSIA
P_V = VAPOR PRESSURE OF LIQUID, PSIA
h_f = PRESSURE DROP IN SUCTION PIPING DUE TO FRICTION
Z = ELEVATION FROM BOTTOM OF VESSEL TO GRADE
Z_1 = DISTANCE FROM GRADE TO ℄ OF PUMP
S.G. = SPECIFIC GRAVITY
NLL = NORMAL LIQUID LEVEL
S.M. = MARGIN OF SAFETY

$$NPSH_a = \left[\frac{(P + P_a - P_V)2.31}{S.G.}\right] + \left[Z - Z_1 - h_f\right] - S.M.$$

P_V = 253.2 PSIA, P = 238.5 PSIG, P_a = 14.7 PSIA, S.G. = 0.488, Z = 15 FT, Z_1 = 3 FT, h_f = 2.4 FT, SM = 0

$$NPSH_a = \left[\frac{(238.5 + 14.7 - 253.2)2.31}{0.488}\right] + \left[15 - 3 - 2.4\right] - 0 = 9.69$$

SAY 10.

NOTE:
WITH 8FT NLL,
ZERO SAFETY
MARGIN IS ASSUMED

Fig 2.2.4 • Calculate available NPSH

- When selecting a specific impeller pattern, the rated flow should be no greater than ten (10) percent to the right of best efficiency point. This will result in operation at close to best efficiency point during normal operation (refer to Figure 2.2.6). Also, selecting a pump to operate too far to the right of best efficiency point can result in it operating in the 'break'. This is when it is at maximum pumping capacity, and the total head is reduced while the suction head is held (the impeller actually acts as an orifice to limit the flow).
- Selecting a pump for the rated flow too far to the left of best efficiency point (oversized pump) can result in cavitation damage caused by internal recirculation (refer to Figure 2.2.7).

Fig 2.2.5 • Application guidelines

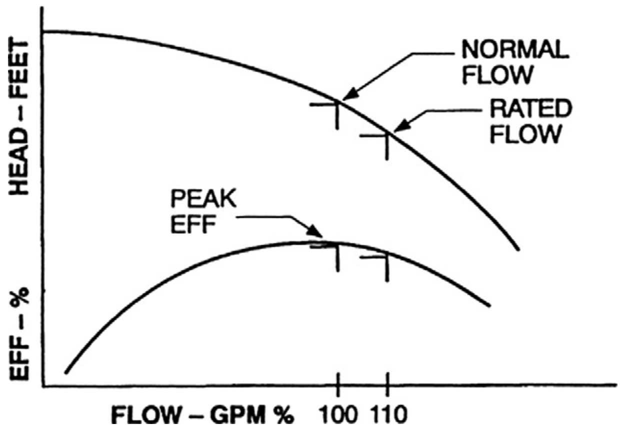


Fig 2.2.6 • Selecting a specific impeller pattern

Guidelines to use when selecting pump style

The choice for selecting the type of pump to use for a given application can vary with specific gravity, operating temperature, pressure conditions, liquid composition and available NPSH. Figures 2.2.11 to 2.2.17 provide guidelines, which can help to simplify the choice.

Using the guidelines presented, let us now focus on three examples of how to select a centrifugal pump for a given process system application (refer to Figure 2.2.18).

Before the appropriate pump and driver can be selected, it is necessary to completely define the process system operating conditions under which the pump will operate. This will include the suction and discharge system resistance, the head (energy) required by the system and the NPSH available (refer to Table 2.2.1).

When the process system is defined, the next step is to complete the tasks presented in Figure 2.2.19.

Based on an assessment of the process system requirements in Table 2.2.1 and the guidelines for selecting pump and driver

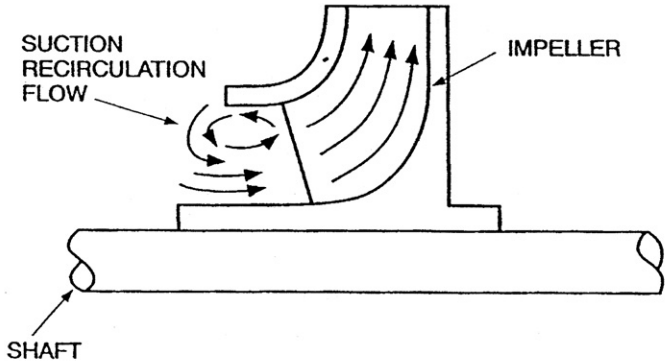


Fig 2.2.7 • Suction recirculation flow pattern

- Centrifugal pumps require bearings to carry radial/axial loads
- Bearing alternatives: Anti-friction, hydrodynamic ring oil lubricated or hydrodynamic pressure lubricated
- Oil lubricated (anti-friction) bearings are used in majority of process pumps to carry loads
- Pressure lubricated hydrodynamic bearings are normally used for high pressure, high horsepower, high speed applications
- Criteria in API Standard 610 for pressure lubrication: when product of pump rated horsepower and rated speed in rev/min is greater than 2.0 million
- Pressure lubrication systems can either be integral or separate, but should include, as a minimum, an oil pump, reservoir, filter, cooler, controls and instrumentation
- Ring oil lubrication may be applied to hydrodynamic journal bearings in less severe service (when dN factor is less than 300,000. A dN factor is the product of bearing, size (bore) in millimeters and the rated speed in revolutions per minute).

Fig 2.2.8 • Bearing application guidelines

- Mechanical seals are often used in pumps handling hazardous as well as non-hazardous liquids that must be contained within the unit.
- The single seal arrangement is most widely used in process industry.
- Single seal design consists of a rotary face in contact with a stationary face.
- For most services, a carbon face mating against tungsten carbide is satisfactory.
- Seals offer the advantages of long life, low maintenance and high reliability.
- In general, seals handling light specific gravity liquids at low temperature and high vapor pressure give most problems in the field.
- Materials for cold service seals must be suitable for temperatures of start-up, cool down and running; the atmospheric side must be held above 0°C (32°F) to prevent ice formation; and there must be enough liquid at the seal surfaces.
- Successful operation of any seal depends largely on correctly specifying liquid conditions of vapor pressure, temperature specific gravity, etc.
- API Standard 682 is an excellent resource for overall mechanical seal application guidelines.
- The pressure in the seal chamber (stuffing box) must be at least 25 psig above the pump suction pressure.

Fig 2.2.9 • Mechanical seal application guidelines

- Pump drivers are normally electric motor or steam turbine.
- Choice of driver type is usually based on plant utility balance plus reliability evaluation of each type to perform within the operating system.
- Motors can be sized by several methods:
- Name plate rating large enough to cover the complete range of pump performance curve.
- Size of motor based on system curve analysis to establish maximum horsepower required at intersection of system curve and pump performance curve.
- API Standard 610 has guidelines for sizing motor drives. A margin of 125% is recommended for motors equal to or less than 18.6 kW (25 hp), 115% for 22.4 kW–56 kW (30 to 75 hp) and 110% for motors rated 75 kW (100 hp) or more.
- Steam turbine drives are normally sized for the power required at pump rated condition. This is possible because turbines can accommodate increased power loads more readily than electric motors.

Fig 2.2.10 • Driver sizing guidelines

- Most commonly applied centrifugal pump – most applications
- Total head limited to 380 mm (15 inch) impeller diameter @3600 rpm (approximately 183 meters [600 feet] head). Larger diameter impellers operate at lower speeds
- Low, medium, high temperature (with cooled bearings, stuffing box)
- Relatively low NPSH required for single suction impeller
- All process services with proper materials selection
- Center of gravity of impeller is outside bearing span
- Axial thrust

Fig 2.2.11 • Single stage, single suction overhung impeller characteristics

- Gaining acceptance as alternative to single stage overhung pump
- Total head limited to approximately 122 meters (400 ft)
- Low temperature applications only
- Relatively low NPSH required
- Limited to approximately 150 kW (200 H.P.)
- Most designs utilized do not incorporate bearings (they use the motor bearings to position the pump shaft)
- Note: Designs are now available that incorporate an anti-friction bearing in the pump housing

Fig 2.2.12 • Single stage in-line pump characteristics

- Use for high pressure applications greater than 13.730 kpa (2000 psi)
- Unlimited head capability
- Maximum flow approximately 114 M^3/hr (500 gpm) (5 plungers)
- Limited to approximately 1119 kW (1500 bhp)
- Constant volume capacity
- Produces pulsations and requires pulsation dampners

Fig 2.2.13 • Plunger-PD pump characteristics

- Used for high head low flow applications
- Total maximum head approximately 762 meters (2500 ft)
- Can be used for all temperatures
- Lowest $NPSH_{required}$ (can use inducer)
- Limited to 300 kW (400 bhp) maximum
- All process services with proper material selection
- Incorporates gear box to increase pump speeds as high as 30,000 rpm

Fig 2.2.14 • Single stage integral gear centrifugal pump characteristics

- Used for high flow applications (greater than 455 M^3/hr [2000 gpm])
- Low, medium head requirements
- Low NPSH required
- Low, medium high temperature (with cooled bearings, stuffing box)
- All process services with proper materials selection
- Center of gravity of impeller is between bearing span
- No speed constraint
- Low axial thrust
- Confirm that suction specific speed is less than 9,000. (Definition and equation is in Chapter 8)

Fig 2.2.15 • Single stage, double suction impeller between bearing pump characteristics

- Used for high head medium flow applications
- Double suction impeller for first stage for low $NPSH_R$
- Low, medium, high temperature (with cooled bearings, stuffing box)
- No speed constraint
- Thrust requires compensation (back to back impellers, balance device)

Fig 2.2.16 • Horizontal multistage between bearing pump characteristics

- Used for low NPSH available applications
- High head capability by adding stages
- Low, medium, high temperature
- Low, medium flow range
- No speed constraint
- Most non abrasive process liquids with proper materials selection

Fig 2.2.17 • Vertical multistage pump characteristics

we can determine that the pump defined in Figure 2.2.19 satisfies all of the guidelines.

For example no. 2 let us select a pump for a boiler feed water application with operating conditions shown in Figure 2.2.20.

This application requires a multi-stage, axial, split case pump based on the criteria that the head (energy) required by the

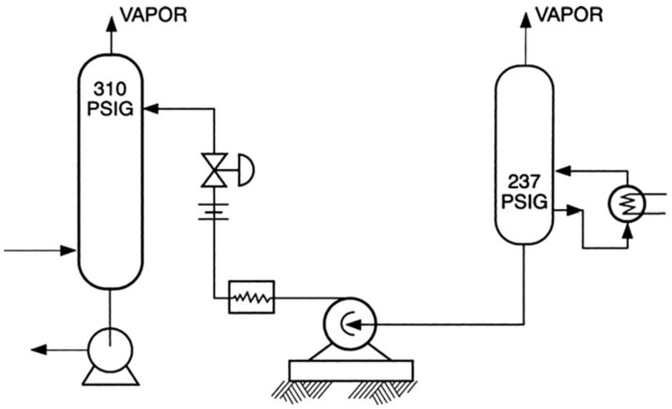


Fig 2.2.18 • Example No. 1: process system

Table 2.2.1 Calculate process system variables

Elevations above grade:		Discharge pressure calculation:	
Pump centerline	3ft	Vessel pressure	310 psig
Inlet nozzle to vessel	72ft	Static elevation head	
Liquid surface in suction vessel		(72-3) × 0.433 × 0.488	= 14.5 psi
	max 32ft	Friction ΔP:	
	min 22 ft	Piping	10 psi
Pressure p, in bottom of suction vessel	237 psig	Orifice Control valve	2 psi 30 psi
Pressure drops:		Exchanger	15 psi
Suction piping	1 psi	Pd ($Vessel_{press}$ + $Elev_{press}$ + losses)	= 381.5 psi
Discharge piping	10 psi	ΔP	Pd−Ps
Flow orifice	2 psi		381.5−240
Control valve	30 psi		141.5 psi
Exchanger	15 psi		
Flow rate:	500 gpm		
Specific gravity	0.488 @ p.t.	$\frac{\Delta P \times 2.31}{s.g.} = \frac{141.5 \times 2.31}{0.488} =$	670 ft
Vapor pressure	251 psia		
Suction pressure calculation:		$NPSH_{AVAILABLE}$ (for boiling liquid)	
Vessel pressure	237 psig	p_a surface + 14.7 − P_v +	
Static elevation head		static elev diff − friction	
(22-3) × 0.433 × 0.488	= 4 psi	237 + 14.7 − 251 + 4 − 1 = 3.7 psi	
Suction line Δp	−1 psi	$NPSH_{AVAILABLE} = \frac{3.7 \times 2.31}{0.486} = 17.5$ ft	
P_s ($Vessel_{press}$ + $Elev_{press}$ − loss)	= 240 psi		

Select pump type based on guidelines:

- Single stage overhung impeller
- Multi stage axial or radial split casing design
- Match NPSHR vs. NPSHA
- Calculate bhp based on pump efficiency
- Determine driver hp rating based on API criteria

Fig 2.2.19 • Tasks for selecting pump and driver

system exceeds the head (energy) which can be provided by a 15 inch single stage impeller.

The pump selected is a Union Pump 3 × 4 MOC, 5 stage axial split casing unit (refer to Figures 7.22 and 7.23 for the performance curve and cross section drawing, respectively). Note that selecting a 9.50″ diameter will result in the pump operating at its best efficiency point (BEP) at rated flow.

For our third example, we shall examine the selection of a pump type with a constraint on NPSH available. A hot well condensate pump installed in a steam turbine condenser system will be used to illustrate this example (refer to Figure 2.2.22 for operating conditions).

It is apparent that the NPSH available is a major constraint for selecting a conventional horizontal pump for pumping condensate from the condenser hot well. For this application, a vertical canned pump is the appropriate selection (refer to Figure 2.2.25).

The feature of this design that makes it suitable for use in this type of service is the fact that the first stage impeller is located at the bottom end of the shaft, and the shaft length can be made

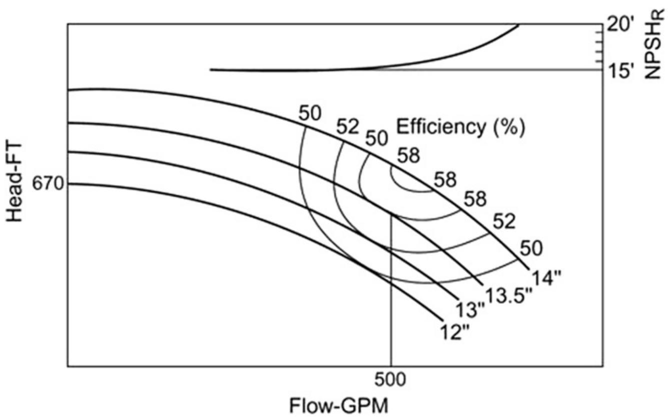
Liquid	**Boiler feed water**
S.G.	0.93
P.T.	220°F
P_s	25 psig
P_d	650 psig
$NPSA_{available}$	26 ft
Flow rate rated	275 gpm
$head_{required}$	1553 ft.

Fig 2.2.21 • Example No. 2: operating conditions

Flow rate	100 gpm
■ S.G.	0.98
■ P.T.	120°F
■ NPSHA	0 ft. at grade
■ P_s	1.96 psig
■ P_d	120 psig
■ Head	3.13

Fig 2.2.22 • Example No. 3: operating conditions

sufficiently long to satisfy the NPSH required by the pump. It is common practice to reference available NPSH to grade elevation for this type of pump design. This allows for variations in the design of concrete foundation height, and location of the suction nozzle centerline from top of foundation (refer to Figure 2.2.26).



PUMP TYPE 3 x 4 x 14A – HHS
SINGLE STAGE OVERHUNG DESIGN

EFFICIENCY = 56%

$$\text{BHP} = \frac{500 \times 670 \times .488}{3960 \times .56} = 73.7$$

NPSHR = 16 FT

NPSHA = 17.5 FT

SPEED = 3550 RPM

IMPELLER DIA = 13 1/2 INCH

Fig 2.2.20 • Example no. 2 pump selection (Courtesy of Union Pump Co.)

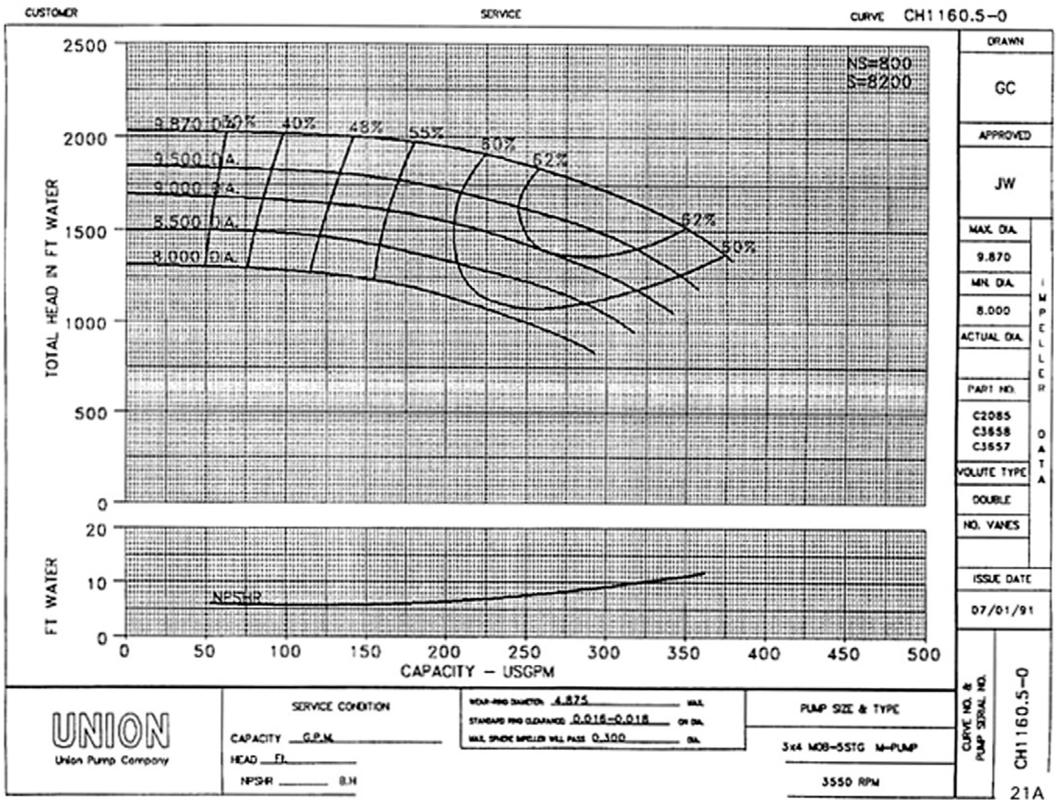


Fig 2.2.23 • A typical centrifugal pump performance curve (Courtesy of Union Pump Co.)

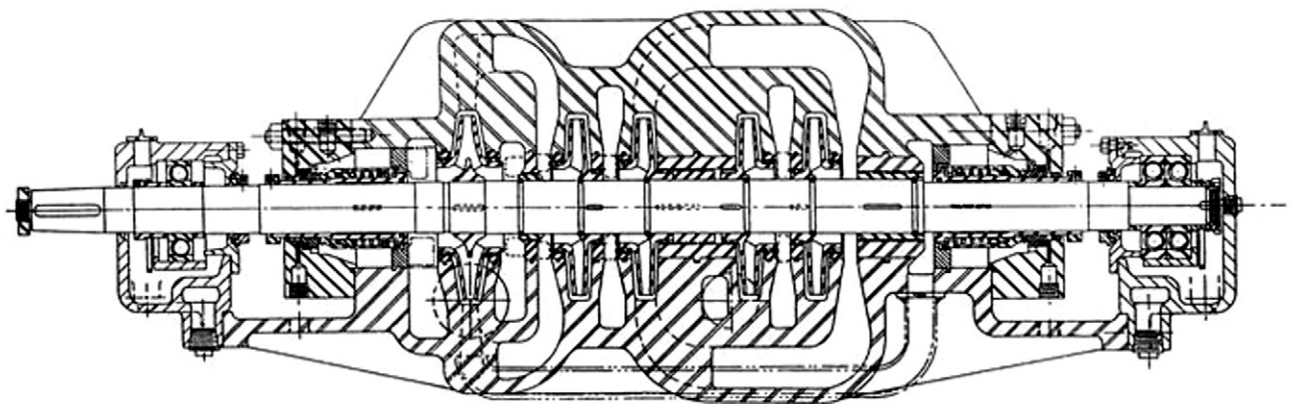


Fig 2.2.24 • Multistage centrifugal pump (Courtesy of Union Pump Co.)

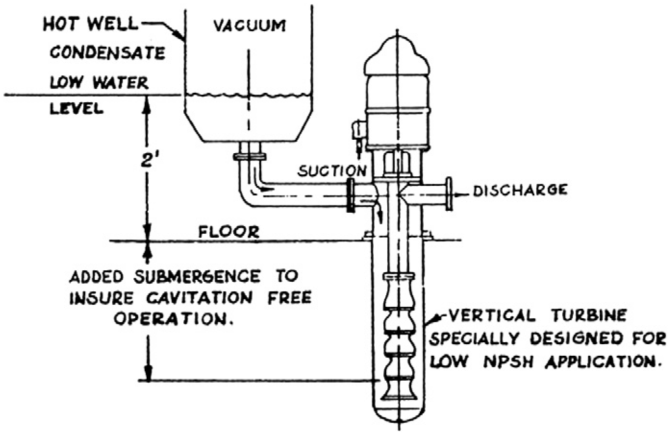


Fig 2.2.25 • Application of vertical pump in condensate hot well service

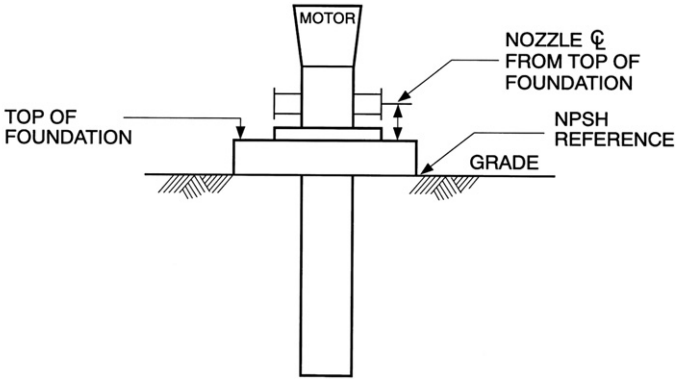


Fig 2.2.26 • Example No. 3 NPSH reference

Best Practice 2.3

Accurately define all liquid and hydraulic conditions on the Centrifugal Pump Data Sheet.

Completely define all possible operating conditions, by meeting with process engineering and site senior operations personnel during the process design phase to accurately determine:

- Flow rates
- Head required
- Liquid properties
- NPSH available

Define the rated operating point to be as close to the pump best efficiency point as possible; but not greater than 10% or less than 50% from the potential pump best efficiency point for centrifugal pump without an inducer.

Lessons Learned

Small changes in operating conditions, especially resulting in higher than specified head required in the field, will cause lower pump reliability and expose the plant to safety issues in hydrocarbon services.

The pump operating conditions noted on the pump data sheet are the result of hydraulic calculations performed prior to pump selection.

The pump head required by the process is a function of inlet pressure, discharge pressure and specific gravity.

Centrifugal pump curve head rise (head at zero flow divided by head at rated conditions) can be 5% or lower.

Any significant change in the total head required during field operation can result in a centrifugal pump operating in a flow range that will lower pump reliability and expose the pump to component failures (bearings, seals, impellers, wear ring seizure and shaft breakage).

Benchmarks

Since the mid-1970s I have made a practice of reviewing all pump operating conditions with a senior process engineer and plant operator, during the process design phase, to ensure that the conditions are specified as accurately as possible prior to pump selection. This practice has resulted in trouble free pump start-up and continued operation at optimum levels of safety and reliability (Pump MTBFs exceeding 80 months).

B.P. 2.3. Supporting Material

Completely define the operating conditions

Correctly stated operating conditions are essential for proper definition and subsequent selection of a specific type and configuration of pump that will meet these specified conditions.

Once it is decided to install a pumping system, a sketch should be drawn to define all of the components which are needed. Some of the factors which need to be considered in completing the sketch and system design include the following:

Flow rate – All flow rates including minimum, normal and rated should be listed in the data sheet. Normal flow is usually the flow required to achieve a specific process operation. The rated flow is normally a set percentage increase over the normal flow and it usually includes consideration for pump wear and the type of operation within the process system. It can amount to as much as ten (10) percent depending upon specific company practice. The minimum flow is important to identify in order to establish if a minimum flow bypass line is required for process or mechanical design considerations.

Head required – The required head that the pump must develop is based on the static pressure difference between the discharge terminal point and the suction source, the elevation difference and the friction losses through system components including suction and discharge side piping, pressure drop through heat exchangers, furnaces, control valves and other equipment. It is represented by the equation in Figure 2.3.1.

$$H = \frac{10.2 \times \Delta P \text{ (at pump flanges)}}{S.G.}$$

ΔP = Total pressure difference between the discharge system and suction system, measured at the pump flanges in barg
S.G. = Specific gravity of the liquid at pump temperature
H = Pump required head in meter-Kg force/Kg mass
Notes: If pressure is measured in KPa, constant = 0.102
If pressure is measured in Kg/cm^2, constant = 10.003
If pressure is measured in Psig, constant = 2.311 and head required is measured in FT-LB force/LB mass

Fig 2.3.1 • Required head equation

Liquid properties – Viscosity, vapor pressure and specific gravity each play an important role in achieving the required level of pump reliability within the operating system. **Viscosity** can impact pump performance to the extent that it may not be justified to even use a centrifugal pump when the viscosity values are greater than 3000–5000 SSU. The hydraulic institute has published curves which can be used to calculate the performance effects resulting from pumping viscous liquid.

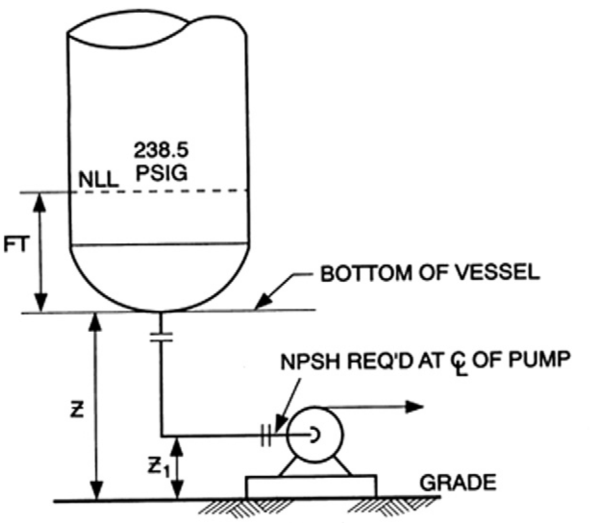
Vapor pressure and **specific gravity** influence the type of pump to use and its mechanical design configuration. **Vapor pressure** is an important property when determining whether there is adequate net energy available at the pump suction to avoid vaporization of the liquid which can lead to performance deterioration and possible shortened life expectancy of the pump.

Specific gravity is the liquid property used to calculate the amount of head a pump has to produce to overcome the resistance of the suction and discharge systems. It is also used as a guideline to determine whether a pump casing design should be of the vertical (radial) split or horizontal (axial) configuration (refer to Figure 2.3.2 for some guidelines).

Use radial split casing for:
- S.G. $\leq$ 0.7 @ pumping temperature
- Pumping temperature $\geq$ 200°C (400°F)
- Flammable or toxic liquids at rated discharge pressures above 6,896 Kpa (1000 psig)

Fig 2.3.2 • Casing configuration guidelines

NPSH available – Net positive suction head available is a characteristic of the process suction system. It is the energy above the vapor pressure of the liquid, measured at the suction



$NPSH_a$ = AVAILABLE NET POSITIVE SUCTION HEAD
P = PRESSURE ABOVE LIQUID INTERFACE, PSIA
P_a = ATMOSPHERIC PRESSURE, PSIA
P_V = VAPOR PRESSURE OF LIQUID, PSIA
h_f = PRESSURE DROP IN SUCTION PIPING DUE TO FRICTION
Z = ELEVATION FROM BOTTOM OF VESSEL TO GRADE
Z_1 = DISTANCE FROM GRADE TO ℄ OF PUMP
S.G. = SPECIFIC GRAVITY
NLL = NORMAL LIQUID LEVEL
S.M. = MARGIN OF SAFETY

$$NPSH_a = \left[\frac{(P + P_a - P_V)2.31}{S.G.}\right] + \left[Z - Z_1 - h_f\right] - S.M.$$

P_V = 253.2 PSIA P_a = 14.7 PSIA Z = 15 FT h_f = 2.4 FT
P = 238.5 PSIG S.G. = 0.488 Z_1 = 3 FT SM = 0

$$NPSH_a = \left[\frac{(238.5 + 14.7 - 253.2)2.31}{0.488}\right] + \left[15 - 3 - 2.4\right] - 0 = 9.69$$

SAY 10.

NOTE:
WITH 8FT NLL,
ZERO SAFETY
MARGIN IS ASSUMED

Fig 2.3.3 • Calculate available NPSH

flange of the pump, which is required to maintain the fluid in a liquid state. In a centrifugal pump it is usually measured in feet of liquid (refer to Figure 2.3.3 for a typical method for calculating NPSH available). It is important to note that the pressure at the suction source cannot be considered equal to the NPSH. In Figure 2.3.3 it can be seen that the source pressure is the same as the vapor pressure, indicating that the liquid is at its boiling point.

When the vapor pressure is subtracted from the suction pressure, the resulting NPSH available is 2.1 psi or ten (10) feet.

- When selecting a specific impeller pattern the ratedflow should be no greater than ten (10) percent to the right of best efficiency point. This will result in operation at close to best efficiency point during normal operation. Also, selecting a pump to operate too far to the right of best efficiency point can result in the pump operating in the ‘break’. A pump is considered operating in the ‘break’ when it is pumping maximum capacity and the total head is reduced while the suction head is held (the impeller actually acts as an orifice to limit the flow).
- Selecting a pump for the rated flow too far to the left of best efficiency point (oversized pump) can result in cavitation damage caused by internal recirculation.

Fig 2.3.4 • Application guidelines

When calculating NPSH available it is prudent to incorporate a margin of safety to protect the pump from potential cavitation damage resulting from unexpected upsets. The actual margin will vary from company to company. Some will use the normal liquid level as the datum point, while others use the vessel tangent or the bottom of the vessel. Typical suggested margins are: two (2) feet for hydrocarbon liquids (including low S.G.), and ten (10) feet for boiling water.

Defining the pump rated point for efficient operation

Since centrifugal pumps are not normally custom designed items of equipment, it is important to ensure that each vendor will quote similar pump configurations for the specific operating conditions set forth on each application data sheet. When establishing which pump characteristic and impeller pattern to select for a specific application, certain guidelines should be followed (refer to Figure 2.3.4).

An accurate definition of all liquid and hydraulic conditions along with a proper centrifugal pump selection will ensure operation within the “Heart of the Curve”, also known as the Equipment Reliability Operating Envelope (EROE). Operating a centrifugal pump in this region will optimize reliability by preventing hydraulic disturbances (see Figure 2.3.5).

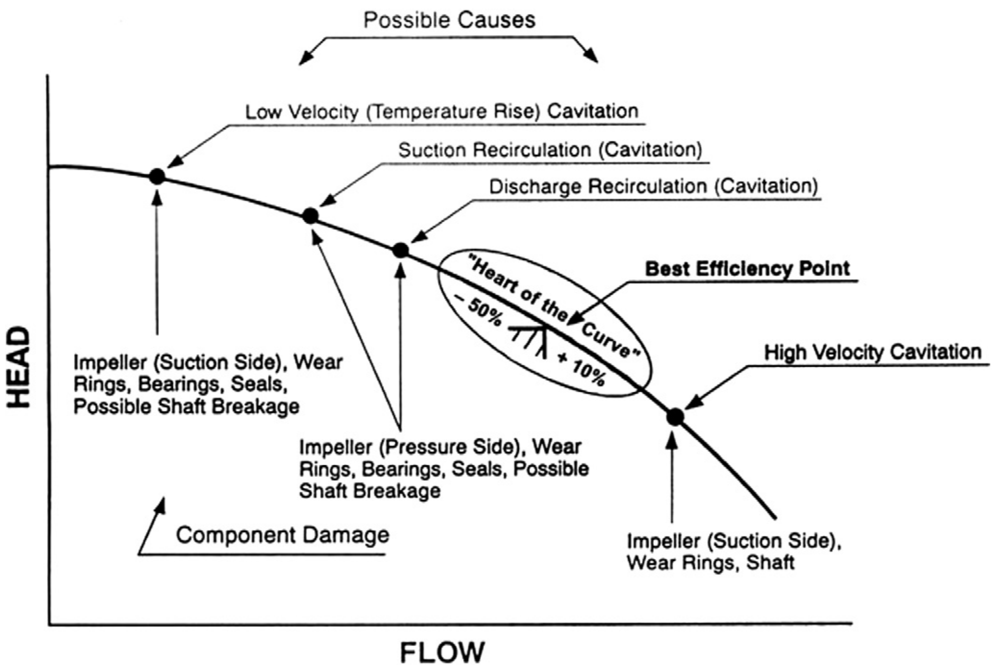


Fig 2.3.5 • Centrifugal pump component damage and causes as a function of operating point

Best Practice 2.4

State the required centrifugal pump casing design – horizontal split or radial and single or double volute – on the pump data sheet.

Review pump vendor experience lists to determine field operating maximum pressures for horizontal split pumps and confirm acceptable field operation with users above 10,342 kpa (1500 psi) operating pressures. Select a radial split pump case if there are reports of horizontal split pump joint wear and leakage.

Single volute pumps will produce large radial forces on the rotor at flow rates other than at the centrifugal pump best efficiency point which will increase pump vibration and reduce wear ring and throat bushing life.

Double volute pumps will minimize the rotor radial forces and should be used whenever available.

Confirm pump vendor availability of double volute design for all centrifugal pumps prior to vendor selection.

Lessons Learned

Leaving pump case design to the EP&C can lead to field safety and reliability issues.

Not being proactive during preparation of the pump data sheets has allowed EP&Cs (contractors) and pump vendors to propose pumps that are a lower cost but expose the end user to field safety and reliability issues.

Benchmarks

These guidelines have been followed since the mid-1970s in all projects to ensure optimum pump field reliability. This best practice has continually produced pumps with trouble free start-up and MTBFs in excess of 80 months.

B.P. 2.4. Supporting Material

Dynamic pumps

Centrifugal pumps can be referred to as 'dynamic' machines. That is to say they use centrifugal force for pumping liquids from a lower to a higher level of pressure. Liquid enters the center of the rotating impeller, which imparts energy to the liquid. Centrifugal force then discharges it through a volute, as shown in Figure 2.4.1.

The centrifugal pump is one of the most widely used fluid handling devices in the refining and petrochemical industry. Every plant has a multitude of these types of pumps in operation. A brief description of the various designs found in operating plants follows. Refer to Table 2.4.1 for the application limits of dynamic pumps.

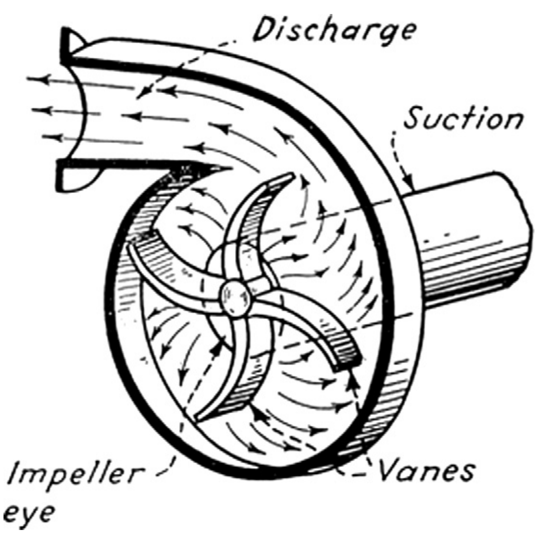


Fig 2.4.1 • Dynamic pump principle

Table 2.4.1 Application limits – Dynamic pumps

Pump type	Pressure KPAA	Head Meters	Flowrate-M^3/hr	Horsepower –KW
Single stage overhung	2,69 (300)	244 (800)	1,591 (7,000)	1,492 (2,000)
Single stage double	2,069 (300)	244 (800)	15,909 (>70,000)	5,968 (>8,000)
Flow between bearing				
Single stage inline	2,069 (300)	244 (800)	1,591 (7,000)	149 (200)
Integral gear centrifugal	6,895 (1,000)	762 (2,500)	455 (2,000)	298 (400)
Multistage horizontal split	13,790 (2,000)	1,892 (6,000)	682 (3,000)	373 (500)
Multistage barrel	20,685 (3,000)	2,439 (8,000)	455 (2,000)	3,730 (>5,000)
Vertical canned pump	10,342.5 (1,500)	1,2189.5 (4,000)	15,909 (>70,000)	746 (1,000)
Sump pumps	689.5 (100)	91.5 (300)	1,591 (7,000)	186.5 (250)
Submersible	689.5 (100)	91.5 (300)	909 (4,000)	112 (150)
Magnetic drive pump	2,069 (300)	244 (800)	682 (3,000)	597 (800)

Note: Customary Values in Parenthesis (PSIA), (FT), (GPM), (HP)

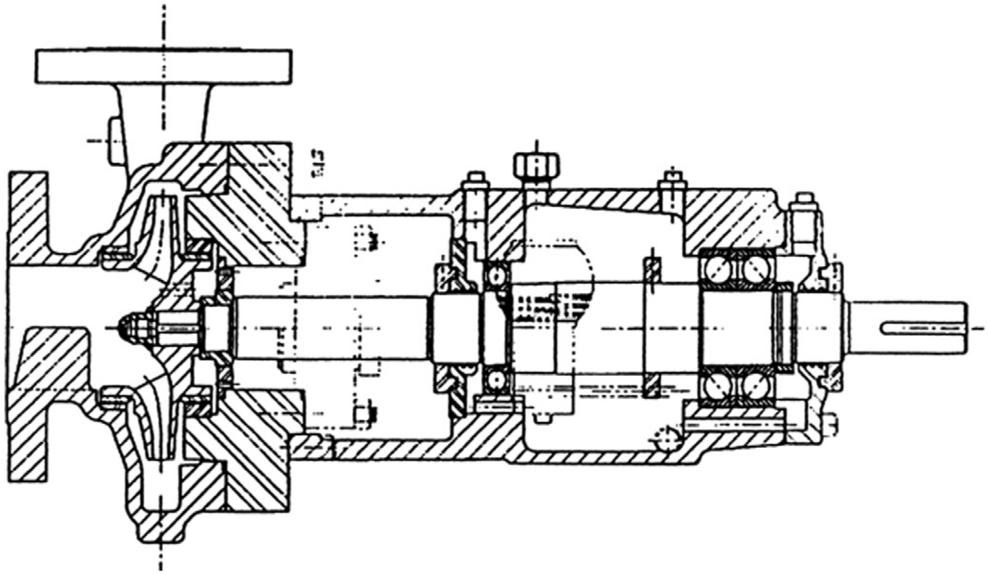


Fig 2.4.2 • Single stage overhung pump (Courtesy of Union Pump Co.)

Single stage overhung pump

The single stage, overhung pump design shown in Figure 2.4.2 is probably the most widely used in industry. Its construction incorporates an impeller fixed to a shaft, which has its center of gravity located outside the bearing support system.

Single stage inline

This type of pump is finding increased usage in applications of low head, flow and horsepower. Refer to Figure 2.4.3.

The advantage of this pump design is that it can be mounted vertically (inline) between pipe flanges and does not require a baseplate. A concrete, grouted, support plate is strongly recommended, however. It should be noted that many inline designs do not incorporate bearings in the pump, and rely on a rigid coupling to maintain pump and motor shaft alignment. Acceptable pump shaft assembled runout with these types of pumps should be limited to 0.001".

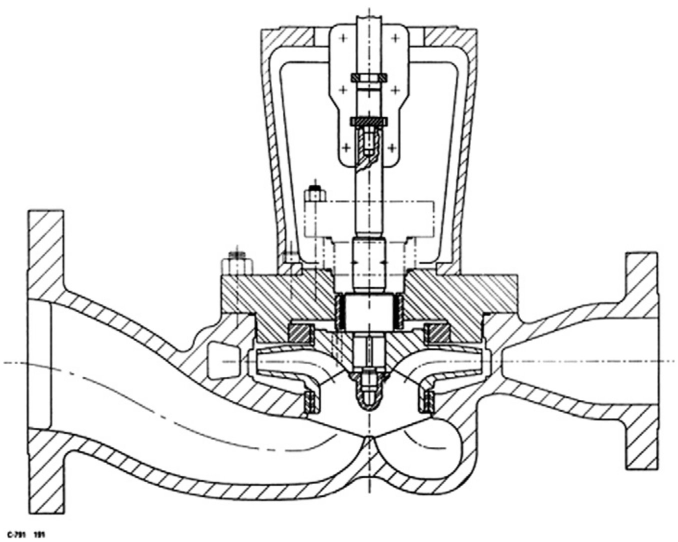


Fig 2.4.3 • Single stage inline (Courtesy of Union Pump Co.)

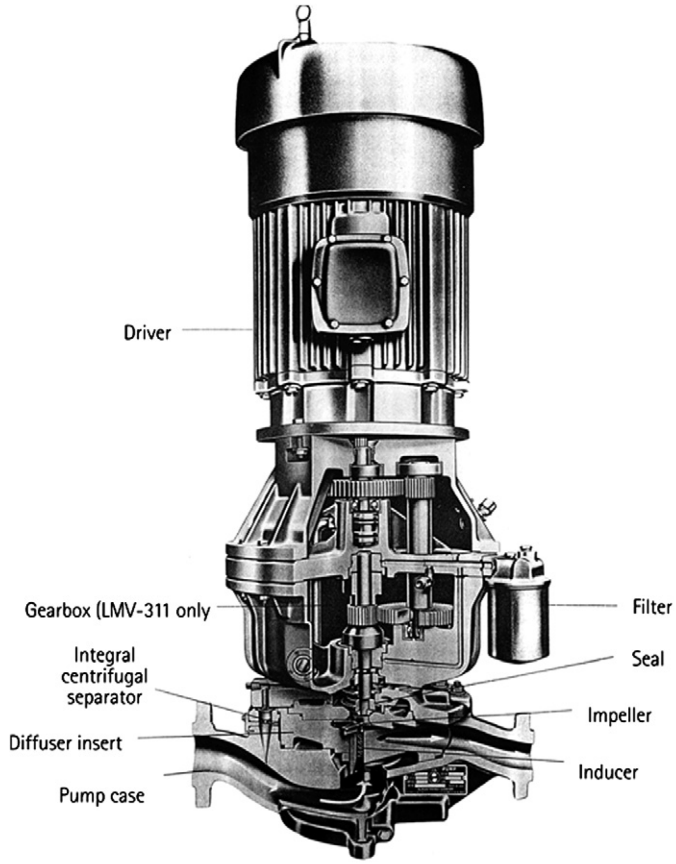


Fig 2.4.4 • Integral gear centrifugal pump (Courtesy of Sunstrand Corp.)

Integral gear centrifugal

This type of pump is used for low flow applications requiring high head. Refer to Figure 2.4.4. The pump case design is similar to the inline, but incorporates pump bearings and an integral gear to increase impeller speeds over 30,000 RPM.

Single stage double flow between bearing

As the name implies, double suction impellers are mounted on between-bearing rotors, as shown in Figure 2.4.5.

This pump design is commonly used when flow and head requirements make it necessary to yield low values of NPSH required. When designing piping systems for this type of pump, care must be taken to ensure equal flow distribution to each end of the impeller to prevent cavitation and vibration.

Multistage horizontal split

When the hydraulic limits of a single stage pump are exceeded, it is common practice to use a multistage pump, shown in Figure 2.4.6.

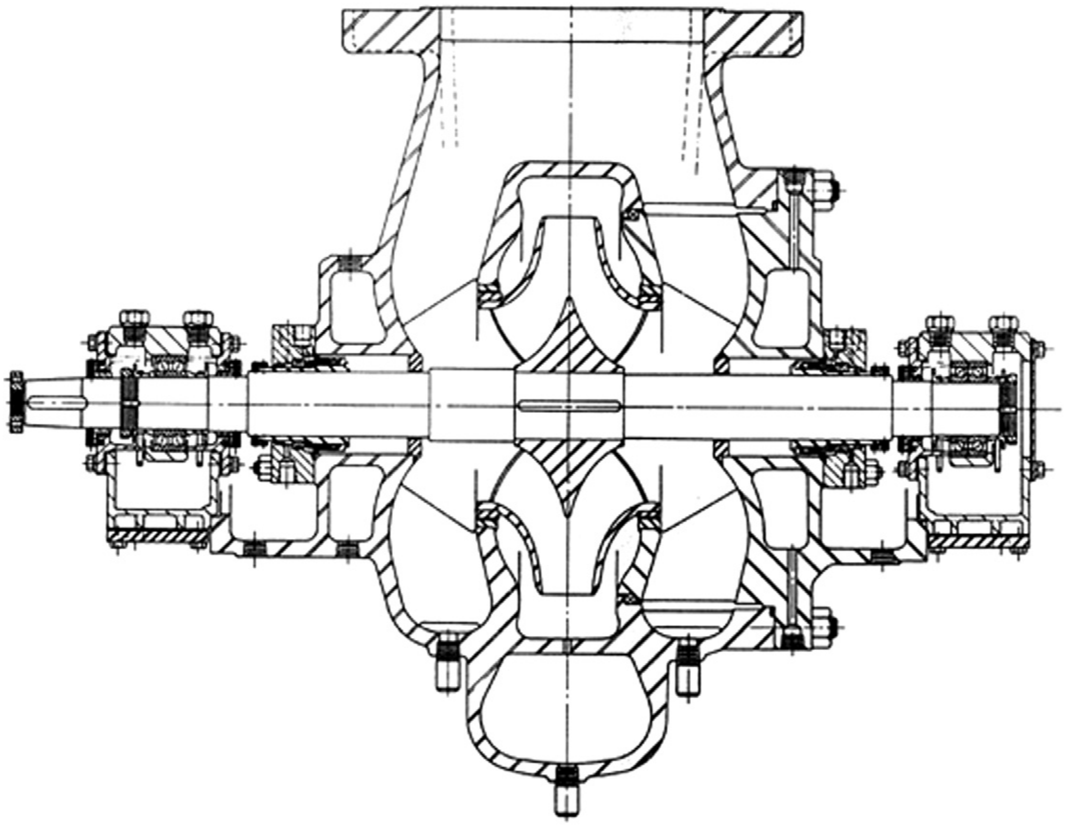


Fig 2.4.5 • Single stage double flow between bearing (Courtesy of Union Pump Co.)

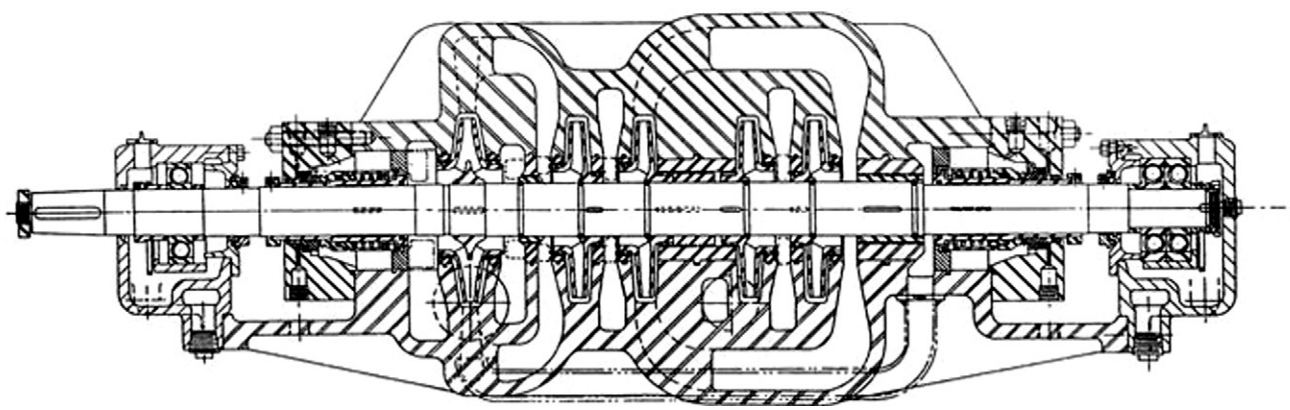


Fig 2.4.6 • Multistage centrifugal pump (Courtesy of Union Pump Co.)

This figure illustrates a horizontal split casing design, which allows the rotor to be removed vertically after the top half casing is unbolted. This type of pump is normally limited to working pressure of approximately 13,790 kpa (2000 psi), temperatures of up to 315°C (600°F) and S.G. of 0.7 or greater. Impeller configuration for this type of pump can be either 'inline' or 'opposed'. The 'opposed' impeller arrangement has the advantage of not requiring a thrust balancing device, which is required for the 'inline' configuration.

Multistage barrel

The so-called barrel casing design is shown in Figure 2.4.7.

It is used for service conditions exceeding those normally considered acceptable for a horizontal split case design. A thrust balance device is required, since the impeller configuration is almost always 'inline'. The circular, mounted, end flange results in excellent repeatability for a tight joint as compared to a horizontal split case design.

Volutes

The volute is that portion of the pump casing where the liquid is collected and discharged by centrifugal force when it leaves the impeller (refer to Figure 2.4.8).

As the liquid leaves the rotating impeller, the volute continually accumulates more liquid as it progresses around the casing. Because we want to keep the liquid velocity reasonably constant as the volume increases, the volute area between the impeller tip and the casing wall must be steadily increased since:

$$Q = AV$$

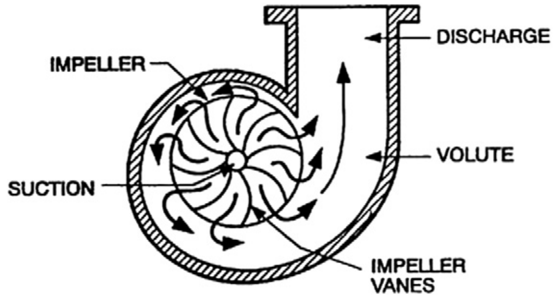


Fig 2.4.8 • Single volute casing form

Where Q = Flow (gpm)
A = Area
V = Velocity

Figure 2.4.8 shows this relationship. The form of volute casing shown in the figure results in uneven pressure distribution around the periphery of the impeller. In a single stage, overhung, impeller type pump with cantilever shaft, designed for high heads, the unbalanced pressure may result in increased shaft deflection and bearing loadings at off-design conditions.

The double volute casing design shown in Figure 2.4.9 equalizes the pressure around the impeller and reduces the unbalanced loading on the bearings. This is accomplished with two (2) similar flow channels which have outlets 180 degrees apart.

Figure 2.4.10 shows the relationship between the radial reaction force acting on an impeller vs. capacity for single and double volute pumps. The radial reaction force is directly proportional to the following parameters:

Impeller produced head
Impeller diameter
Impeller discharge width (b_2)
Specific gravity

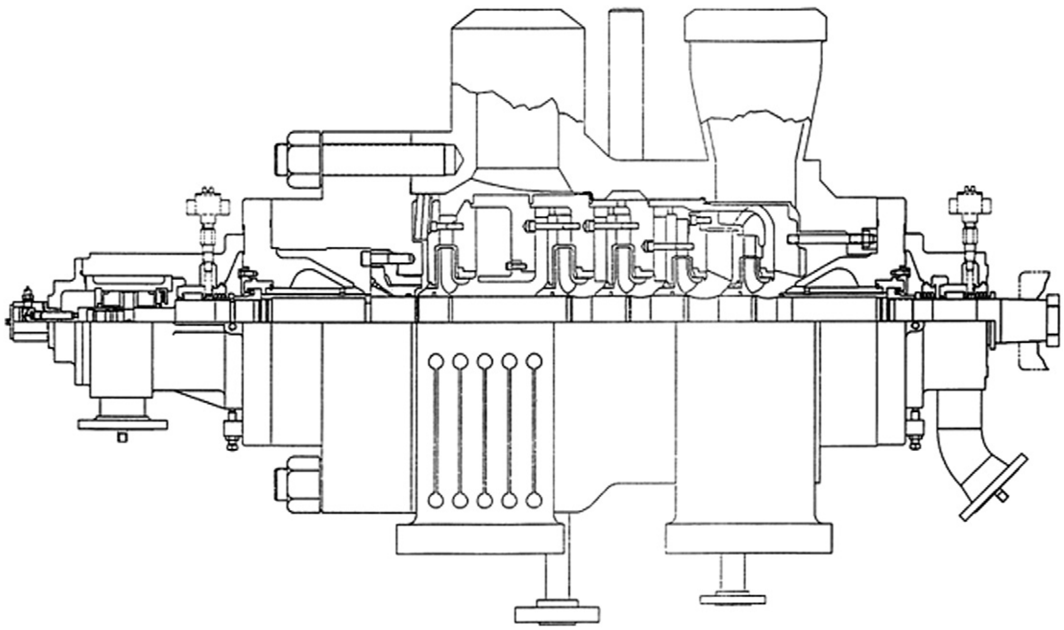


Fig 2.4.7 • Multistage barrel (Courtesy of Demag Delaval)

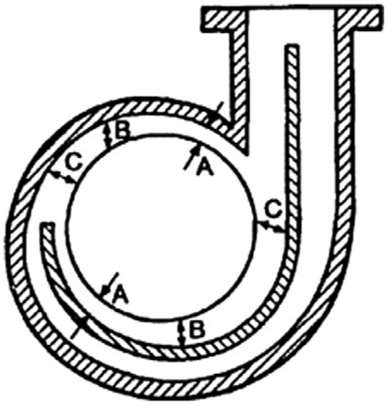


Fig 2.4.9 • Dual volute casing form

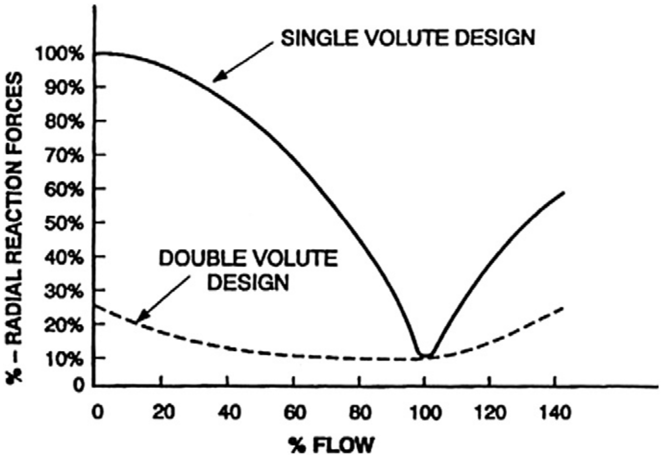


Fig 2.4.10 • Volute radial reaction forces versus capacity

Best Practice 2.5

Double suction pump optimum reliability is achieved by:

- Having minimum 3 pipe diameter straight pipe into the suction
- Suction piping routed perpendicular to the pump centerline
- Keeping suction specific speed below 10,000 (US customary units)

Lessons Learned

Double suction pumps have the greatest reliability problems of all centrifugal pump types, and require special attention during the selection process.

Failure to follow the double suction pump best practices noted above has resulted in the following field reliability issues and MTBFs less than 12 months:

- Impeller cavitation
- Pump seizure – at wear rings
- Shaft breakage
- Thrust bearing failure
- Seal failure

Benchmarks

This best practice has been used since the mid-1980s during the pump selection and model review phases for all double suction pumps. Prior to that time, all of the reliability issues noted in the lessons learned section above were experienced, and many pumps were classified as "bad actors" (MTBFs below 12 months).

B.P. 2.5. Supporting Material

Piping

Piping accounts for the connection of the machinery to the environment surrounding the equipment. Improper piping assembly, like improper foundation installation, has resulted in reduced rotating equipment reliability.

General practices

Figure 2.5.1 presents piping considerations that will result in proper installation of equipment of high reliability.

Piping must always be floated to the machine and NOT first mounted on the machine. A machine is definitely not a pipe support. During construction, observe that piping is first mounted to vessels in pipe racks, and then – and only then – floated to the machine. Any other procedure is totally unacceptable.

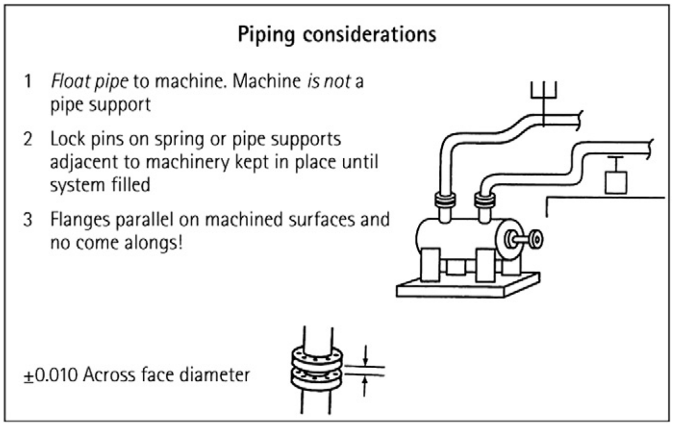


Fig 2.5.1 • Piping considerations

In bolting piping to machinery, lock pins on springs or pipe supports must be kept in place until the system is filled with liquid or gas, since they are designed to support the piping

during the operation of equipment. Installation of piping to equipment flanges must be performed with care. Bolts should be freely removable from mating piping and equipment flanges without the use of force (come alongs). Most importantly, flanges must be parallel on the machine surfaces of the mating and equipment flange within +/−0.010 across the face diameter. It is wise to always observe if piping has been removed or reassembled during turnarounds. Frequently when this activity is performed, proper procedures are not followed and excessive stresses are exerted on the equipment casing. Since the equipment casing supports the bearings, which ultimately support the shaft by anti-friction bearings or a thin oil film in the case of hydrodynamic bearings, improper piping assembly can significantly affect machinery operation. Keep this fact in mind.

Suction specific speed

N_{SS}, known as suction specific speed, is determined by the same equation used for specific speed N_S but substitutes $NPSH_R$ for H (pump head). As the name implies, N_{SS} considers the inlet of the impeller and is related to the impeller inlet velocity. The relationship for N_{SS} is:

$$N_{SS} = \frac{N\sqrt{Q}}{(NPSH_R)^{3/4}}$$

Where : N = speed
Q = flow – GPM
$NPSH_R$ = Net Positive Suction Head Required

$NPSH_R$ is related to the pressure drop from the inlet flange to the impeller. The higher the $NPSH_R$, the greater the pressure drop. The lower the $NPSH_R$, the less the pressure drop. From the equation above, we can show the relationships between $NPSH_R$, N_{SS}, inlet velocity, inlet pressure drop and the probability of flow separation in Figure 2.5.2.

N_{SS}	$NPSH_R$	Inlet velocity	Inlet passage ΔP	Probability of flow separation
14,000 (High)	Low	Low	Low	High probability
8,000 (Low)	High	High	High	Low probability

Fig 2.5.2 • N_{SS} related to flow separation probability

Based on the information presented in Figure 2.5.2, it can be seen that flow separation will occur for high specific speeds resulting from low inlet velocity. The critical question the pump user needs answered is 'At what flow does the disturbance and resulting cavitation occur? ' This is not easily answered, because the unstable flow range is a function of the impeller inlet design as well as the inlet velocity. A general answer to this question is shown in Figure 2.5.3.

The onset flow of recirculation increases with increasing suction specific speed

Fig 2.5.3 • Recirculation as a function of N_{SS}

Another way of stating Figure 2.5.3 is: 'The higher the value of N_{SS}, the sooner the pump will cavitate when operating at flows below the BEP'. Therefore, before an acceptable value of N_{SS} can be determined, the process system and characteristics of the pumped liquid must be defined.

Best Practice 2.6

Pumps with inducers should use a constant flow control scheme to prevent inducer wear and resulting pump cavitation.

The equipment reliability operating envelope (EROE) for inducers to prevent inducer cavitation and wear out is approximately +/−10% in flow.

Process changes will always be such that a flow change of greater than +/−10% will be present.

Most inducer pumps have radial vane pump impellers yielding a very flat performance curve which results in rapid flow changes for small changes in the process head required.

Design the control scheme to automatically vary the bypass flow for changes in process flow to always maintain a constant pump flow and therefore ensure optimum inducer and impeller MTBF.

Lessons Learned

Operating an inducer supplied pump outside the inducer best efficiency point by more than +/−10% will lead to excessive inducer wear, pump cavitation and high speed pump damage.

I have experienced numerous high speed, inducer supplied pump failures that either use manual minimum flow control systems or automatic minimum flow control systems that are not integrated with the process control system.

This arrangement permits the high speed inducer pump to operate over a flow range much greater than the acceptable flow range for an inducer, which results in significantly reduced pump MTBF.

Observed failure modes have been: inducer, impeller, bearing failure, seal failure, gearbox failure and shaft breakage.

Benchmarks

This best practice has been used since the mid-1990s, after experience of numerous high speed inducer pump failures in a chemical plant. In this case, operating revenue was reduced by more than $5MM in a year. In addition, entire high speed pumps and gearboxes had to be replaced. MTBFs prior to the control system modifications were less than 6 months. Since these modifications MTBFs have exceeded 80 months.

B.P. 2.6. Supporting Material

Objectives of the operating company

During normal operation, the prime objective of the operating company is to maintain maximum reliability and maximum product throughput at minimum operating (maximum pump efficiency) cost (refer to Figure 2.6.1).

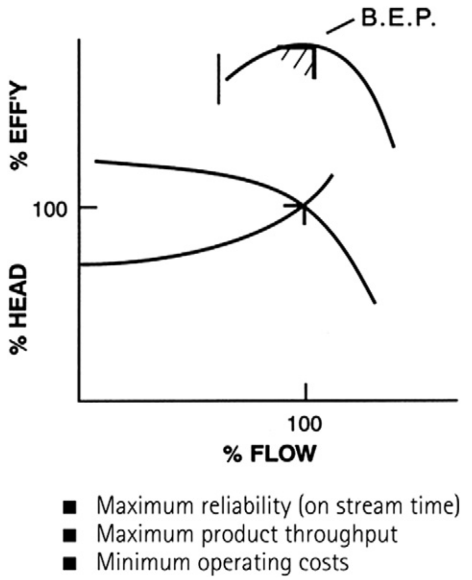


Fig 2.6.1 • The end user's objectives

To meet the operating company objectives, a particular pump is selected, on the basis of the optimum process flow rate and the required system head (energy). Since the pump characteristics will change with wear, erosion or fouling, and the system characteristics will vary, a reliable control and protection system must be selected to continuously meet the objectives noted in Figure 2.6.1 regardless of whether the selected pump is dynamic or positive displacement. Regardless of type, the operating company's objectives are met by meeting process throughput requirements in the most efficient manner.

There are only two options available to vary the process throughput requirements (refer to Figure 2.6.2).

Before proceeding, it is helpful to introduce the concept of an equivalent vessel. Another way to state the operating company's objectives is to state that they want to process all the throughput produced. If we visualize any process system as consisting of a vessel into which the produced product flows, the objective then is to remove (pump) the throughput from the vessel at the same rate of flow as it enters the vessel. If this is accomplished, the following parameters will remain constant in the vessel:

- Flow in and out
- Level
- Pressure

Any point in any process can be thought of as an "equivalent vessel". By controlling either flow, level or pressure, the operating company's objective will be realized (refer to Figure 2.6.3). Therefore the objective of any pump control system will be to maintain a constant, controlled variable set point (flow, level or pressure). The output of the controller can then either vary:

Head required (the process)
Head produced (the pump characteristics)

These options are shown in Figure 2.6.4 for a centrifugal pump and Figure 2.6.5 for a PD pump.

Adjusting head (energy) required

Head required in a pumping system can be changed by adjusting the discharge system resistance using pressure control flow control or level control (refer to Figures 2.6.4 and 2.6.5).

Each of these methods results in closing a throttle valve in the discharge piping which increases the head (energy) required and reduces the flow rate. This action requires more energy (head) to overcome the increased system resistance (refer to Figure 2.6.6).

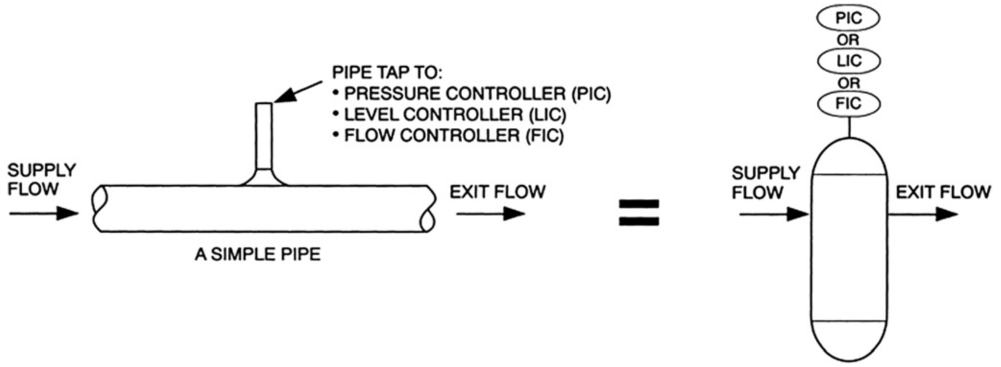
Throttling the discharge of a positive displacement pump increases the discharge pressure, and consequently the power required to overcome the increased head required while passing the same flow rate through the pump. This scheme is not efficient and is usually replaced by a "bypass" throttling arrangement (refer to Figure 2.6.5).

Effects of throttling pump suction

Generally, throttling the pump suction will produce harmful effects and should be avoided (absolute pressure is reduced at the impeller inlet), except in certain cases involving series pump operation and 'hot well' condensate pumps specifically designed for such services (refer to Figure 2.6.7 for the effects of suction throttling).

■ Adjust the head required
■ Adjust the head produced

Fig 2.6.2 • Methods for varying pump throughput



Case	Equivalent Vessel Variable
SUPPLY = EXIT FLOW	CONSTANT
SUPPLY FLOW GREATER THAN EXIT FLOW	INCREASES
SUPPLY FLOW LESS THAN EXIT FLOW	DECREASES

Fig 2.6.3 • Reduce it to an equivalent vessel

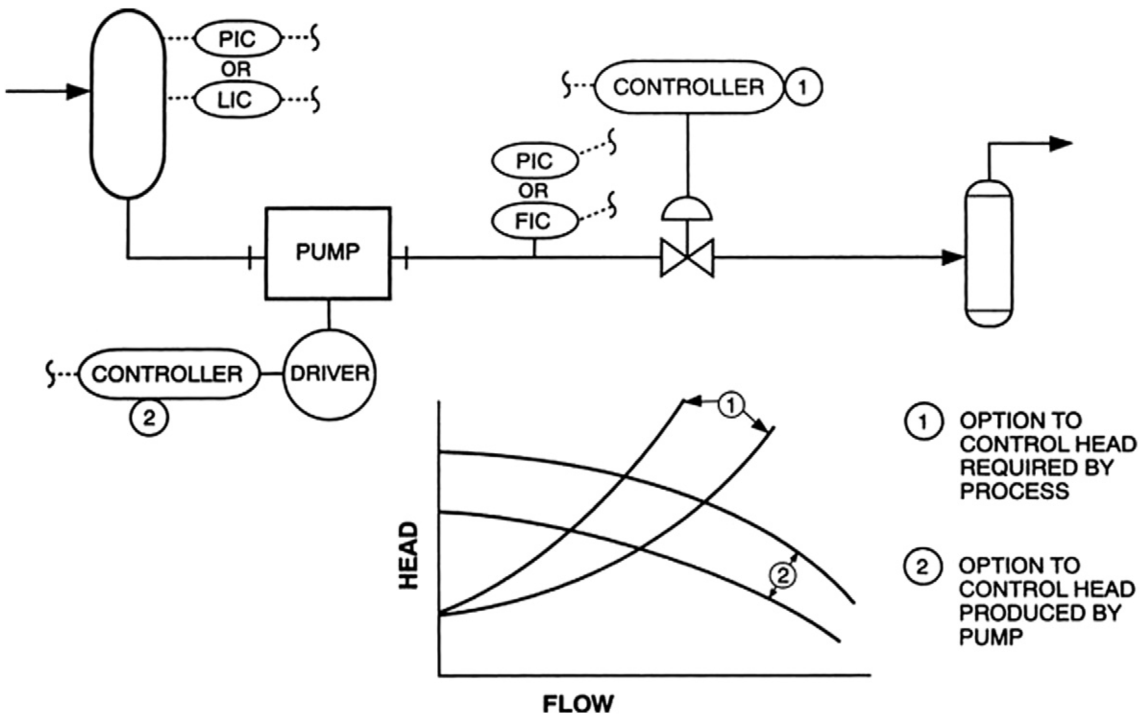


Fig 2.6.4 • Centrifugal pump control options

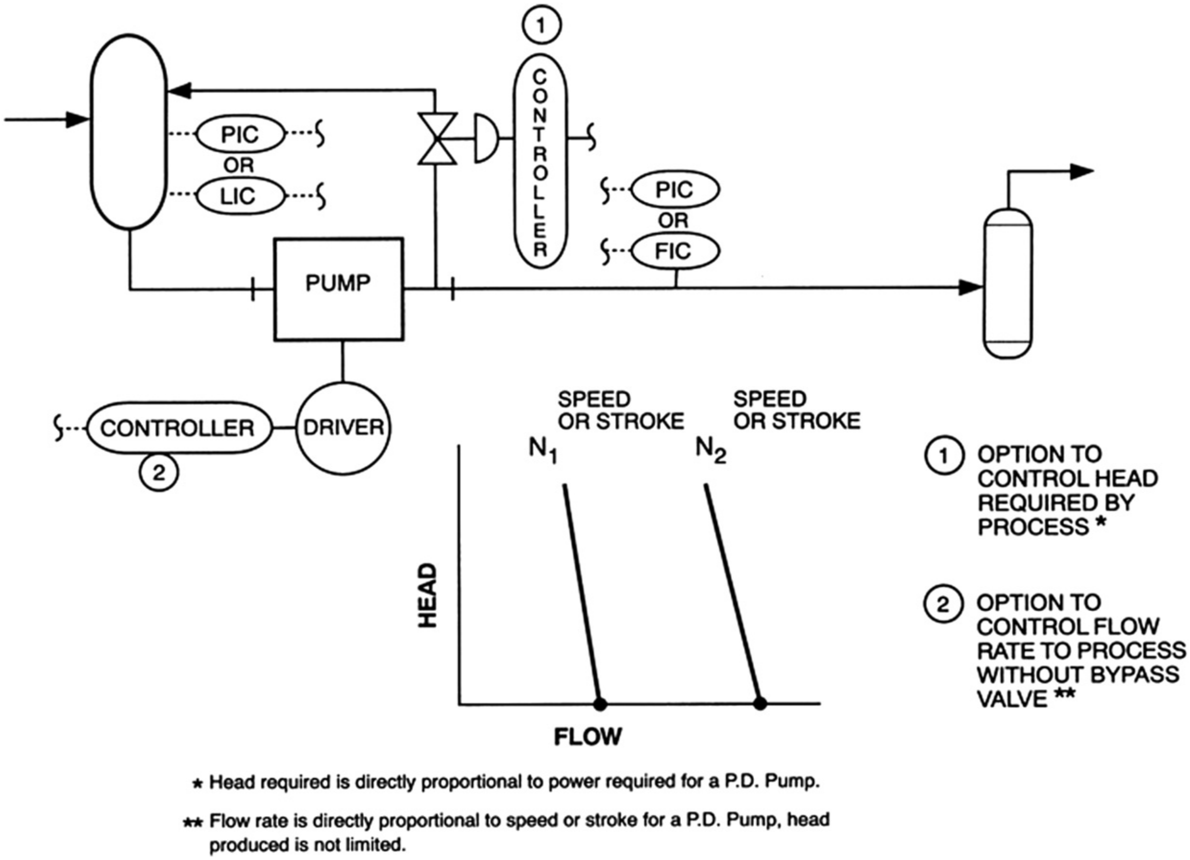


Fig 2.6.5 • P.D. pump control options

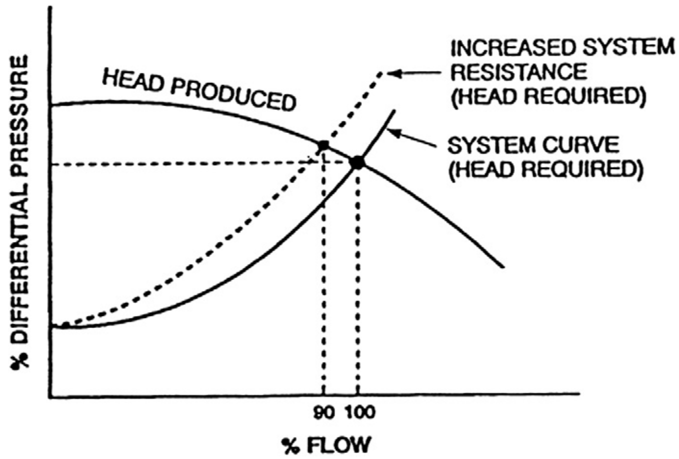


Fig 2.6.6 • Effect of discharge throttling

- Cavitation
- Overheating
- Seal failure
- Bearing failure

Fig 2.6.7 • Effects of suction throttling

Best Practice 2.7

Operate centrifugal pumps within the "Equipment Reliability Operating Envelope" (EROE) to achieve maximum mean time between failure (MTBF).

The Equipment Reliability Operating Envelope (EROE), also called the 'heart of the curve', assures maximum centrifugal pump MTBF by avoiding all operating areas of hydraulic disturbances.

We define the general EROE range as + 10% to −50% in flow from the pump best efficiency point.

This range will be reduced for double flow pumps and high speed inducer (see B.P: 2.6) pumps. Please refer to the supporting material below for additional details.

Lessons Learned

Failure to establish EROE limits will lead to low MTBF of centrifugal pumps

We have found that approximately 80% of centrifugal pump reliability reductions (sources of low MTBF) are due to process changes that cause the pump to operate in either a high flow or low flow range. This exposes the pump to hydraulic disturbances resulting in low MTBF.

Establishing operator EROE targets for all critical site pumps and all bad actor pumps (pumps with one or more component failures per year) will ensure optimum centrifugal pump safety and MTBFs.

Benchmarks

This best practice has been used since the late 1990s in refineries, chemical plants and in SAGD (steam assisted gravity drainage) applications in heavy oil fields. Once this best practice had been implemented, pump MTBFs that were less than 12 months ("bad actors") were improved to greater than 80 months.

B.P. 2.7. Supporting Material

Effects of the process on pump reliability and MTBF

The effect of the process on machinery reliability is often neglected as a root cause of machinery failure. It is a fact that process condition changes can cause damage and/or failure to every major machinery component. For this discussion, the most common type of driven equipment – pumps will be used.

There are two (2) major classifications of pumps; positive displacement and kinetic, of which centrifugal types are the most common. A positive displacement pump is shown in Figure 2.7.1. A centrifugal pump is shown in Figure 2.7.2.

It is most important to remember that all driven equipment (pumps, compressors, fans, etc.) react to the process system requirements. They do only what the process requires. This fact is noted in Figure 2.7.3 for pumps.

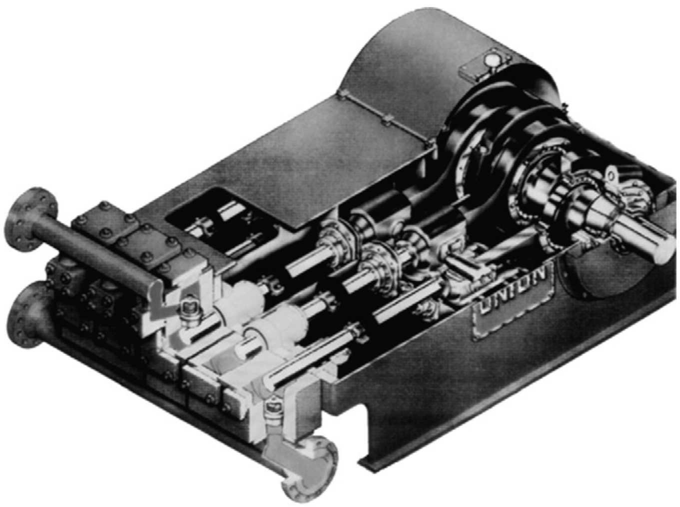


Fig 2.7.1 • Positive displacement plunger pump

Centrifugal (kinetic) pumps and their drivers

Centrifugal pumps increase the pressure of the liquid by using rotating blades to first increase its velocity, and then reduce the velocity of the liquid in the volute. Refer again to Figure 2.7.2.

A good analogy to this procedure is a football (soccer) game. When the ball (liquid molecule) is kicked, the leg (vane) increases its velocity. When the goal tender (volute), (hopefully) catches the ball, its velocity is significantly reduced and the pressure in the ball (molecule) is increased. If an instant replay 'freeze shot' picture is taken of the ball at this instant, it will show that its volume is reduced and its pressure is increased.

The characteristics of any centrifugal pump are hence significantly different from positive displacement pumps, and are noted in Figure 2.7.4.

Refer again to Figure 2.7.3, and note that all pumps react to the process requirements. Based on the characteristics of centrifugal pumps noted in Figure 2.7.4, the flow rate of all types of centrifugal pumps is affected by the Process System. This fact is shown in Figure 2.7.5.

Therefore, the flow rate of any centrifugal pump is affected by the process system. A typical process system with a centrifugal pump installed is shown in Figure 2.7.6.

The differential pressure required (proportional to head) by any process system is the result of the pressure and liquid level in the suction and discharge vessel and the system resistance (pressure drop) in the suction and discharge piping. Therefore, the differential pressure required by the process can be changed by adjusting a control valve in the discharge line. Any of the following process variables (PV) shown in Figure 2.7.6, can be controlled:

- Level
- Pressure
- Flow

As shown in Figure 2.7.5, changing the head required by the process (differential pressure divided by specific gravity) will change the flow rate of any centrifugal pump!

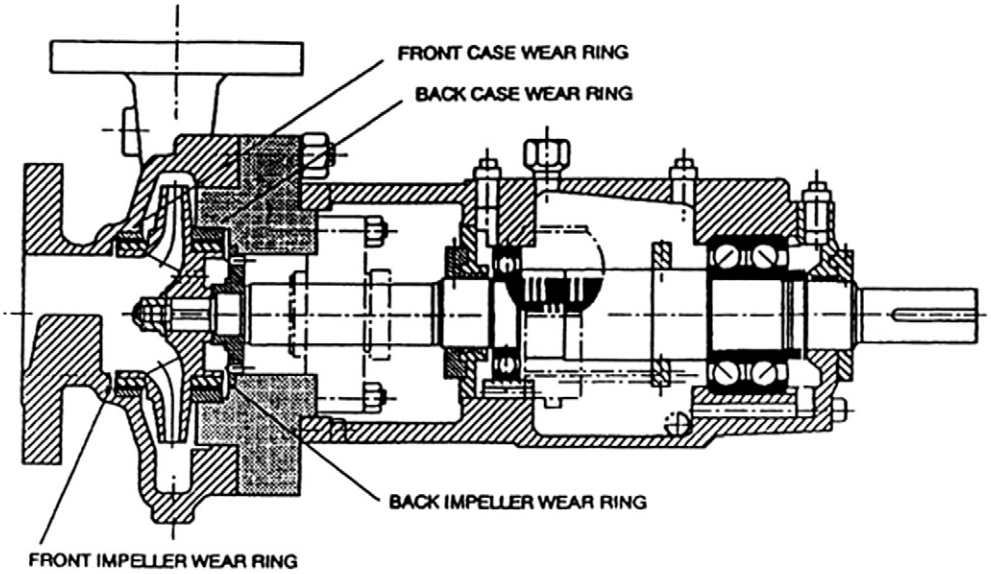


Fig 2.7.2 • Centrifugal pump

- Pumps produce the pressure required by the process
- The flow rate for the required pressure is dependent on the pump's characteristics

Fig 2.7.3 • Pump performance

- Variable flow
- Fixed differential pressure produced for a specific flow*
- Does not require a pressure limiting device
- Flow varies with differential pressure (P_1–P_2) and/or specific gravity

  * ensuring specific gravity is constant

Fig 2.7.4 • Centrifugal pump characteristics

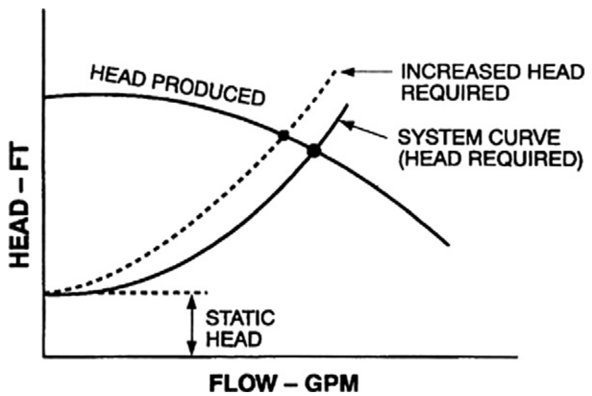


Fig 2.7.5 • A centrifugal pump in a process system

Figure 2.7.7 shows that all types of mechanical failures can occur based on where the pump is operating based on the process requirements.

Since over 95% of the pumps used in this refinery are centrifugal, their operating flow will be affected by the process. Figure 2.7.8 shows that centrifugal pump reliability and flow rate is affected by process system changes.

At this point it should be easy to see how we can condition monitor the centrifugal pump operating point. Refer to Figure 2.7.9.

Driver reliability (motors, steam turbine and diesel engines) can also be affected by the process when centrifugal driven equipment (pumps, compressor and fans) are used.

Refer to Figure 2.7.10 and observe a typical centrifugal pump curve.

Since the flow rate will be determined by the process requirements, the power (BHP) required by the driver will also be affected. What would occur if an $8^1/_2$" diameter impeller were used, and the head (differential pressure) required by the process was low? Answer: Since the pressure differential required is low, the flow rate will increase and for the $8^1/_2$" diameter impeller, the power required by the driver (BHP) will increase.

Therefore, a motor can trip out on overload, a steam turbine's speed can reduce or a diesel engine can trip on high engine temperature. These facts are shown in Figure 2.7.11.

Auxiliary system reliability is also affected by process changes. Auxiliary systems support the equipment and their components by providing clean, cool fluid to the components at the correct differential pressure, temperature and flow rate.

Typical auxiliary systems are:

- Lube oil systems
- Seal flush system
- Seal steam quench system
- Cooling water system

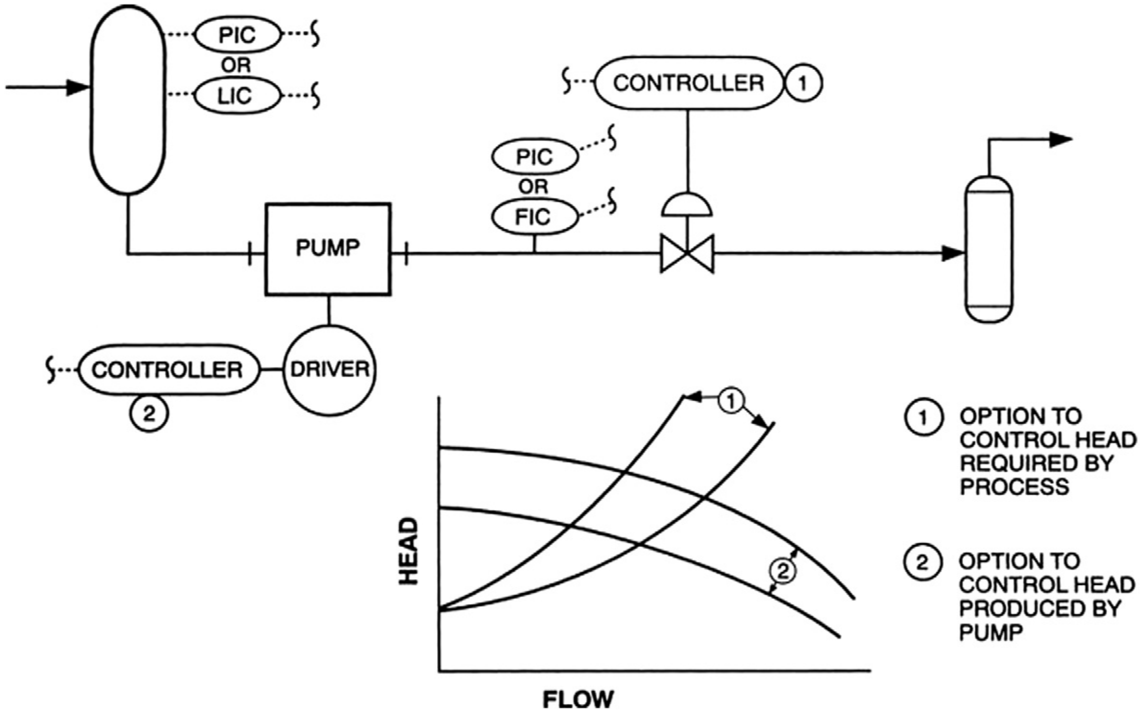


Fig 2.7.6 • Centrifugal pump control options

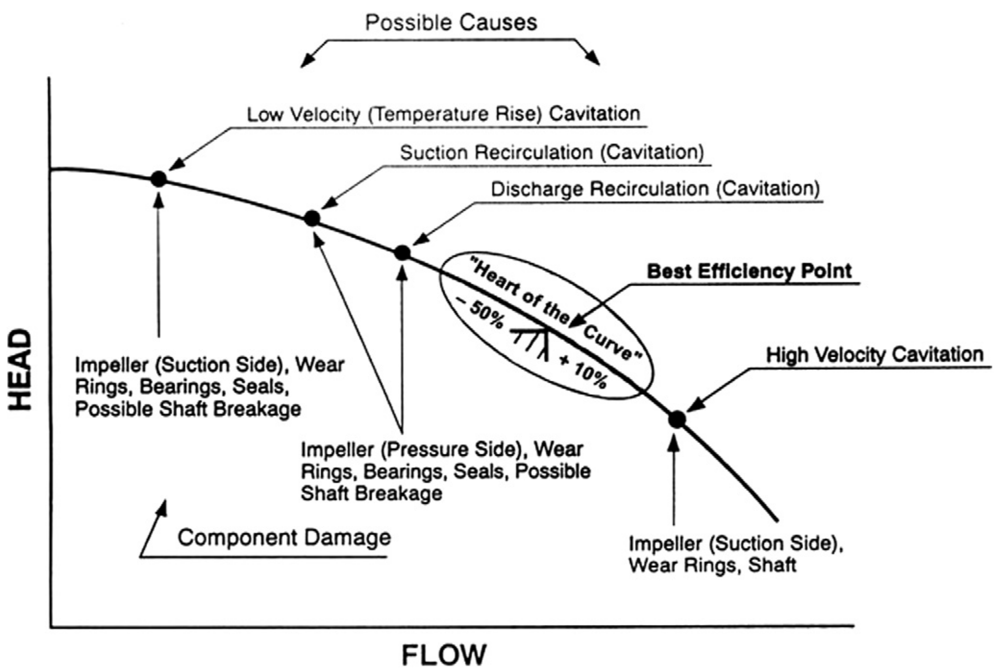


Fig 2.7.7 • Centrifugal pump component damage and causes as a function of operating point

The reliability of machinery components (bearings, seals, etc.) is directly related to the reliability of the auxiliary system. In many cases, the root cause of the component failure is found in the supporting auxiliary system.

As an example, changes in auxiliary system supply temperature, resulting from cooling water temperature or ambient air temperature changes, can be the root cause of component failure. Figure 2.7.12 presents these facts.

- Is affected by process system changes (system resistance and S.G.)
- Is not affected by the operators!
- Increased differential pressure ($P_2 - P_1$) means reduced flow rate
- Decreased differential pressure ($P_2 - P_1$) means increased flow rate

Fig 2.7.8 • Centrifugal pump reliability

- Monitor flow and check with reliability unit (RERU) for significant changes
- Flow can also be monitored by:
  - Control valve position
  - Motor amps
  - Steam turbine valve position

Fig 2.7.9 • Centrifugal pump practical condition monitoring

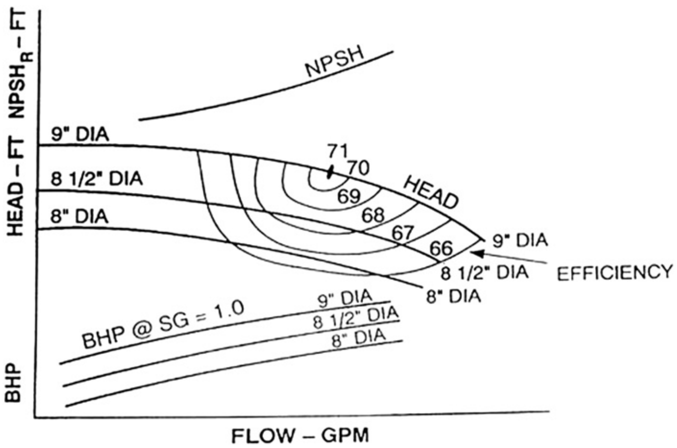


Fig 2.7.10 • A typical centrifugal pump performance curve

- Motors can trip on overload
- Steam turbines can reduce speed
- Diesel engines can trip on high engine temperature

Fig 2.7.11 • Effect of the process on drivers

- Is directly related to auxiliary system reliability
- Auxiliary system reliability is affected by process condition changes
- “Root causes” of component failure are often found in the auxiliary system

Fig 2.7.12 • Component (bearing and seal) reliability

- Monitor auxiliary system condition
- Inspect auxiliary system during component replacement

Fig 2.7.13 • Always "think system"

As a result, the condition of all the auxiliary systems that supports a piece of equipment must be monitored. Please refer to Figure 2.7.13.

EROE (Equipment Reliability Operating Envelope) determination

As noted in Figure 2.7.14, process changes will vary the flow of any centrifugal pump. If the flow to a centrifugal pump is too high or too low, hydraulic disturbances will be present that can change the pumped fluid pressure and/or temperature. Since the majority of mechanical seal applications use the pumped fluid in the seal chamber, the seal chamber pressure and/or temperature will be affected. These changes will directly impact mechanical seal life and reliability.

Decreased Pump Flow:
- Increased P2
- Decreased P1
- Decreased S.G.

Increased Pump Flow:
- Decreased P2
- Increased P1
- Increased S.G.

Fig 2.7.14 • Process effects on centrifugal pump flow

Figure 2.7.15 shows a typical centrifugal pump head vs. flow curve with the following items noted:

The ‘desirable region’ of operation – heart of the curve or EROE
Regions of hydraulic disturbances – on the upper portion of the curve
The pump components affected – on the lower portion of the curve

The ‘heart of the curve’ is the flow region for any centrifugal pump that will be free of hydraulic disturbances, and where the seal fluid should be free of vapor if the seal fluid conditions stated on the pump and seal data sheets are present during pump field operation.

This Flow Region is also called the:

EROE – The Equipment Reliability Operating Envelope

Figure 2.7.16 presents facts concerning the EROE.

In many pump installations, a neither a flow meter nor a suction pressure gauge is installed. A calibrated suction pressure gauge can be installed in the suction pipe drain connection (always present). **Be sure to obtain a MOC (management of change) and work permit and any other plant required permission prior to installing a suction pressure gauge as the pumped fluid could be sour (H_2S), flammable and/or carcinogenic.**

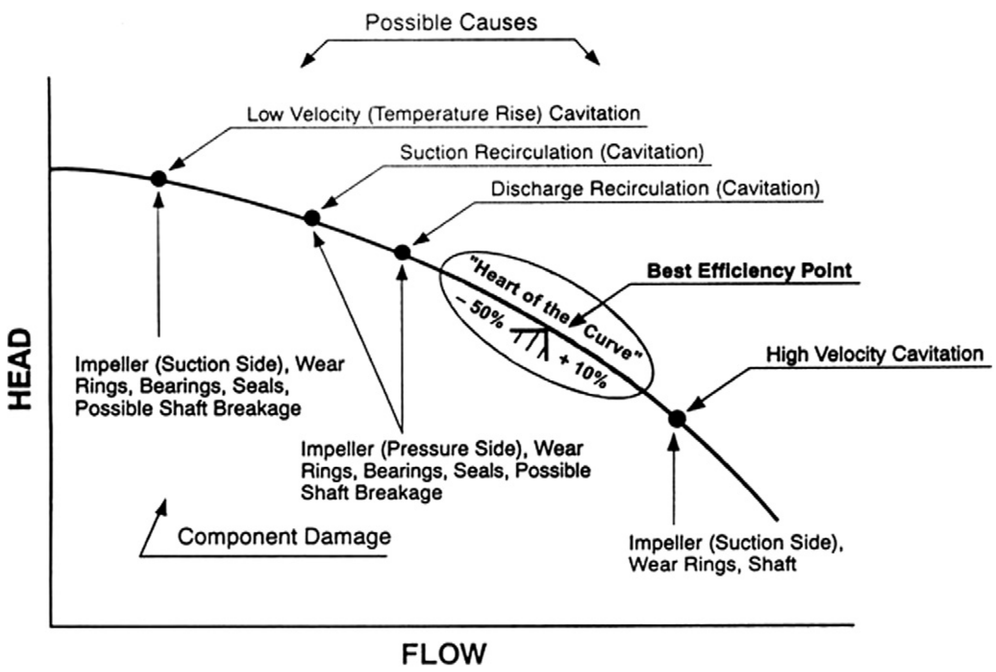


Fig 2.7.15 • Centrifugal pump head vs. flow curve

1. The EROE flow range is + 10% and − 50% of the pump best efficiency point (BEP) flow
2. All 'bad actor pumps' (more than one component failure per year) should be checked for EROE.
3. To determine that the pump is operating in EROE:
   - Calculate the pump head required
   - Measure the flow
   - Plot the intersection of head and flow on the pump shop test curve

Fig 2.7.16 • EROE Facts

If a flow meter is not installed, Figure 2.7.17 defines the options that are available to determine the pump flow, so the EROE can be obtained.

The flow values in Figure 2.7.17 can be determined by hand calculations using the equations available in any pump text (power equation and pump temperature rise equation).

It can be seen that the EROE will provide a reasonable guide that usually will eliminate hydraulic disturbances that can cause seal chamber pressures and temperatures to change and lead to premature seal wear and/or failure. Note that the stated EROE low flow range can be reduced if the pump or fluid have any of the characteristics noted in Figure 2.7.18.

1. Measure motor amps and calculate power
2. Record control valve position, valve differential pressure, fluid S.G. and calculate valve flow (pump flow).
3. Measure pump pipe differential temperature and calculate pump efficiency
4. Obtain an ultrasonic flowmeter to measure flow
5. For items 1 and 3, locate the calculated value (power or efficiency) on the pump test curve to determine pump flow

Fig 2.7.17 • Available pump flow determination options

- Pumps with suction specific speeds > 8,000 (customary units)
- Double suction pumps
- Water pumps with low NPSH margin
- Fluids with S.G. < 0.7
- Pumps with Inducers

Fig 2.7.18 • Factors that can reduce low flow EROE range

Therefore, we always recommend that the first step in seal condition monitoring be determination of pump operation within its EROE. If the 'bad actor' pump is operating outside its EROE, we recommend the action shown in Figure 2.7.19.

If seal reliability does not improve when operating within the EROE, further investigation is required concerning the process conditions in the seal chamber and/or flush system.

- Consult operations to determine if process changes can be made to operate in EROE
- Define target EROE parameters for operations (flow, amps, control valve postion, delta T)

Fig 2.7.19 • If a centrifugal pump is outside its EROE

Best Practice 2.8

Monitor pump EROE in control room for critical centrifugal pumps and define EROE targets.

Pump flow range can be monitored in the control room by inputting the pump shop test curve and dumping the following transmitter signals into spreadsheets to calculate the pump head and flow:

- Inlet pressure
- Discharge pressure
- Flow
- SG – A constant value input from actual liquid sample

EROE targets should be established in flow or the following other methods if flow meters are not installed for each pump:

- Control valve position
- Motor amps
- Pump inlet and discharge piping differential temperature

Lessons Learned

Critical centrifugal and 'bad actor pumps' require constant surveillance by operators to ensure optimum safety and reliability.

It has been my experience that majority of centrifugal pump mechanical seal and bearing failures are the result of the pump being forced out of the EROE by changes in process required head (inlet pressure, discharge pressure and/or specific gravity) without the operators being aware of this.

Benchmarks

This best practice has been followed since 2000 in heavy oil field pump applications, where the operating conditions require the use of slurry pumps which are subject to rapid wear. The use of control room operating point monitoring with EROE alarms has extended pump MTBFs to over 36 months, from less than 12 months before this best practice was implemented.

B.P. 2.8. Supporting Material

Please refer to B.P: 2.7 for additional supporting material.

Table 2.8.1 Pump Component Condition Monitoring

	Pump Component Condition Monitoring			
Performance				
Item #				
Group Name				
Input				
P_1 (kg/cm^2)				
P_2 (kg/cm^2)				
S.G.				
Pump Speed (RPM)				
Flow Rate (m^3/hr)				
Calculate				
Head (m)	-	-	-	-
Flow Determination[1]				
Control Valve Position				
CV for Valve and Trim Type[2]				

Table 2.8.1 Pump Component Condition Monitoring—Cont'd

Fluid S.G.				
Control Valve ΔP				
Calculated Valve Flow	-	-	-	-
Motor Amps				
Volts				
Power Factor				
Motor Eff'y				
Calculated Power	-	-	-	-
Flow From Pump Curve[3]				
T1 (Deg. C)				
T2 (Deg. C)				
CP (Specific Heat)				
Calculated Head	-	-	-	-
Calculated Pump Eff'y	-	-	-	-
Flow From Pump Curve[3]				

Table 2.8.1 Pump Component Condition Monitoring—Cont'd				
Pump Maintenance Required[4]				
EROE Determination				
BEP Flow[5]				
EROE Min Flow	-	-	-	-
EROE Max Flow	-	-	-	-
Is Pump in EROE	-	-	-	-
EROE Targets for Operations				
Flow				
Amps				
Pump Δ T (Measured on Inlet and Discharge Pipes)				

Table 2.8.1 Pump Component Condition Monitoring—Cont'd
Notes
1. If a flow meter is not available, the pump flow will be determined by one of the alternative methods noted in tab 7 of the supplementary manual. These alternative methods are: 1. Portable Ultrasonic Flowmeter 2. Control Valve Position to calculate flow 3. Motor Amps to calculate power 4. Pump Δ T to calculate pump efficiency.
2. The CV can be found on the valve manufacturer's curve for specific valve type and trim. Note that most of the major valve manufacturers post these curves on their websites.
3. Flow will be estimated using the original shop test curve. Note that if the pump is not in good condition these estimates will not be very accurate.
4. Pump maintenance should be considered if the head and flow (operating point) when plotted on the test curve is approx. 10% below test curve flow for calculated head.
5. Flow at highest efficiency taken from the shop test curve in the supplementary manual.

Best Practice 2.9

Ensure that centrifugal pumps have a minimum head rise to shutoff (zero flow) of 7%.

The lower the head rise of a centrifugal pump (pump head produced at zero flow divided by head produced at rated point), the greater the change in pump flow for a given change in process head required.

The operating flow for centrifugal pumps with head rises that are below 7% will be easily changed by small variances in the process head required.

Try to select a centrifugal pump with the highest head rise possible, while taking into consideration the flow range and NPSH available for the required pump.

Lessons Learned

Centrifugal pump flow is determined by the head required by the process. If the pump curve is flat (zero head rise), flow will reduce to zero with any increase in head required by the process, and subject the pump to extensive damage and possible seizing.

Centrifugal pumps with flat head curves have been subject to extensive damage resulting from minimum flow bypass valves not opening when required. Pumps with head rise less than 5% are usually ‘bad actors’ (more than one failure per year).

Benchmarks

This best practice has been used since the mid-1970s to ensure highest centrifugal pump safety and MTBF (greater than 80 months) in all pump applications.

B.P. 2.9. Supporting Material

The centrifugal curve

When a centrifugal pump is designed, its performance characteristics are defined for a range of flows and head (energy) produced for fixed impeller geometry and a variety of impeller diameters. Figures 2.9.1 and 2.9.2 show the relationships between the head (energy) produced and the flow through a single stage centrifugal pump. By the affinity laws, pumps of the same type, fitted with impellers of similar design, will have characteristics curves of the same shape.

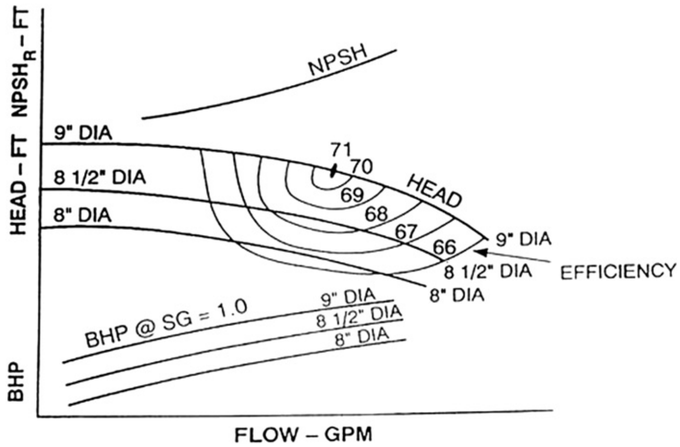


Fig 2.9.1 • A typical centrifugal pump performance curve

$H = \frac{V^2}{2g}$ H = energy expressed in feet of liquid where:

$H = \text{Head in } \frac{m - kgf}{kgM}\left(\frac{ft - lbf}{lbM}\right)$

V = velocity in m/s (ft/s)

g = acceleration due to gravity in m/s^2 (ft/s^2)

$\text{Power} = \frac{Q \times H \times \rho \times g}{\eta \times 3.6 \times 10^6}$ (33,000 if in BHP)

Where:
Power – kW (BHP)
Q = Flow in m^3/hr (GPM)
ρ = Density in kg/m^3 (lb/ft^3)
η (Efficiency) = ratio of power output to power input
$NPSH_R$ = energy required to avoid vaporization in pump suction passage.

Fig 2.9.2 • Centrifugal pump performance definitions

Head produced

The head produced by a centrifugal pump varies inversely with the flow rate. The curve head rise is a function of the impeller inlet and discharge blade angles. Typical centrifugal pump head rise values from design point to shutoff are 5–15%.

When the head required by the process exceeds that which can be produced by a single stage centrifugal pump, multistaging is used to produce the energy required by the system. Multistaging is simply two or more impellers acting in series within a single casing to produce the total head (energy) required. It is common practice for each impeller to produce an equal amount of energy (refer to Figure 2.9.3).

- total net system energy (ft. head) – 366 meters (1200 ft.)
- number of impellers selected – 4
- energy (ft. head) produced for impeller – $\frac{366 \text{ M (1200 Ft)}}{4}$ = 91.5 M (300 Ft)/Impeller

Fig 2.9.3 • Example – multistaging

Flow

The flow rate of a centrifugal pump varies inversely with the head (energy) required by the process. For a given impeller design operating at a constant speed, increased process head requirements will reduce centrifugal pump flow rates. Since the typical head rise values for centrifugal pumps are 5–15%, a relatively small change in process head requirements can result in significant flow reductions and possible impeller recirculation on operation near zero flow (shutoff).

Efficiency

The pump efficiency is at its maximum at the pump design point using the maximum diameter impeller. Refer to Figure 2.9.1. The pump design point, often referred to as the BEP – best efficiency point – is the flow where minimum losses occur in the pump stationary passages and the impeller. At off–design flows, separation losses (low flows) and turbulence losses (high flows) increase internal produced head losses and reduce pump efficiency.

Horsepower

The horsepower required by a centrifugal pump varies directly with the specific gravity of the pumped liquid. Horsepower is the only parameter on a typical centrifugal pump curve that is affected by the specific gravity of the pumped liquid. Most pump curves present the horsepower curve based on water SG = 1.0. For pumped liquids of any other SG value, the horsepower on the pump curve must be multiplied by the actual specific gravity.

$NPSH_R$

The net positive suction head required by a centrifugal pump varies approximately with the square of the flow rate, since it is a measure of the pressure drop from the pump suction flange to the eye of the first impeller.

The $NPSH_R$ is also influenced by the pump rotational speed, and varies somewhat less than the rotational speed squared.

The limits of the centrifugal pump curve

The centrifugal pump curve has high and low flow limits, which can cause significant mechanical damage to the pump if not avoided. At the low flow end of the curve, flow recirculation can damage a pump, while at the high flow end, excessive $NPSH_{REQUIRED}$, horsepower and choke flow can result in mechanical damage to impellers, casing, shaft, bearings and seals. Each of these factors is discussed below.

Low flow operation

As we examine these factors we can see that oversizing a centrifugal pump will result in low flows through the impeller. A portion of the flow will reverse itself and set up turbulence as it reenters. The abrupt change in direction and very high acceleration can result in cavitation on the back side of the impeller vane (refer to Figure 2.9.4).

Oversizing an impeller can significantly affect performance and mechanical reliability, as indicated in Figure 2.9.5.

Pumps are designed to operate at minimum radial thrust loads at their best efficiency point. Low flow operation results in

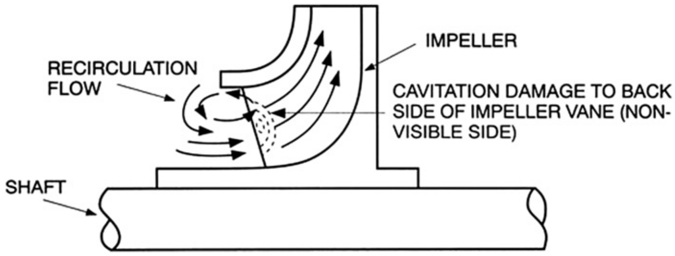


Fig 2.9.4 • Recirculation flow pattern in impeller at low flows

Operation at low flows can result in:
- Internal recirculation damage to impeller
- Operation at less than best efficiency point
- High radial loads
- Bearing failures
- Seal failures
- High internal temperature rise and requirement for minimum flow bypass

Fig 2.9.5 • Effects of pump oversizing

high radial loads, which can cause premature bearing failures unless bearings are selected to accept these higher loads in anticipation of operation at low flows. Pressure surges and flashing of the liquid can also occur at low flows. This can cause loading and unloading of the mechanical seal faces, which can result in a seal failure. Depending on the fluid being pumped, low flow operation can result in a high temperature rise through the pump, because the amount of energy absorbed by the liquid is low compared to that absorbed by friction losses. Refer to Figure 2.9.6 for calculation of the temperature rise through a pump.

$$\text{RISE, DEG C} = \frac{H}{367{,}100 \times C_P}\left[\frac{1}{\text{EFF'Y} - 1}\right]$$

$$\left(\text{RISE, DEG C} = \frac{H}{778 \times C_P}\left[\frac{1}{\text{EFF'Y} - 1}\right]\right)$$

$$H = \text{head in } \frac{m - kgf}{kgM}\left(\frac{ft\text{-}lbf}{lbM}\right)$$

eff'y = efficiency at pumping rate
CP = specific heat, kJ/kg-°C (BTU/LB–°F)
367,100 = m. kg/kJ (778 = ft. LB/BTU)

Fig 2.9.6 • Temperature rise through a pump

The above relationship can also be used to determine the approximate flow rate of any centrifugal pump, by measuring the pipe temperature rise. Referring to the particular pump shop test curve for the calculated efficiency will allow the approximate pump flow rate to be determined. Note: This approach assumes the pump is in new condition. A worn pump will reduce the flow to a greater extent.

High flow operation

Selecting a pump to operate far to the right of best efficiency point can also result in potential problems, as highlighted in Figure 2.9.7.

Operation at high flows can result in:
- High to overloading horsepower with reduced system resistance
- Operation in the "break" of head capacity curve (significant changes in head with no change in flows)
- Higher NPSH required than available
- Recirculation cavitation at impeller tips

Fig 2.9.7 • Effects of pump operation at high flows

Pump curve shapes

The characteristic curves normally associated with centrifugal pumps can be flat, drooping, rising, stable and unstable depending upon their shape. Figure 2.9.8 illustrates the different curve shapes and Figure 2.9.9 defines each type. The pump curve shape can play a significant role in determining if stable operation in a given process system is possible. Flat or drooping head curves (Figure 2.9.8 – curves 1 and 2) can result in unstable operation (varying flow rates). Pumps should be selected with a rising head curve or controlled such that they always operate in the rising region of their curve.

Increasing head produced by a centrifugal pump

The affinity laws can be used to increase the head available from a centrifugal pump. Head produced by a centrifugal pump is a function of impeller tip speed. Since tip speed is a function of impeller diameter and rotational speed, two options are available. The characteristic curve can be affected by either a speed change or a change in impeller diameter with speed held constant. Figure 2.9.10a and Fig. 2.9.10b show this relationship.

The affinity laws

In actual practice, the affinity laws provide an approximation between flow, head and horsepower as pump impeller diameter or speed is varied. The values actually observed will vary somewhat less than predicted by the affinity laws. That is, the actual exponents in the affinity equations are slightly less than their stated values and are different for each pump. This results from friction in hydraulic passages and impellers, leakage losses and variation of impeller discharge vane angles when diameters are changed. Pump manufacturers should be contacted to confirm actual impeller diameters and speed changes to meet new duty requirements.

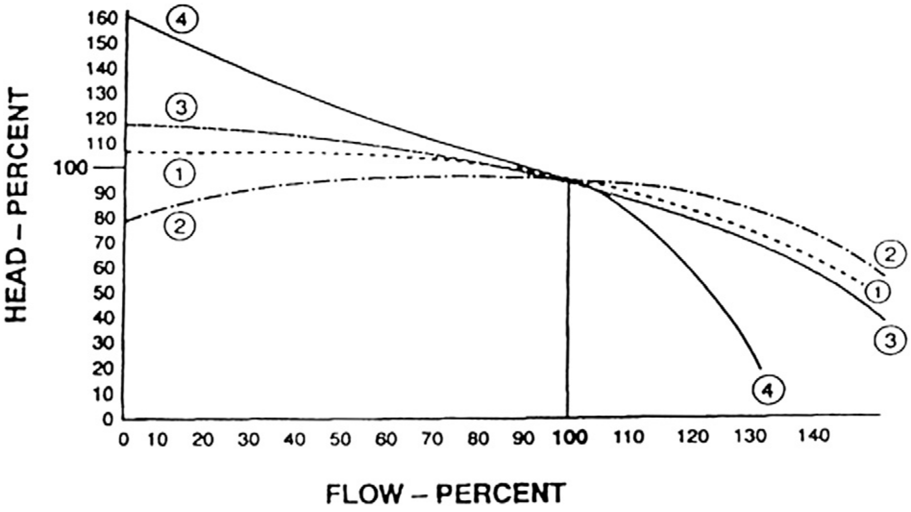


Fig 2.9.8 • Centrifugal pump performance curve shapes

Flat curves (1 and 2), show little variation in head for all flows between design point and shut-off
Drooping curve (2), head at shut-off is less than head developed at some flows between design point to shut-off
Rising curve (1,3,4), head rises continuously as flow decreases from design point to shut-off
Steep curve (4), large increase between head developed at design flow and that developed at shut-off
Stable curve (1,3,4), one flow rate for any one head
Unstable curve (2), same head can be developed at more than one flow rate

Fig 2.9.9 • Definition of characteristic curve shape

Increasing head produced by a pump

$$H_1 = \left(\frac{D_1}{D}\right)^2 H \quad \text{or} \quad H_1 = \left(\frac{N_1}{N}\right)^2 H$$

where: D_1 = diameter of changed impeller
D = diameter of existing impeller
H_1 = head in feet changed impeller
H = head in feet existing impeller
N_1 = speed (RPM) changed condition
N = speed (RPM)existingcondition

Fig 2.9.10a • The affinity laws

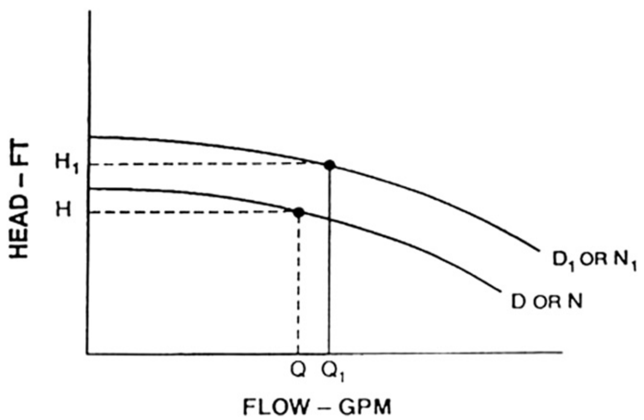


Fig 2.9.10b • The affinity laws

Best Practice 2.10

Install discharge flange orifices for centrifugal pumps with less than 5% head rise to ensure stable pump operation.

Centrifugal pumps with flat head vs. flow curves (less than 5% head rise from rated to zero flow) produce rapid flow changes for small process changes.

Size the impeller for increased head (to compensate for the orifice pressure drop).

Installing a discharge orifice in the pump volute or on the discharge flange will produce the desired head rise, and will result in a pump characteristic that produces gradual flow changes for small process changes.

This procedure is especially effective for high speed pumps which have a characteristic flat, low head rise performance curve.

Lessons Learned

Failure to correct pumps, especially high speed type (integral gear centrifugal), with discharge orifices, to produce a performance curve characteristic of sufficient head rise will lead to low pump MTBF.

Low head rise curves, less than 5%, have resulted in the following reliability issues:

- Rapid inducer and/or impeller wear
- Pump seizure
- Bearing failure
- Seal failure
- Shaft breakage
- Gear box failure

Benchmarks

This best practice has been used since the mid-1970s in all high speed pump applications, and where any centrifugal pump had a low head rise characteristic. This has resulted in centrifugal pump operation of optimum safety and reliability. MTBFs using this best practice have been greater than 80 months.

B.P. 2.10. Supporting Material

Integral gear centrifugal pump

This type of pump is used for low flow applications requiring high head. Refer to Figure 2.10.1. The pump case design is similar to the inline, but incorporates pump bearings and an integral gear to increase impeller speeds over 30,000 RPM.

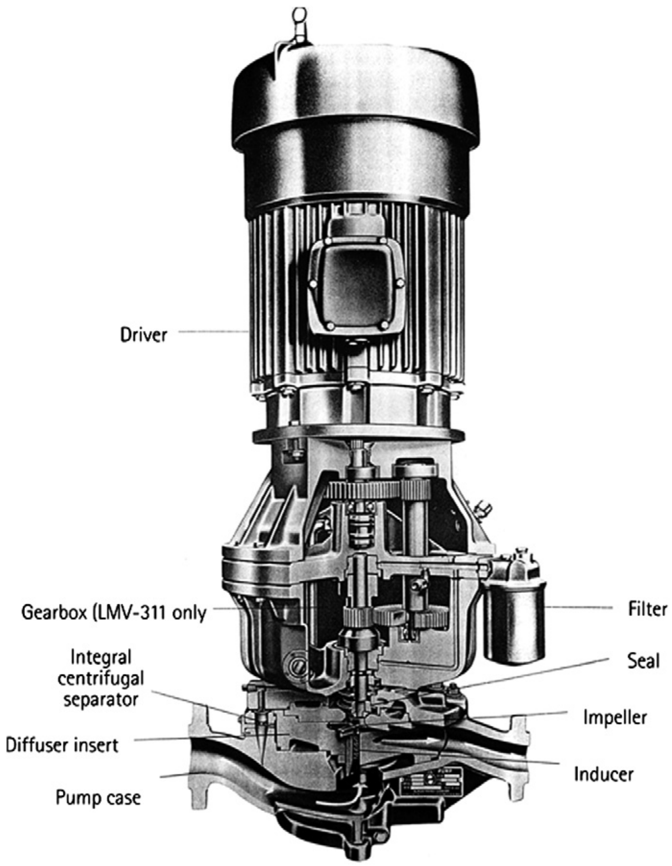


Fig 2.10.1 • Integral gear centrifugal pump (Courtesy of Sunstrand Corp.)

Best Practice 2.11

Define NPSH margins for the maximum flow point of a centrifugal pump to prevent cavitation for all operating cases.

Ensure that the NPSH margin is acceptable at the maximum possible flow, based on the process control system design.

Confirm that the pump is protected, to ensure that there will be a recirculation margin at minimum operating flow.

Accurately define all liquid characteristics (vapor pressure, viscosity, specific gravity and pumping temperature).

Lessons Learned

Pump NPSH available to NPSH required margins are traditionally set for only the rated operating point. Centrifugal pump flow is a result of process system requirements and the installed process control system.

Centrifugal pump applications that process liquids with high vapor pressures are susceptible to low NPSH margins when operating at heads less than the rated head. Failure to select the NPSH margin considering the installed control and protection system has resulted in reduced pump safety and reliability.

Benchmarks

I have used this best practice since 1990 for pumps with a high vapor pressure (typically specific gravity below 0.7). This best practice has resulted in high vapor pressure applications of optimum safety and reliability (MTBFs exceeding 80 months).

B.P. 2.11. Supporting Material

Defining the process system

Reviewing the proposed process system prior to the purchase of a pump is strongly recommended. Figure 2.11.1 presents a typical process system with various control alternatives (flow, level, pressure).

The approach that should be followed when purchasing a pump is to define the required operating range of the pump based on the process system design and process requirements.

Once the operating range is defined, hydraulic calculations will determine the required flows, heads and $NPSH_{AVAILABLE}$. Care should be taken to define liquid composition and temperature as accurately as possible, since these items will determine the vapor pressure which in turn defines the $NPSH_{AVAILABLE}$. The steps for defining the process requirements are summarized in Figure 2.11.2.

Once the process requirements have been accurately defined, a pump can be selected that will meet these requirements without the risk of hydraulic disturbances.

Selecting a pump for hydraulic disturbance free service

Having discussed the concepts that are used to prevent liquid disturbances, and the requirements for accurately defining the

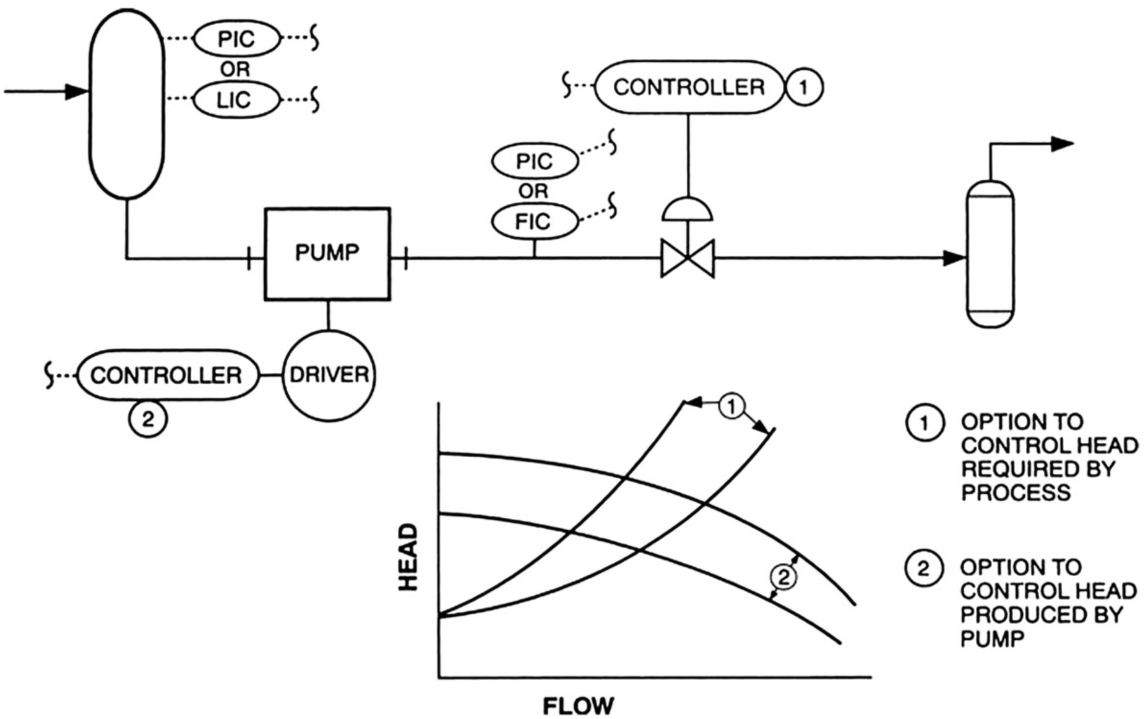


Fig 2.11.1 • Centrifugal pump control options

- Define operating range of application
- Accurately define liquid characteristics
  - Vapor pressure
  - Pumping temperature
  - Viscosity
- Perform hydraulic calculations for all required flow rates to determine:
  - Head required
  - $NPSH_{AVAILABLE}$

Fig 2.11.2 • Preventing liquid disturbances by accurately defining process requirements

1. NPSH margin at maximum operating flow
2. Approximate recirculation margin at minimum operating flow
3. NPSH margin at minimum operating flow

Fig 2.11.3 • Hydraulic disturbances – areas of concern

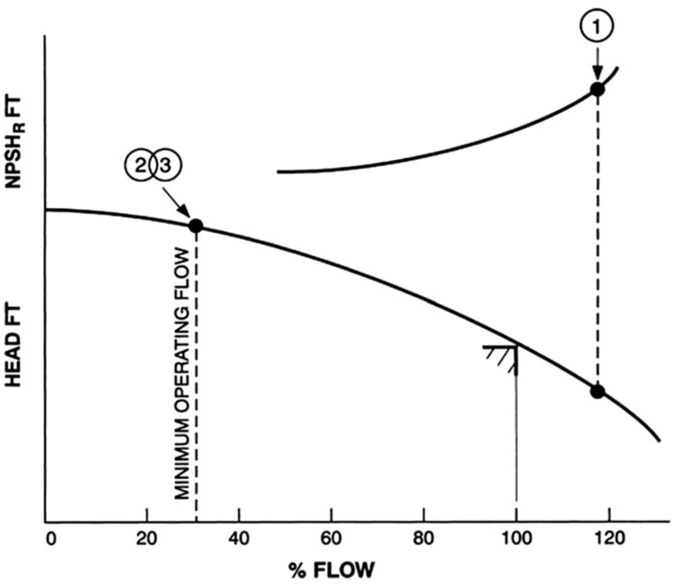


Fig 2.11.4 • Hydraulic disturbance – areas of concern

process system requirements, this information can be used to select a pump free of hydraulic disturbances.

Based on previous discussions, there are three areas of concern to ensure trouble-free operation (see Figure 2.11.3).

Refer to Figure 2.11.4 for a typical pump performance curve.

The practical approach is to select a type of pump that will enable operation under all conditions in Figure 2.11.3 if possible. Figure 2.11.5 presents guidelines for selecting a pump free of hydraulic disturbances.

The guidelines presented in Figure 2.11.5 attempt to cover all situations; however, technical discussions with the pump vendor is encouraged.

Step Action

1. Confirm $NPSH_A \geq NPSH_R$ at maximum operating flow. If margin less than two (2) feet, require witnessed $NPSH_R$ test. If $NPSH_R > NPSH_A$:
   Increase $NPSH_A$ by:
   - Increasing suction drum level
   - Decreasing pumping temperature
   - Decreasing suction line losses

   Reselect pump (if possible)
   Select canned pump
2. For the pump selected calculate N_{SS} based on pump BEP conditions. Note: if double suction first stage impeller, use 1/2 of BEP flow
3. If $N_{SS} > 8000$, contact pump vendor and require following data for actual pump fluid and conditions:
   - Predicted onset flow of cavitation caused by recirculation for actual fluid conditions
   - Reference list of proposed impeller (field experience)
4. Compare cavitation flow to minimum operating flow. If this value is within 10% of minimum operating flow:
   - Reselect pump if possible
   - Install minimum flow bypass
   - Consider parallel pump operation
5. Calculate liquid temperature rise at minimum operating flow. If value is greater than 5% of pumping temperature:
   - Calculate $NPSH_A$ based on vapor pressure at calculated pumping temperature. If $NPSH_A < NPSH_R$:
   - Install minimum flow bypass
   - Consider parallel pump operation

Fig 2.11.5 • Guidelines for selecting pumps free of hydraulic disturbances

Before proceeding, an important question regarding the typical pump performance curve needs to be asked. Why is the $NPSH_R$ curve not drawn to zero flow like the head curve? Based on the information presented in this course, you should be able to answer this question. Consider the following facts:

- The standard shop test fluid for all pumps is water
- The causes of vaporization at low flows

Hopefully your answer took the following form:

'Liquid disturbances can occur at low flows if the vapor pressure of the pumped liquid exceeds the surrounding pressure of the liquid.'

'Flow separation and/or liquid temperature rise which can occur at low flows will either reduce the surrounding pressure on a liquid or increase its vapor pressure.'

'Since the actual liquid characteristics are not known when the standard pump curve (tested on water) is drawn, the vendor stops the $NPSH_R$ curve where flow separation and liquid temperature rise can cause liquid disturbances.'

Therefore, trouble free operation to the left of this point is dependent on the pumped liquid and must be discussed with the pump vendor.

In conclusion, preventing liquid disturbances at the project design phase requires a thorough, accurate investigation of both the process and pump characteristics and some serious decisions on required action.

Justification of required action will be easier if the operating company looks beyond the project costs, and examines what the actual cost (that could occur through lost production) will be if a pump experiences hydraulic disturbances. Most operating companies have documented case histories of problem pumps that will provide facts relating to the 'total cost' of operating problem pumps for the life of a project (refer to Figure 2.11.6).

Determine the cost effectiveness of pump selection not only on the project (capital investment) costs, but on the cost to the operating company of unreliable pumps

Fig 2.11.6 • Justifying the selection of trouble free pumps

Best Practice 2.12

Require that suction specific speed for low NPSH required pumps is less than 10,000 (US units) to minimize the possibility of recirculation.

Low NPSH required for centrifugal pumps is attained by having a large impeller suction area and/or by using a double suction impeller (which has twice the suction area of a conventional impeller).

Recirculation is a hydraulic disturbance, which is the result of low fluid velocity into the impeller.

The increase of impeller suction area to reduce cavitation can produce recirculation if the increase in impeller suction area is too great.

Suction specific speed predicts the probability of recirculation.

Lessons Learned

Selecting a pump for low NPSH required values without assuring the suction specific speed is below 10,000 (US units) will result in a pump of low reliability and high maintenance.

The writer has personally experienced recirculation at the rated operating point with a double flow impeller that was not checked for suction specific speed (the suction specific speed exceeded 17,000). The recirculation forces resulted in the following damage to the pump:

- Impeller damage
- Shaft breakage
- Bearing failure
- Seal failure

Benchmarks

I have used this best practice since the mid 1980s, to ensure that pumps are not susceptible to recirculation at flows in the operating range of the pump (–50% of the pump best efficiency point). Pump MTBFs using this best practice have exceeded 80 months.

B.P. 2.12. Supporting Material

Liquid disturbances in centrifugal pumps are the major cause of low pump reliability. The pressures generated by cavitation can exceed 689,500 kpa (100,000 psi). Cavitation caused by many different factors is responsible for pump lost service time as a result of various pump component failures shown in Figure 2.12.1.

- Bearings
- Seals
- Impellers
- Shaft
- Wear rings

Fig 2.12.1 • Cavitation caused pump component failures

Proper pump selection guidelines require a centrifugal pump to be selected in the 'heart of the curve', and that sufficient NPSH AVAILABLE be present to avoid mechanical damage (see Figure 2.12.2). If the centrifugal pump operates only in the 'heart of the curve', also known as the equipment reliability operating envelope (EROE), costly field problems caused by liquid disturbances will be eliminated.

Maintaining a liquid inside a pump

Design objectives

All pumps are designed to increase the energy of a fluid while maintaining the fluid in its liquid state. Each centrifugal pump is designed to produce a specific amount of head at a specific flow rate based on a specified fluid density. Once the pump is designed, any reduction in the fluid density will result in a reduction in flow rate since the pump now requires greater head (energy) and can produce this value only at a reduced flow rate (see Figure 2.12.3).

One way to rapidly reduce the density of a fluid is to change its phase. If a liquid suddenly changes to a vapor, the impeller head required to meet the same process differential pressure requirements increases by a factor of 300 or more (see Figure 2.12.4). As a result, the pump will not be able to move

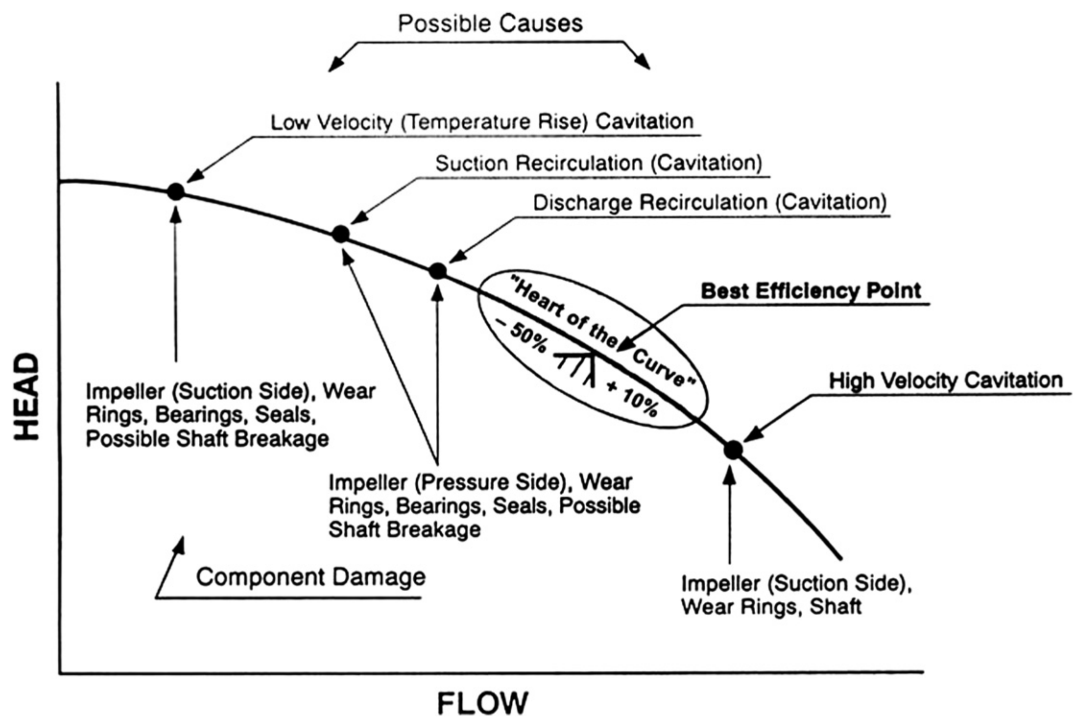


Fig 2.12.2 • Centrifugal pump component damage and causes as a function of operating point

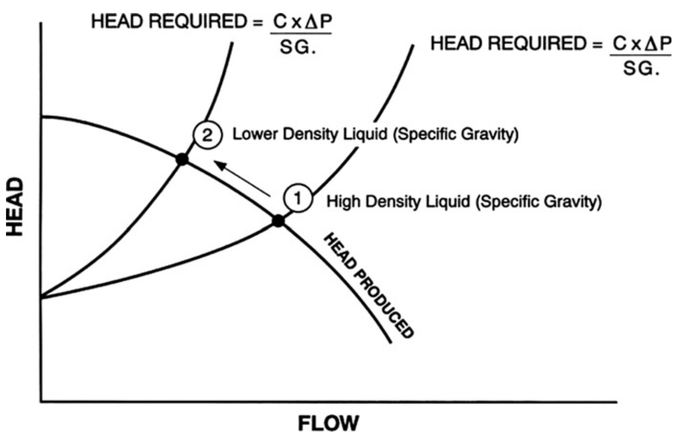


Fig 2.12.3 • Head required by lower density fluid

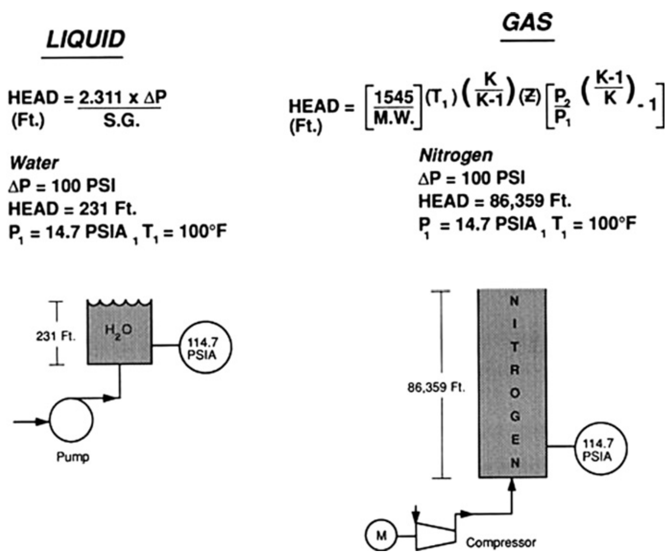


Fig 2.12.4 • Fluid head

the product since the maximum head produced by the pump will be much less than the head now required. The discharge pressure gauge will drop in pressure. This is commonly known as 'vapor lock' or 'loss of suction'.

Vapor pressure

How can a liquid change phase inside a pump? The vapor pressure for any fluid is that pressure at which the fluid changes from its liquid to vapor phase. Vapor pressure changes with fluid temperature. The vapor pressure for any fluid can be obtained from its Mollier diagram. For a given fluid temperature, the vapor pressure is the intersection of that temperature and the saturated liquid line. Figure 2.12.5 is the Mollier diagram for ethylene.

As can be seen from this figure, either a reduction in pressure (holding fluid temperature constant) or an increase in temperature (holding pressure constant) will cause a phase change. Another way of stating these facts is presented in Figure 2.12.6.

Before we leave this subject, let's examine how water can change phase (refer to Figure 2.12.7).

The most common cause of phase change for water is an increase in temperature at constant atmospheric pressure. When the vapor pressure exceeds the pressure surrounding the liquid, 101.3 kpa (14.7 psia), the liquid will change phase or 'boil'. However, water can also change phase if its pressure is

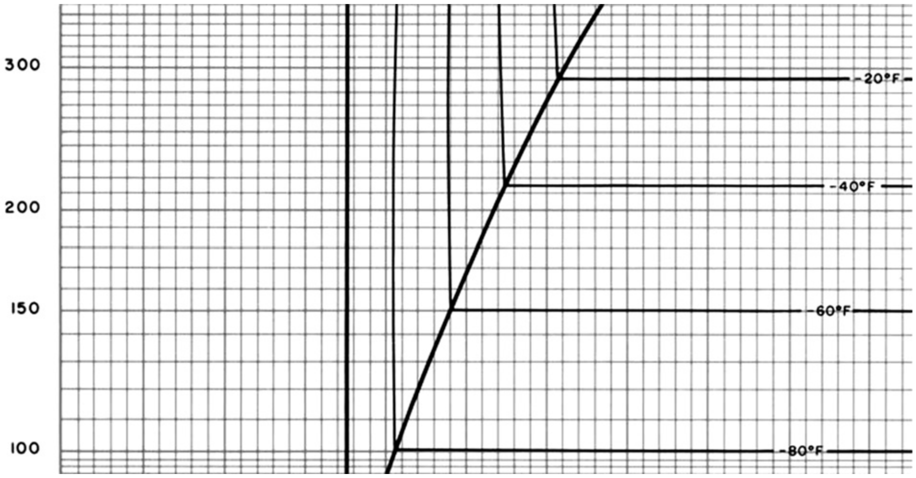


Fig 2.12.5 • Ethelyne Mollier diagram

A fluid will remain a *liquid* as long as its vapor pressure is less than the pressure acting on the liquid

Fig 2.12.6 • Maintaining a liquid

decreased while the temperature is held constant (see Figure 2.12.7). If the pressure of the water is less than its vapor pressure, water vapor will form. Therefore, any action inside a pump that either reduces the pressure of a liquid or significantly increases its temperature can cause the liquid to change to a vapor.

What can reduce the pressure of a liquid, or significantly increase its temperature inside a pump? To answer this question, the entire cross section of a pump must be examined. In Figure 2.12.8, the flow path from the pump suction flange to the impeller vane is plotted against pressure in the pump. It can be seen that the pressure decreases from the suction to the leading edge of the impeller vanes, then increases rapidly once the liquid is inside the impeller vanes – where vane tip speed (U) and

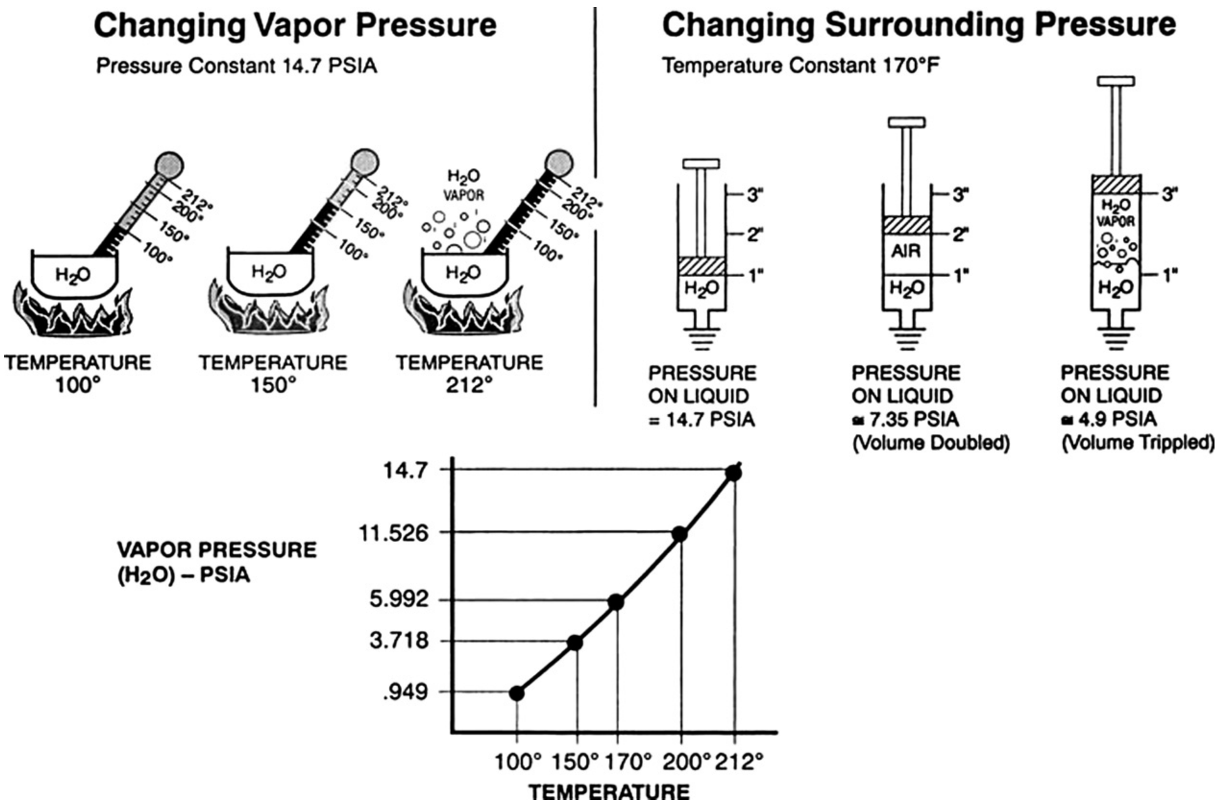


Fig 2.12.7 • Causes of liquid phase change – H_2O

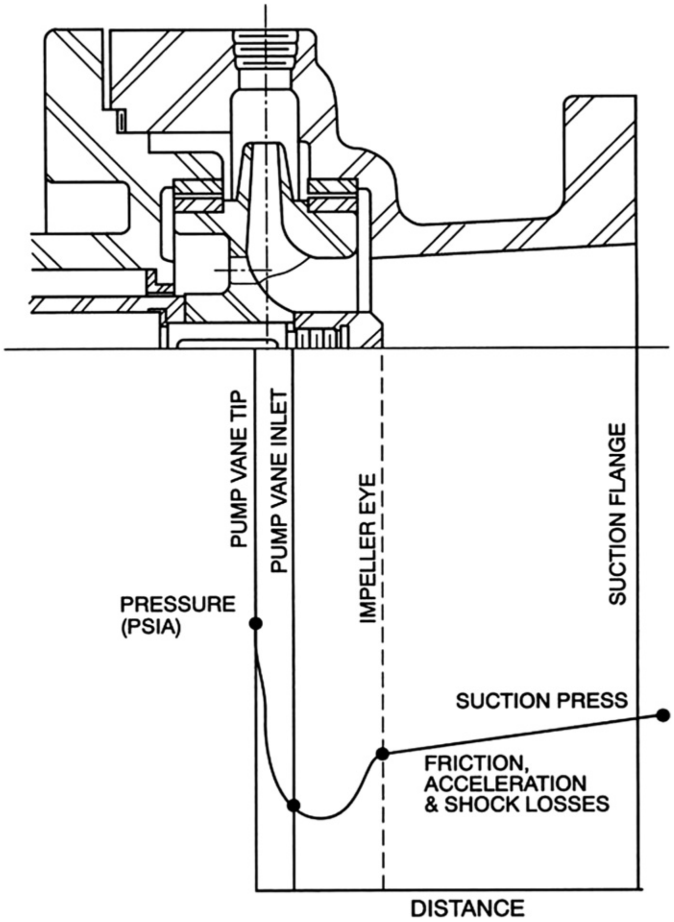


Fig 2.12.8 • Flange to vane entrance losses

liquid tangential velocity (VT) increase head (energy). The total pressure reduction from the suction flange to the impeller vanes is the result of:

- Friction losses in the flow passage
- Liquid acceleration
- Entry shock losses at the impeller vane tips

Therefore, each pump has a distinct pressure drop for any given flow, which is the result of pump case, inlet volute and impeller design. If the pressure drop from the suction flange to the impeller vane reduces the pressure below the liquid's vapor pressure, vapor will be formed at the impeller vanes.

There are also two (2) other causes of vapor formation within an impeller. Low flow velocities at any location within the impeller can cause separation of flow stream lines, and lead to low pressure areas (cells) inside the impeller. If the pressure within these cells is less than the fluid's vapor pressure, vapor will form (refer to Figures 2.12.9 and 2.12.10).

The curvature of the impeller vanes will always result in lower velocities on the pressure side of the vane (the side that cannot be readily observed). Therefore, vaporization caused by flow separation occurs on the pressure or 'backside' of the vane. This phenomenon is commonly known as recirculation.

Pump efficiency is low at low flows:

Liquid temperature rise increases by

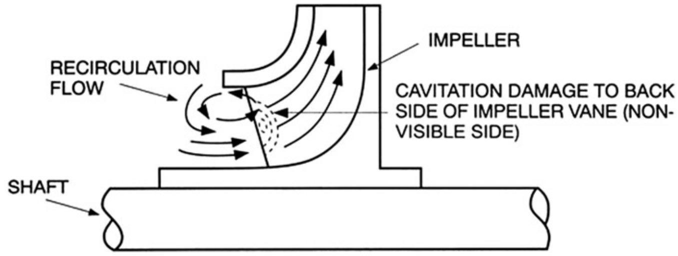


Fig 2.12.9 • Recirculation flow pattern in impeller at low flows

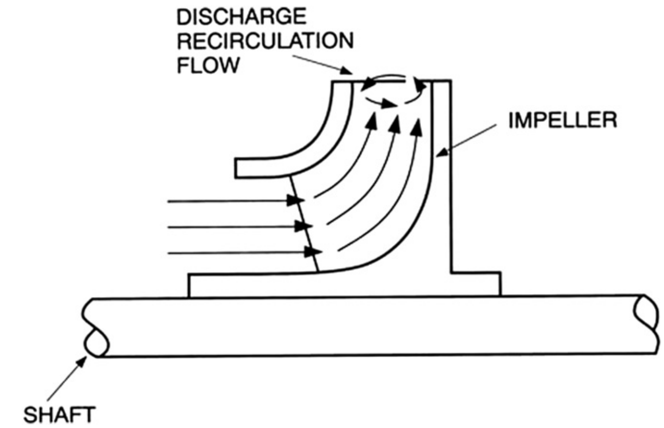


Fig 2.12.10 • Discharge recirculation flow pattern

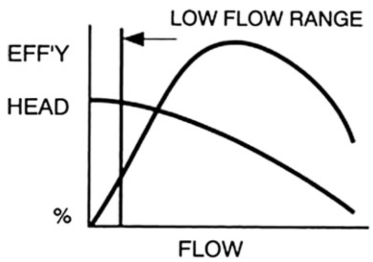


Fig 2.12.11 • Low flow temperature rise can cause vapor formation

$$\Delta T = \frac{\text{Pump head}}{337,100 \times C_P} \times \left[\frac{1}{\text{Pump efficiency}} - 1\right]$$

$$\times \left(\Delta T = \frac{\text{Pump head}}{778 \times C_P} \times \left[\frac{1}{\text{Pump efficiency}} - 1\right]\right)$$

Where: Pump head is calculated from data in $\frac{\text{m-kgf}}{\text{kgM}} \left(\frac{\text{ft-lb}_f}{\text{lb}_M}\right)$

Cp is specific heat of the fluid in $\frac{\text{KJ}}{{}^\circ\text{C} - \text{kg}} \left(\frac{\text{BTU}}{{}^\circ\text{F -LBmass}}\right)$

367,100 is conversion factor $\frac{\text{m-kgf}}{\text{kJ}}$

778 is conversion factor $\frac{\text{FT} - \text{LB}_F}{\text{BTU}}$

Pump efficiency is expressed as a decimal.

If the temperature rise increases the fluid's vapor pressure above the surrounding pressure, the fluid will vaporize.

- Internal inlet pressure losses
- Formation of low pressure cells at low flows
- Liquid temperature rise at low flows

Fig 2.12.12 • Causes of vaporization within a centrifugal pump

When operating at low flows, the efficiency of the impeller is significantly reduced, thus increasing the liquid temperature rise. As previously mentioned, the vapor pressure of any liquid increases with temperature. Increased temperature can cause the liquid to vaporize at the impeller vanes. Referring to Figure 2.12.12, it can be seen that low specific gravity liquids with high vapor pressures are the most susceptible to vaporization caused by low flow operation. Note that increased wear ring clearances can worsen this situation, since the higher temperature liquid will mix with the cooler liquid entering the impeller. Figure 2.12.12 summarizes the causes of vaporization within a centrifugal pump.

Causes of damage

In the above section the causes of vapor formation within a pump were described. In this section the causes of damage to pump components will be discussed.

Cavitation

Figure 2.12.13 presents the definition of cavitation.

Cavitation is the result of released energy when an increase of pressure surrounding the fluid causes the saturated vapor to change back to a liquid

Fig 2.12.13 • Cavitation definition

This indicates that vapor must be present before cavitation can take place. The sources of vapor formation were discussed and are summarized in Figure 2.12.12. Referring back to Figure 2.12.8, it can be seen that as soon as the liquid enters the impeller vane area, its energy and pressure rapidly increase. When its pressure liquid exceeds the vapor pressure, the vapor bubbles will collapse and cavitation will occur. Ways to prevent cavitation are shown in Figure 2.12.14. They will be discussed in more details later on in this chapter.

Cavitation is prevented by preventing vapor formation within a pump

Fig 2.12.14 • Preventing cavitation

The effects of fluids on component damage

The energy released during cavitation caused by inlet pressure losses, recirculation or low flow temperature rise varies as a function of the fluid type and the amount of vaporization. In Figure 2.12.15, we have drawn a generic Mollier diagram to

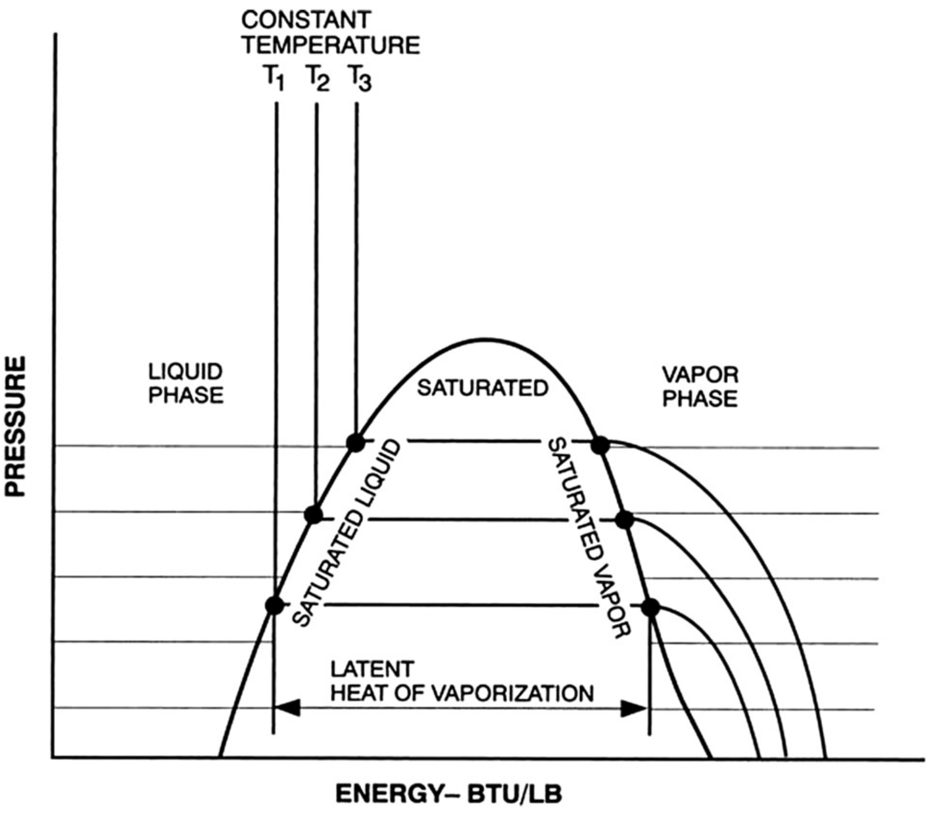


Fig 2.12.15 • Mollier diagram

show that the fluid type (latent heat of vaporization for a given temperature) and degree of vaporization determine the energy released when the vapor is recompressed to a liquid. Note that the abcissa is BTU/lb. When cavitation occurs, a given amount of energy (BTU/lb) will be transferred from the fluid to the impeller. Energy can also be expressed in ft-lb_F/lb_M by multiplying as follows:

$$\frac{\text{BLU}}{\text{lb}_M} \times \frac{778\ \text{ft} - \text{lb}_F}{\text{BLU}}$$

It can therefore be seen that during cavitation, the energy transferred by the fluid to the vanes can greatly exceed the head produced by the impeller.

In general, single component liquids produce the highest energy values during cavitation and are therefore the most damaging fluids. Hydrocarbon mixtures produce less energy, and have higher viscosities which reduce damage. Regardless of composition, all liquids produce high noise levels during cavitation, which typically sounds as if solid ‘rocks’ are passing through the pump. Always remember that there are different causes of vaporization which result in cavitation.

Preventing hydraulic disturbances

In the previous sections, the causes of liquid disturbances in centrifugal pumps were discussed. In this section, practical advice on how to prevent purchasing troublesome pumps in the design phase, and practical solutions on how to solve existing field problems caused by liquid disturbances, will be presented.

The project design phase

Action taken during the early stages of a project can significantly increase pump reliability and safety by eliminating all sources of vaporization within a pump. Sources of vaporization exist both in the process and in the pump. Before presenting solutions, a number of important concepts must be reviewed and presented.

Concepts

Specific speed

Specific speed is a non-dimensional value that is a function of pump speed, flow and head:

$$N_S = \frac{N\sqrt{Q}}{HD^{3/4}}$$

$$\begin{aligned} \text{Where :}\quad & N_S = \text{specific speed} \\ & N = \text{pump speed rpm} \\ & Q = \text{pump flow gpm} \\ & Hd = \text{pump produced head } \frac{\text{ft/lb}_F}{\text{lb}_M} \end{aligned}$$

Note: For double suction impellers, Q = Q/2

Specific speed is used extensively in both pump and compressor design, to optimize stage efficiency for a given value of flow and head required. In pump design, specific speed is used to optimize the following design parameters:

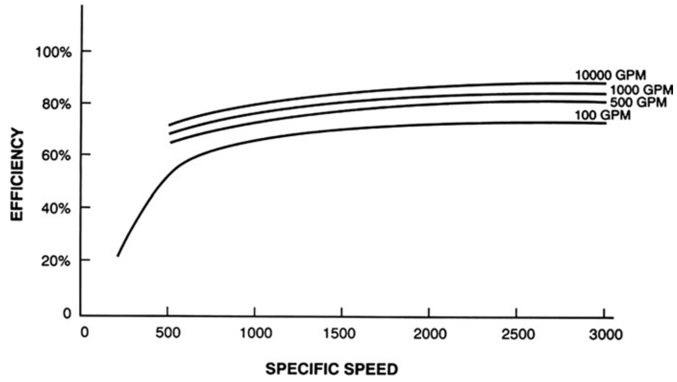


Fig 2.12.16 • Efficiency as a function of pump specific speed and flow

- Impeller discharge flow velocity
- Impeller tip speed
- Impeller inlet and discharge blade angles
- Discharge throat velocity

Figure 2.12.16 presents a plot of pump specific speed (US units) vs. efficiency for various flow rates. This is a generic chart, and its precise form will vary slightly from pump vendor to pump vendor, based on the specifics of pump design. It can be used for estimating purposes in determining pump efficiency to obtain pump required horsepower.

Net Positive Suction Head Available

$NPSH_A$ has been previously discussed. As we have learned, it must be greater than the $NPSH_R$ in order to prevent cavitation. Methods for determining $NPSH_A$ have been presented.

Net Positive Suction Head Required

Figure 2.12.17 shows the NPSH required within a typical centrifugal pump. It can be seen that this value is actually the pressure drop from the suction flange to the impeller vane inlet expressed in energy terms (head).

Perhaps now, we can truly understand why $NPSH_{AVAILABLE}$ must be $\geq NPSH_{REQUIRED}$. As we have learned,

$$\begin{aligned} &\textit{If}\ NPSH_A \geq NPSH_R \\ &\textit{Then},\ \text{the fluid will not vaporize} \\ &\textit{Therefore},\ \text{no vaporization, no cavitation, no damage} \end{aligned}$$

However, it must be remembered that $NPSH_A \geq NPSH_R$ is only one of the requirements that must be met to prevent vaporization. The following causes of vaporization must also be prevented:

- Low velocity stall
- Low flow temperature rise

Low flow temperature rise and fluid vaporization can be determined by the relationship shown in Figure 2.12.11, and the pumped fluid characteristics. However, the determination of low velocity stall or recirculation requires an understanding of the concept of suction specific speed.

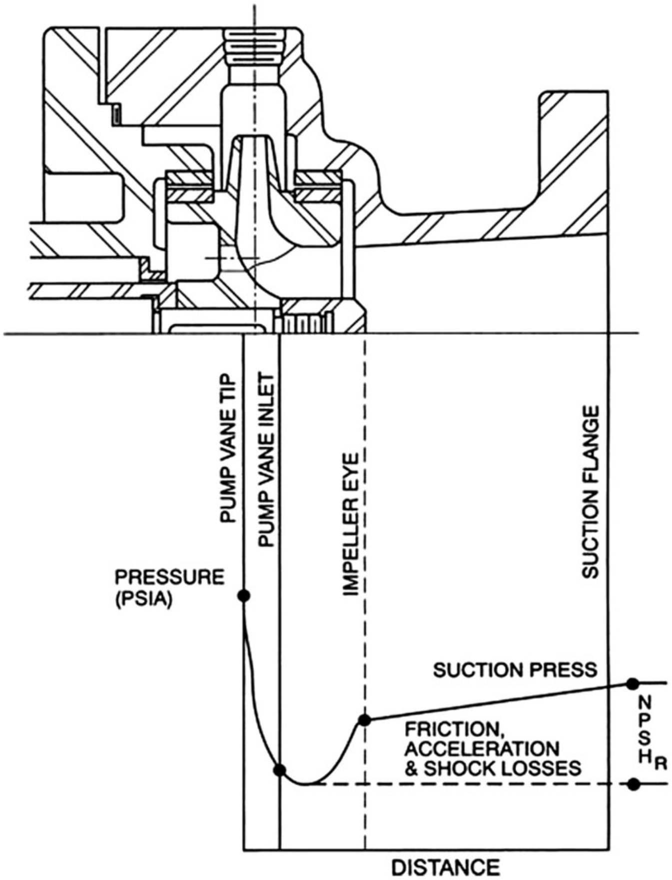


Fig 2.12.17 • Flange to vane entrance losses

Suction specific speed

N_{SS}, known as suction specific speed, is determined by the same equation used for specific speed N_S, but substitutes $NPSH_R$ for H (pump head). As the name implies, N_{SS} considers the inlet of the impeller and is related to the impeller inlet velocity. The relationship for N_{SS} is:

$$N_{SS} = \frac{N\sqrt{Q}}{(NPSH_R)^{3/4}}$$

Where: N = speed
Q = flow – GPM
$NPSH_R$ = Net Positive Suction Head Required in Ft.

As previously explained, $NPSH_R$ is related to the pressure drop from the inlet flange to the impeller. The higher the $NPSH_R$, the greater the pressure drop, and vice versa. From the above equation, we can show the relationships between $NPSH_R$, N_{SS}, inlet velocity, inlet pressure drop and the probability of flow separation. It is given in Figure 2.12.18.

Based on this information, it can be seen that flow separation will occur for high specific speeds resulting from low inlet velocity. The critical question the pump user needs answered is 'At what flow does the disturbance and resulting cavitation occur'? This is not easy to answer, because the unstable flow range is a function of the impeller inlet design as well as the inlet velocity. A general answer to this question is shown in Figure 2.12.19.

N_{SS}	$NPSH_R$	Inlet velocity	Inlet passage ΔP	Probability of flow separation
14,000 (High)	Low	Low	Low	High probability
8,000 (Low)	High	High	High	Low probability

Fig 2.12.18 • N_{SS} related to flow separation probability

The onset flow of recirculation increases with increasing suction specific speed

Fig 2.12.19 • Recirculation as a function of N_{SS}

Which can also be stated as; 'The higher the value of N_{SS}, the sooner the pump will cavitate when operating at flows below the BEP'. Therefore, before an acceptable value of N_{SS} can be determined, the process system and pumped liquid characteristics must be defined.

Defining the process system

Reviewing the proposed process system prior to the purchase of a pump, as previously discussed, is strongly recommended. Figure 2.12.20 presents a typical process system with various control alternatives (flow, level, pressure).

The approach that should be followed when purchasing a pump is to define its required operating range on the basis of the process system design and process requirements.

Once the operating range is defined, hydraulic calculations will determine the required flows, heads and $NPSH_{AVAILABLE}$. Care should be taken to define liquid composition and temperature as accurately as possible, since these factors will determine the vapor pressure which in turn defines the $NPSH_{AVAILABLE}$. Steps for defining the process system are summarized in Figure 2.12.21.

Once the process requirements are accurately defined, a pump can be selected that will meet these requirements without the risk of hydraulic disturbances.

Selecting a pump for hydraulic disturbance free service

The concepts used to prevent liquid disturbances, and the requirements for accurately defining the process system requirements, can then be used to select a pump free of hydraulic disturbances.

Based on previous discussions, there are three areas of concern to ensure trouble-free operation (see Figure 2.12.22).

Figure 2.12.23 shows a typical pump performance curve. The practical approach is to select a type of pump that will enable operation under all conditions if possible.

Figure 2.12.24 presents guidelines for selecting a pump free of hydraulic disturbances, which attempt to cover all situations; however, technical discussions with the pump vendor are encouraged whenever necessary.

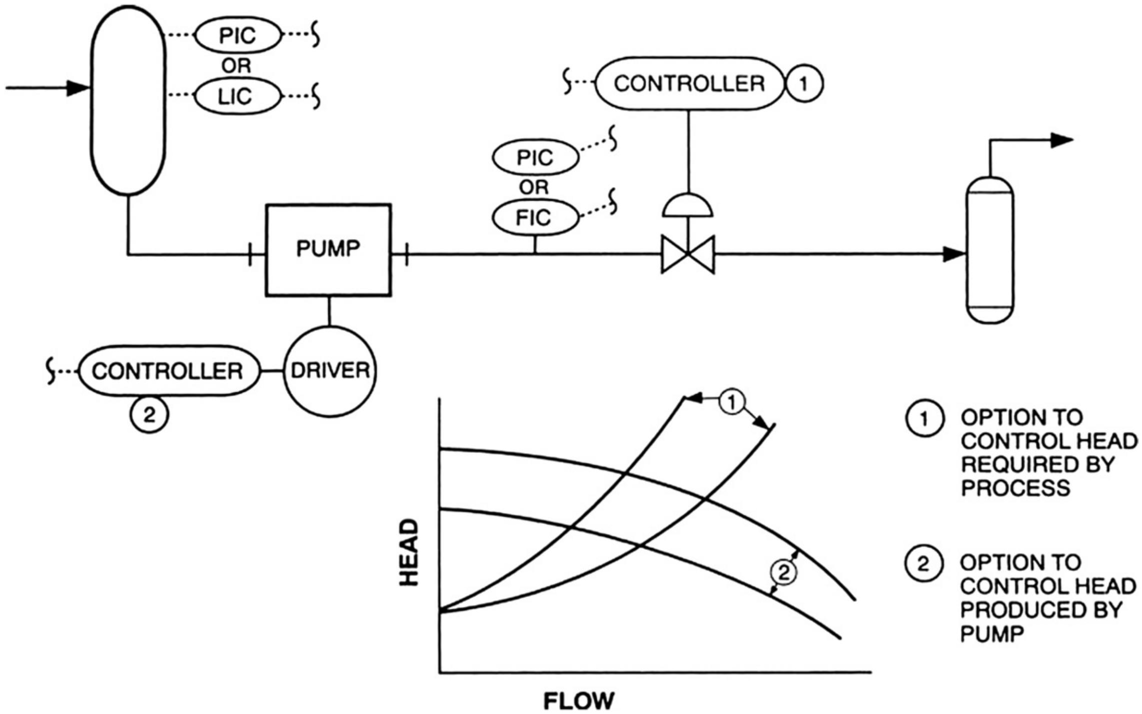


Fig 2.12.20 • Centrifugal pump control options

- Define operating range of application
- Accurately define liquid characteristics:
  - Vapor pressure
  - Pumping temperature
  - Viscosity
- Perform hydraulic calculations for all required flow rates to determine:
  - Head required
  - $NPSH_{AVAILABLE}$

Fig 2.12.21 • Preventing liquid disturbances by accurately defining process requirements

1. NPSH margin at maximum operating flow
2. Approximate recirculation margin at minimum operating flow
3. NPSH margin at minimum operating flow

Fig 2.12.22 • Hydraulic disturbances – areas of concern

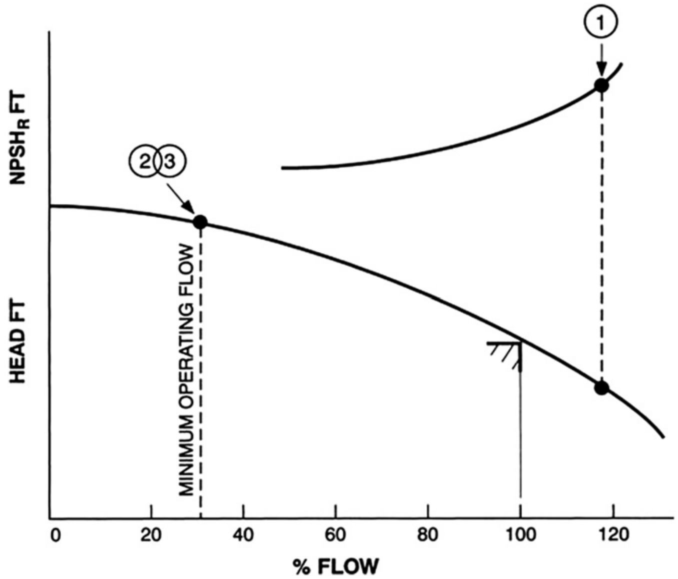


Fig 2.12.23 • Hydraulic disturbance – areas of concern

Before proceeding, an important question regarding the typical pump performance curve needs to be asked. Why is the $NPSH_R$ curve not drawn to zero flow like the head curve? Based on the information presented in this course, you should be able to answer this question. Consider the following facts:

- The standard shop test fluid for all pumps is water
- The causes of vaporization at low flows

Hopefully your answer took the following form:

'Liquid disturbances can occur at low flows if the vapor pressure of the pumped liquid exceeds the surrounding pressure of the liquid'.

'Flow separation and/or liquid temperature rise which can occur at low flows will either reduce the surrounding pressure on a liquid or increase its vapor pressure'.

Step Action

1. Confirm $NPSH_A \geq NPSH_R$ at maximum operating flow. If margin less than two (2) feet, require witnessed $NPSH_R$ test. If $NPSH_R > NPSH_A$:
   - Increase $NPSH_A$ by:
     - Increasing suction drum level
     - Decreasing pumping temperature
     - Decreasing suction line losses
   - Reselect pump (if possible)
   - Select canned pump
2. For the pump selected calculate N_{SS} based on pump BEP conditions. Note: if double suction first stage impeller, use 1/2 of BEP flow
3. If $N_{SS} > 8000$, contact pump vendor and require following data for actual pump fluid and conditions:
   - Predicted onset flow of cavitation caused by recirculation for actual fluid conditions
   - Reference list of proposed impeller (field experience)
4. Compare cavitation flow to minimum operating flow. If this value is within 10% of minimum operating flow:
   - Reselect pump if possible
   - Install minimum flow bypass
   - Consider parallel pump operation
5. Calculate liquid temperature rise at minimum operating flow. If value is greater than 5% of pumping temperature:
   - Calculate $NPSH_A$ based on vapor pressure at calculated pumping temperature. If $NPSH_A < NPSH_R$:
     - Install minimum flow bypass
     - Consider parallel pump operation

Fig 2.12.24 • Guidelines for selecting pumps free of hydraulic disturbances

'Since the actual liquid characteristics are not known when the standard pump curve (tested on water) is drawn, the vendor stops the $NPSH_R$ curve where flow separation and liquid temperature rise can cause liquid disturbances'.

Therefore, trouble free operation to the left of this point is dependent on the pumped liquid and must be discussed with the pump vendor.

In conclusion, preventing liquid disturbances in the project design phase requires a thorough, accurate investigation of both the process and pump characteristics and some serious decisions on required action.

Justification of required action will be easier if the operating company looks beyond the project costs and examines what the actual cost (lost in production) will be if a pump experiences hydraulic disturbances. Most operating companies have documented case histories of problem pumps that will provide proven facts relating to the 'total cost' of operating problem pumps for the life of a project (refer to Figure 2.12.25).

Determine the cost effectiveness of pump selection not only on the project (capital investment) costs, but on the cost to the operating company of unreliable pumps

Fig 2.12.25 • Justifying the selection of troublefree pumps

Field operation

The preceding section described requirements to ensure optimum pump reliability during the project design phase. This section will describe how to detect and correct hydraulic disturbances in the field.

Determining the potential for damage

Hydraulic disturbances are detected by monitoring the conditions noted in Figure 2.12.26.

- Loud noise – continuous or varying
- Suction and discharge pressure pulsations
- High values of overall vibration
- Drop in produced head of >3%
- High values of vane passing frequency in overall spectrum
- Pressure pulsations in inlet and discharge piping
- Possible high bearing temperatures

Fig 2.12.26 • Indicators of hydraulic disturbances

Once detected, the root cause of the disturbance must be established. As previously discussed, vaporization of liquid must be present for cavitation to occur, so the requirement is to determine the root cause of vaporization. As previously discussed, there are three (3) primary root causes of vaporization:

- Internal inlet pressure losses
- Formation of low pressure cells at low flows
- Liquid temperature rise at low flows

Confirmation of process conditions and specific tests are required to determine the root cause of the problem.

Determining the cause of hydraulic disturbances

Confirmation of stated value of $NPSH_A$

Refer to the Pump Data Sheet and/or the Hydraulic Calculation Sheet for that pump service to determine the stated $NPSH_A$. Proceed to check the $NPSH_A$ at field operating conditions. Using the relationship:

$$NPSH_A = \frac{2.311 \times (P_{suction-PSIA} - P_{vapor\ pressure-PSIA})}{\text{S.G. at pumping temperature}}$$

Substituting in the equation the following actual values:

- Pump suction pressure (down stream of suction strainer)
- Actual vapor pressure at measured pumping temperature
- Actual S.G. at measured pumping temperature

Compare calculated $NPSH_A$ to predicted $NPSH_A$. If actual $NPSH_A <$ predicted $NPSH_A$, modify operation if possible to

attain the predicted value. If this is not possible, the following alternatives exist:

- Operate pump at lower flow rate to reduce $NPSH_R$
- Modify pump to reduce $NPSH_R$ at operating flow rate

A summary of this discussion is presented in Figure 2.12.27.

- Obtain predicted value of $NPSH_A$
- Calculate $NPSH_A$ using actual conditions
- If $NPSH_A$ actual < $NPSH_A$ predicted:
  - Increase $NPSH_A$ if possible
  - Operate pump at lower flow rate (if cost effective)
  - Modify pump to reduce $NPSH_R$

Fig 2.12.27 • Confirmation of $NPSH_A$ and recommended action

Internal pressure loss test

The test outlined in Figure 2.12.28 will confirm if the liquid disturbance is caused by pump inlet pressure losses resulting from high liquid velocity.

- Close pump discharge control valve to reduce flow
- If pump noise significantly reduces and conditions noted in Figure 2.12.26 become stable, cause is confirmed

Fig 2.12.28 • Test to confirm high velocity cavitation

If high velocity cavitation is confirmed and the stated value of $NPSH_A$ is confirmed, possible solutions are presented in Figure 2.12.29.

- Increase $NPSH_A$ until quiet operation is achieved
- Reduce pump throughout if cost effective
- Operate two pumps in parallel
- Increase impeller eye area
- Impeller material change
- Purchase new pump with acceptable $NPSH_R$

Fig 2.12.29 • High velocity cavitation solutions

Modification of impeller eye area is not always possible and the pump vendor must be consulted to confirm it is possible and that satisfactory results have been achieved.

Before leaving this subject, mention of the inlet piping arrangement is required. Suggested inlet piping arrangements are presented in Figure 2.12.30.

Failure to conform to the guidelines presented here can lead to hydraulic disturbances caused by:

- Entrained vapor
- Additional internal ΔP caused by turbulence
- Liquid separation from impeller vanes.

- Piping runs directly vertically or horizontally into pump without high pockets that can cause vapor formation
- Minimum straight suction pipe runs of:
  - Three (3) pipe diameters – single suction
  - Five (5) pipe diameters – double suction
- Double suction pumps should have pipe elbows perpendicular to the pump shaft
- The 'belly' of an eccentric reducer should be in the bottom location

Fig 2.12.30 • Inlet piping arrangements to avoid hydraulic disturbances caused by process piping

Low flow hydraulic disturbance test

The test outlined in Figure 2.12.31 will determine if the liquid disturbance is caused by low flow circulation or temperature rise.

- Open pump discharge control valve or bypass to increase flow
- If pump noise significantly reduces and conditions noted in Figure 2.12.26 become stable, cause of either low flow recirculation or temperature rise cavitation is confirmed
- Calculate liquid flow temperature rise to confirm if recirculation is the root cause

Fig 2.12.31 • Test to confirm low flow hydraulic disturbances

- Increase pump flow rate, if possible, until quiet operation is achieved
- Modify inlet volute to increase impeller inlet velocity (if sufficient $NPSH_A$ exists)
- Install impeller with reduced eye area (assuming sufficient $NPSH_A$ exists)*
- Install minimum flow bypass to increase flow rate (to eliminate low flow temperature rise)
  *Note: wear ring modifications are required

Fig 2.12.32 • Solutions – low flow hydraulic disturbances

Possible solutions are presented in Figure 2.12.32.

Internal volute and/or impeller eye modifications are not always possible. The pump vendor must be consulted to confirm these modifications are acceptable and satisfactory results have been achieved.

Justification of proposed action plan

Regardless of the cause of hydraulic disturbances, the problem cannot be resolved without management endorsement of a cost effective action plan. As previously mentioned, all action plans must be justified by cost savings. In

the field, the largest loss of revenue is usually due to lost product revenue resulting from unplanned critical equipment downtime.

Regardless of the proposed action, be sure to show lost product revenue against capital investment for the problem solution.

Best Practice 2.13

Obtain references from pump vendors for pump bearings with DN numbers above 350,000 when ring oil lubrication is being offered to determine if oil circulation in the bearing housing is required.

DN is a measure of the surface speed of the shaft. It is the product of the shaft speed in revolutions per minute and the bore of the bearing in millimeters.

The higher the DN number, the greater the frictional heat that will be generated by the bearing.

DN values above 350,000 produce a significant amount of heat, and require oil circulation through a cooler to maintain the proper oil viscosity in the bearing.

Small oil consoles will solve the problem and result in a pump with a high MTBF. They can be easily retrofitted in the field if not originally supplied.

Lessons Learned

Pump bearings with a DN number above 350,000 are prone to low bearing MTBF, and should be supplied with a small dedicated pressurized lubrication system.

Many pump bearing low MTBFs (less than 12 months) have been resolved by installing small, dedicated, lube oil circulation systems whenever pump vendor experience dictates that the pump will operate at a DN above 350,000.

Benchmarks

This best practice has been used since the mid-1990s to increase pump high DN bearing MTBFs from less than 12 to greater than 100 months.

B.P. 2.13. Supporting Material

Bearings

Functions of bearings

Bearings provide support for the rotating element. They are required to carry radial and axial loads. The basic relationship for any bearing design is:

$$P = \frac{F}{A}$$

Where :

P = Pressure on the supporting oil film (hydrodynamic) or element (anti-friction) in P.S.I.

F = The total of all static and dynamic forces acting on the bearing in lb force

A = The bearing area in in^2

In many cases, a pump bearing may perform for years and suddenly fail. This abrupt change in performance characteristics is usually due to a change in the forces acting on it. Factors that can increase bearing load are shown in Figure 2.13.1.

In this section we will discuss the functions and various types of bearings used in pumps. There are three (3) basic types of bearings that are used in centrifugal pumps (refer to Figure 2.13.2).

Application guidelines

Centrifugal pumps in the process industry are normally fitted with bearings appropriate to the application and pump design (refer to Figure 2.13.3).

The L-10 rating life for anti-friction bearings is defined as the number of hours at rated bearing load that 90% of a group of identical bearings will complete or exceed (25,000 hours of operation) before the evidence of failure. Failure evidence is generally defined as a 100% increase in measured vibration.

API Standard 610 recommends applying pressurized hydrodynamic bearings when the product of the pump rated horsepower and the rated speed, in revolutions per minute, exceeds 2.7 million. Pressure lubrication systems may be integral or

- Increased process pipe forces and moments
- Foundation forces (‘soft’ foot, differential settlement)
- Fouling or plugging of impeller
- Misalignment
- Unbalance
- Rubs
- Improper assembly clearances
- Thermal expansion of components (loss of cooling medium, excessive operating temperature)
- Radial forces (single volute – off design operation)
- Poor piping layouts (causing unequal flow distribution to the pump)

Fig 2.13.1 • Sources of forces

- Anti-friction
- Hydrodynamic ring oil lubricated
- Hydrodynamic pressure lubricated

Fig 2.13.2 • Basic types of bearings

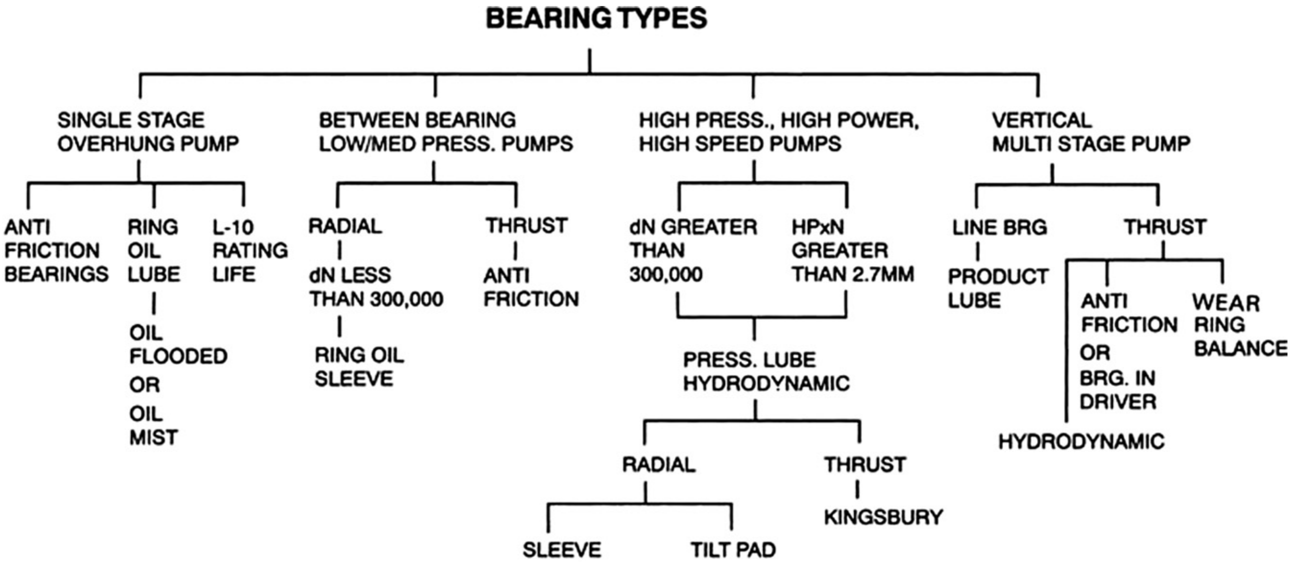


Fig 2.13.3 • Bearing application guidelines

separate, but should include as a minimum an oil pump, reservoir, filter, cooler, controls and instrumentation.

The D-N number is the product of bearing size (bore) in millimeters and the rated speed in revolutions per minute. It is used as a measure of generated frictional heat, and to determine the type of bearing to be used and type of lubrication required.

Used for all anti-friction bearings

$$L-10 = \frac{16700}{N} \bullet \left[\frac{C}{F}\right]^3$$

where: L-10 = hours of operation 90% of a group of identical bearings will complete or exceed
N = pump speed – RPM
C = the total force (kg [lbs]) required to fail the bearing after 1,000,000 revolutions (failure defined as 100% increase in measured vibration)
F = the total of all actual forces acting on the bearing (kg [lbs])

Fig 2.13.4 • L-10 Life

- Is a measure of the rotational speed of the anti-friction bearing elements
- D-N number = bearing bore (millimeters) × speed (RPM) Is used to determine bearing type and lubrication requirements

D-N range	bearing type	lubrication type
Below 100,000	anti-friction	sealed
100,000 – 300,000	anti-friction	regreasable
Below 300,000	anti-friction	oil lube (unpressurized)
Above 300,000	sleeve, multi-lobe or tilt pad	oil lube (pressurized)

Fig 2.13.5 • D-N Number

Figures 2.13.4 and 2.13.5 present the relationships for these two important factors.

Anti-friction bearings

Ball bearings

The anti-friction bearing most commonly used in centrifugal pumps for carrying radial and thrust loads is the ball bearing. The design of each type of bearing has its individual advantages for specific load carrying requirements (refer to Figure 2.13.6).

Roller bearings

Anti-friction roller bearings are capable of accepting pure radial loads, pure thrust loads or various combinations. They have been applied as crankshaft bearings in the power ends of some power pump designs. Refer to Figure 2.13.7. Choice of this type of bearing resides with the manufacture and is not usually an option for the user.

Lubrication

The purpose of anti-friction bearing lubrication is to increase bearing life, keep the balls or rollers separated, dissipate the heat generated in and conducted into the bearing, and to prevent corrosion. There are various methods of oil lubrication for ball and

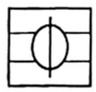
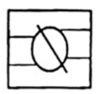
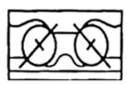
TYPE	LOAD CAPACITY	DESCRIPTION
• SINGLE-ROW DEEP GROOVE	EQUAL THRUST CAPACITY IN EACH DIRECTION. MODERATE TO HEAVY RADIAL LOADS	
• SINGLE-ROW ANGULAR CONTACT	HIGHER RADIAL LOAD THAN DEEP GROOVE. HEAVY THRUST LOAD IN ONE DIRECTION	
• DUPLEX, SINGLE-ROW BACK TO BACK (DB)	HEAVY RADIAL LOADS, COMBINED RADIAL/THRUST LOADS, REVERSE THRUST	

Fig 2.13.6 • Typical ball bearing load capacity guidelines

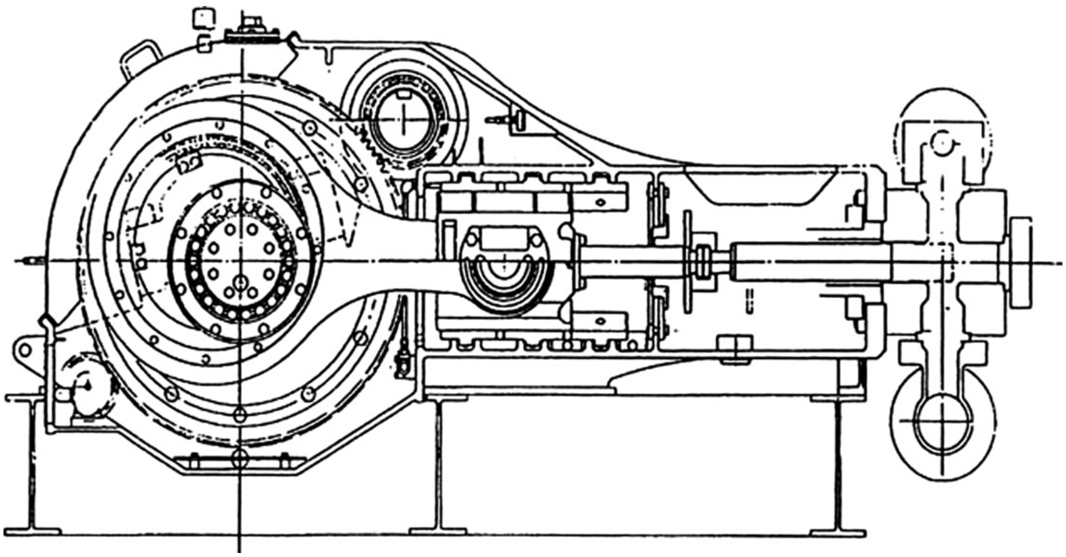
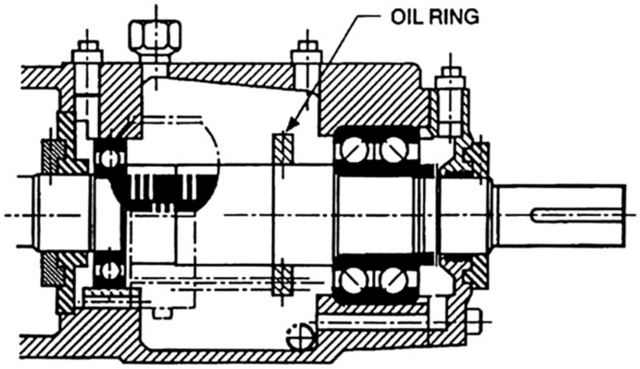
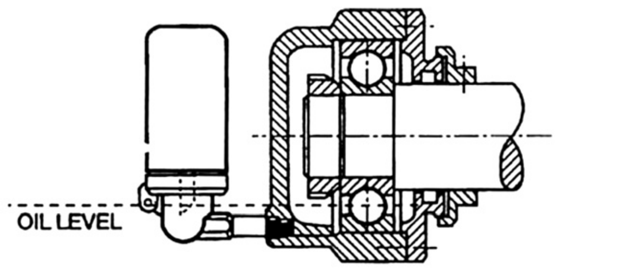


Fig 2.13.7 • Anti-friction roller bearings (Courtesy of Union Pump Co.)



• IN RING OIL LUBRICATION, OIL IS RAISED FROM A RESERVOIR BY MEANS OF A RING WHICH RIDES LOOSELY ON THE SHAFT AND ROTATES WITH THE JOURNAL
• OIL LEVEL IS AT CENTER OF LOWEST BEARING

Fig 2.13.8 • Ring oil lubrication



• IN FLOODED LUBRICATION, LUBRICATION IS ACHIEVED BY MAINTAINING AN OIL LEVEL IN THE RESERVOIR AT ABOUT THE CENTER OF THE LOWEST BEARING
• CONSTANT LEVEL OILER MAINTAINS LEVEL

Fig 2.13.9 • Flood type lubrication

roller bearings with the two most common methods being the ring oil and flooded method. Refer to Figures 2.13.8 and 2.13.9.

A word of caution – both types of lubrication usually incorporate constant level oilers to maintain oil level at a specified height in the bearing housing. The oil level in the constant level oiler is not the level of the oil in the bearing housing. The constant level oiler is a reservoir that provides oil to maintain a constant level in the bearing housing.

Proper oil level must be confirmed by visual inspection or by using a dip stick.

Oil mist lubrication is another acceptable method of lubricating antifriction bearings. It is gaining acceptance since it provides a controlled environment in the bearing housing in both operating and non-operating pumps. A mist generator console is often used to provide services to many pumps simultaneously. Use of oil mist lubrication should be investigated in environments where sand or salts exist (refer to Figure 2.13.10).

Features of oil mist system include:

- Mist generator provides mixture of air and atomized oil under pressure
- Reduced operating temperature resulting from airflow preventing excessive oil accumulation
- Oil consumption is low, controlled with instrumentation
- Entrance of grit and contaminants is prevented since air is under pressure
- Reliable continuous supply of lubricant is available
- Ring oil can be installed as back-up to mist system (called "wet sump system")

Fig 2.13.10 • Mist oil system features

Hydrodynamic bearings

The function of the hydrodynamic bearings is to continuously support the rotor with an oil film that is less than one (1)

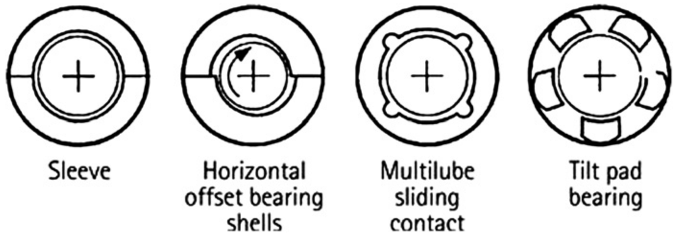


Fig 2.13.11 • Journal bearing types

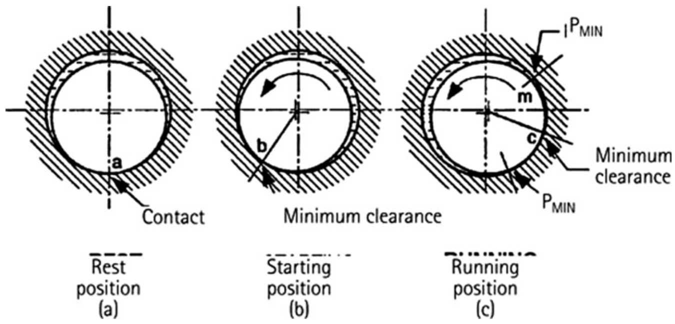


Fig 2.13.12 • Hydrodynamic bearing performance

thousandth of an inch. There are various types available — refer to Figure 2.13.11.

Sleeve bearings

Sleeve bearings are commonly used as radial bearings in centrifugal pumps. Lubrication is usually supplied by an external pressurized system, although some slow speed sleeve bearing applications (dN less than 400,000) can utilize ring oil lubrication.

Tilt pad bearing

Tilt pad journal bearings are sometimes used for high horsepower, high speed, centrifugal pumps where rotor stability may be a concern. They are always oil-pressurized bearings.

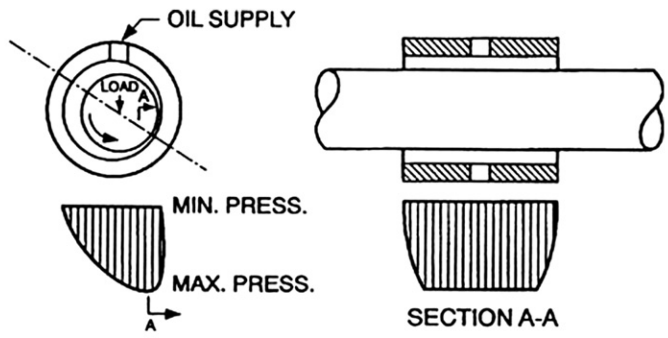
Multilobe bearing

This type of bearing has found some limited use in centrifugal pumps, but it can also be applied on high speed pumps where increased damping and stiffness is desired for lightly loaded bearings (most vendors will select tilt pad bearings).

Hydrodynamic bearing performance

When at rest, the journal settles down and rests at the bottom of the bearing (refer to Figure 2.13.12a). As the journal begins to rotate, it rolls up the left side of the bearing, moving the point of contact to the left. There is then a thin film of oil between the contact surfaces, and fluid friction takes over for metal to metal contact (refer to Figure 2.13.12b). The journal slides and begins to rotate, dragging more oil between the surfaces, forming a thicker film and raising the journal. As the speed of rotation increases, the oil drawn under the journal builds up pressure that forces the journal up and to the right in the direction of rotation shown, until a condition of equilibrium is reached resulting in a point of minimum clearance (refer to Figure 2.13.12c).

As the oil film builds up under the journal, the center of the journal moves and the location of minimum clearance is away from the load line. The pressure distribution across the bearing varies and the maximum unit pressure reaches a value about twice the average pressure on the projected area of the bearing (refer to Figure 2.13.13).



THE AVERAGE UNIT PRESSURE ON THE PROJECTED AREA (BEARING LENGTH X DIAMETER) IS GIVEN BY THE EQUATION

$$P = \frac{6 Z V d}{2C^2} K'$$

Z = ABSOLUTE VISCOSITY OF LUBRICANT, CP
V = JOURNAL SURFACE VELOCITY, FPS
d = JOURNAL DIAMETER, INCHES
C = DIAMETRAL CLEARANCE BETWEEN JOURNAL AND BEARING
K' = FACTOR DEPENDING ON BEARING CONSTRUCTION AND RATIO OF LENGTH TO JOURNAL DIAMETER

Fig 2.13.13 • Bearing pressure distribution

The permissible unit pressure (PSI) is a function of static and dynamic forces and the dimensions of the bearing (projected area). Industry guidelines for continuously loaded bearings range from 50 to 300 PSI. Pressures exceeding 300 PSI can potentially result in breakdown of the oil film and subsequent metal to metal contact.

Best Practice 2.14

Install mini-oil continuous flow consoles for high DN bearings (above 350,000 or when vendor 2 year field experience is not present for the proposed oil lubrication system (typically ring oil)).

Calculate the bearing DN as the product of the shaft speed in revolutions per minute and the bearing bore in millimeters.

If the calculated value exceeds 350,000, obtain vendor experience lists and contact references to ensure that the offered, non-continuous flow, bearing lubrication arrangement has resulted in MTBFs greater than 40 months.

If experience cannot be obtained, require continuous oil circulation.

Lessons Learned

Achieving optimum MTBFs with high DN pumps is not possible unless continuous lubrication is used.

It has been my experience that high DN pump bearings typically are 'bad actors' (MTBF less than 12 months) if they are not equipped with continuous oil flow consoles.

Benchmarks

This best practice has been since the 1980s to obtain high DN pump bearing MTBFs in excess of 100 months.

B.P. 2.14. Supporting Material

Please refer to B.P. 2.13 for reference material.

Best Practice 2.15

Remove temporary pump suction strainers after plant start-up whenever possible. If not possible, confirm proper strainer design and supply with a differential pressure gauge.

Check all temporary pump strainers for evidence of debris after one month of initial pump operation. If they show no evidence of debris, remove them!

If strainers show evidence of debris the following action is recommended:

- Ensure that the strainer design is rigid (perforated sheet metal with the appropriate mesh size over the perforated plate facing the direction of flow)
- Install a differential pressure gauge or differential pressure transmitter around the strainers and have the differential checked by operators on their daily rounds.

Lessons Learned

Many pump failures have been caused by plugged, poorly designed temporary suction strainers.

Many pump failures have resulted either from plugged strainers that have not been monitored, or strainers that were made of only mesh and sheet metal frame, that collapsed and entered the pump. In many cases the entire pump had to be overhauled, which exposed the process unit to shut down during the period of spare pump repair.

Benchmarks

This best practice has been used since the mid-1980s, resulting in pump applications with optimum safety and MTBF in excess of 80 months.

B.P. 2.15. Supporting Material

When reviewing the P&IDs for all pump applications, pay attention to the symbol TS and remember that the T stands for temporary.

It is recommended that each pump application be reviewed with the process licensor's engineers to ensure that the strainer can be removed without reducing the pump MTBF.

Also consult plant operators, if the pump application in question is presently operated in your plants, to ensure that the strainer can be removed.

If the strainer cannot be removed, it must be of permanent design (perforated plate with appropriate mesh size if required attached to the plate facing the direction of flow) and supplied with a differential pressure gauge or transmitter for daily monitoring.

Best Practice 2.16

Use automatic minimum flow bypass valves, for centrifugal pumps, whenever the pumped liquid specific gravity is less than 0.7 and/or for hot fluids.

Liquids having a specific gravity below 0.7 are known as high vapor pressure liquids, and they will become a vapor under atmospheric conditions.

These liquids also are usually close to their boiling point at field conditions, and therefore have a low net positive suction head requirement and are susceptible to cavitation and recirculation.

Automatic minimum flow bypass valves should be used in all applications involving such fluids, for each pump.

The minimum flow set point should be based on suction-specific speed calculations, and the pump vendor's recommendation for the exact fluid being pumped.

As a general rule, the minimum flow point should not be lower than the point at which the NPSH required curve ends at low flow on the pump curve.

Lessons Learned

The majority of EP&Cs and end users do not have specific requirements for the use of automatic minimum flow bypass valves. This fact leads to low MTBFs for pumps that can handle high vapor pressure liquids.

Many pumps which handle fluids with high vapor pressures (specific gravity less than 0.7) do not have automatic minimum flow bypass valves or manual bypass valves without any indication to operators in the control room of low flow. Pumps operating with high vapor pressure liquids at low flow conditions are prone to impeller and wear ring failure and seizure.

Benchmarks

This best practice has been used in pump applications since the 1990s, which has resulted in pumps of the highest safety and reliability and MTBFs exceeding 80 months.

B.P. 2.16. Supporting Material

Protecting the pump

Control and protection are interrelated. Since the objective of the control function is to maximize production throughput at minimum cost, it can sometimes cause the pump to operation in a region which can cause harmful effects (refer to Figure 2.16.1).

Centrifugal pump:

At or below minimum flow operation:

- Overheating
- High radial loads
- Internal recirculation damage
- End of curve flow
- Electric motor overload
- Turbine or engine overspeed
- Damage to pump/driver
- Positive displacement pump
- Closed discharge valve
- Damage caused by overpressure
- Electric motor overload damage
- Overspeed of direct acting pump
- Damage to pump

Fig 2.16.1 • Consider these factors for pump protection

Pump protection systems

Now that the potential effects of operating a pump outside a 'safe region' have been identified, an overview to illustrate the various methods available to protect each type of pump will be provided (refer to Figure 2.16.2).

Centrifugal pump:

- Minimum flow bypass:
- Manual
- Automatic by external electronic or pneumatic signal
- Motor overload protection
- Size driver for end of curve power
- Install orifice in discharge to increase system resistance and limit pump operation
- Flow or pressure limiting device
- Overspeed (variable speed driver)
- Governor
- Positive displacement pump
- Overpressure/overload
- Relief valve sized for full flow
- Overspeed protection (direct acting pump)

Fig 2.16.2 • Pump protection methods

Centrifugal pump protection systems

Since centrifugal pump flow varies inversely with pump head required, the pumps must be protected against low flow operation and high horsepower requirements (high pump flow). Minimum flow bypass systems and proper driver sizing and/or protection systems are therefore required.

Minimum flow bypass protection

Depending on the characteristics of the pumped fluid, continuous operation at low flow conditions can damage a centrifugal pump. Figure 2.16.3 shows the three (3) types of minimum flow bypass systems – manual, automatic external and automatic internal. As a rule of thumb, minimum flow protection should be provided for any pump application that can continuously operate below the pump vendor's minimum specified flow rate.

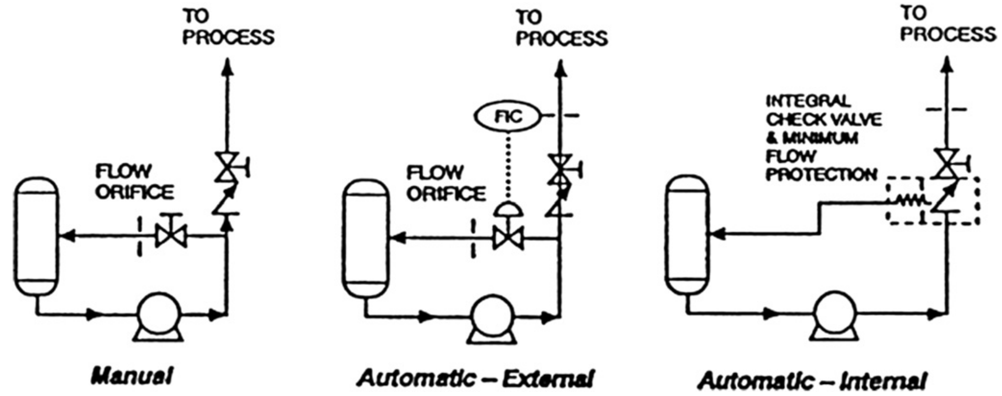


Fig 2.16.3 • Pump minimum flow protection

Pump discharge temperature rise calculations should be performed for all multistage, low specific gravity (less than 0.8), and low NPSH available applications, in order to determine if an automatic minimum flow bypass system is required.

Automatic minimum flow bypass systems can be either external or internal, but must reliably open at the specified set point. All instruments and valves used must be capable of repeatable operation. Failure of a minimum bypass system to function when it is needed can cause catastrophic pump damage.

External, minimum flow, bypass systems incorporate an external sensing transmitter, controller, conventional two way control valve and flow orifice assembly. An internal, minimum flow, bypass system incorporates all of the above components internal to the minimum flow bypass valve. It is recommended that internal, minimum flow, bypass valves be used only in clean pumpage systems, where the internals will not be affected by corrosion or blockage.

Motor overload protection

If the motor driver is not sized for horsepower capability at the end of the operating curve, the motor will trip on motor overload breaker protection unless some means is provided to prevent an excess horsepower requirement.

One alternative is to install an orifice to limit flow and therefore power requirements. If an orifice is installed, be aware that erosion and/or corrosion is possible, and require inspection at turnarounds.

Since horsepower is a function of flow and pressure, these variables can be limited to prevent motor overload.

Automatic motor overload protection can limit motor amps to a preset value by using a controller with input amps to control system level, flow or pressure as required.

Best Practice 2.17

Install flow switches in manual minimum flow bypass systems to prevent centrifugal pump damage from recirculation and vaporization.

If a pump is supplied with a manual minimum flow control valve, it should be equipped with a flow switch set to actuate in the control room at levels which are 10% before the minimum flow value is reached.

Lessons Learned

The failure to respond immediately to low centrifugal pump flow results in significant pump damage and exposure to safety issues.

I have observed that most centrifugal pumps that are subject to recirculation and/or vaporization are only supplied with manual minimum flow control valves which are not instrumented to detect low flow. Failure to open these valves during low flow conditions subjects the pump to low MTBFs caused by: bearing failure, mechanical seal failure, impeller and case wear ring seizure.

Benchmarks

This best practice has been used in projects since the 1990s, and has resulted in optimum pump safety and reliability (MTBFs exceeding 80 months).

B.P. 2.17. Supporting Material

Refer to B.P. 2.16 above for reference material.

Best Practice 2.18

Avoid the use of internal type minimum flow bypass valves whenever there is a possibility of debris or fouling in the pumped liquid.

Avoid the use of internal-type, minimum flow bypass valves with internal orifices and springs for upstream, oil field applications, boiler feed pumps and any other pumps that can come into contact with solid particles.

Use external, minimum flow, bypass valve arrangements that have a flow transmitter which will open a conventional control valve supplied with properly sized orifice(s).

External type minimum flow control valves allow operator monitoring of valve position and positively prevent valve sticking and failure to open.

Lessons Learned

Internal type minimum flow bypass valves have internal components that can affect their ability to open at the minimum flow set point. Since they have no external position monitoring device, catastrophic consequences can result in services that can contain solids in the pumped liquid.

Since 1990, several instances of complete, high pressure, boiler feed pump failure have been observed (where the pump was completely destroyed, and a new high pressure radial split pump became necessary) in systems that used internal-type minimum flow bypass valves.

Benchmarks

Since 1990 I have used the best practice of requiring external type minimum flow bypass control systems with multiple orifice chambers for all boiler feed water applications.

B.P. 2.18. Supporting Material

Minimum flow bypass protection

Depending on the characteristics of the pumped fluid, continuous operation under low flow conditions can produce centrifugal pump damage. Figure 2.18.1 shows the three (3) types of minimum flow bypass systems – manual, automatic external and automatic internal. As a rule of thumb, minimum flow protection should be provided for any pump application that can continuously operate below the pump vendor's minimum specified flow rate.

Pump discharge temperature rise calculations should be performed for all multistage, low specific gravity (less than 0.8), and low NPSH available applications to determine if an automatic minimum flow bypass system is required.

Automatic minimum flow bypass systems can be either external or internal type but must reliably open at the specified set point. All instruments and valves used must result in repeatable operation. Failure of a minimum bypass system to function when required can cause catastrophic pump damage.

External minimum flow bypass systems incorporate an external sensing transmitter, controller, conventional two way control valve and flow orifice assembly. An internal minimum flow bypass system incorporates all of the above components internal to the minimum flow bypass valve. It is recommended that internal minimum flow bypass valves be used only in clean pumpage systems where the internals will not be affected by corrosion or blockage.

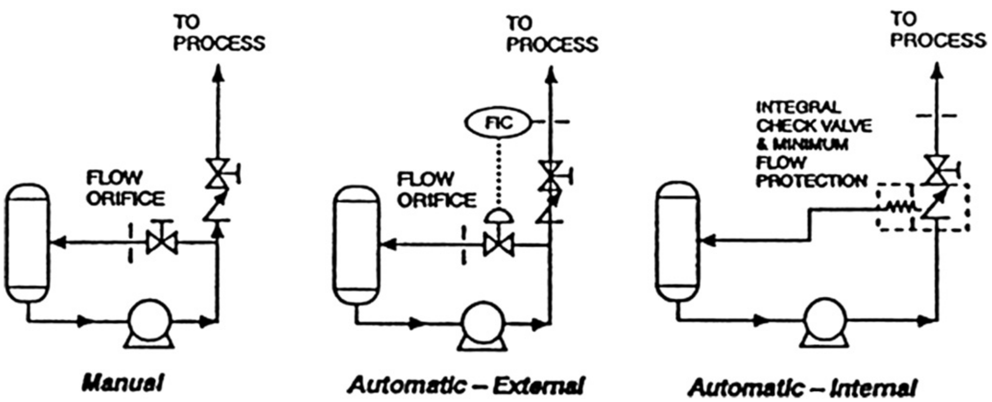


Fig 2.18.1 • Pump minimum flow protection

Best Practice 2.19

Monitor parallel centrifugal pump operation by the use of manual or automatic differential pipe temperature to prevent parallel pump failure.

Pumps that operate in parallel are subject to significant component damage if they are not individually protected against minimum flow – and usually they are not.

When pumps operate in parallel, the pump with the lowest head produced (worn pump and/or pump operating at the lowest speed) will be forced to reduced flow.

Typically, for cost saving reasons, the minimum flow protection for the pumps is installed in the pump discharge header and therefore does not provide individual pump protection.

As the flow to any centrifugal pump decreases (see supporting material below), the pump efficiency reduces and approaches 0% at zero flow. As a result, the pump differential temperature, which is close to zero degrees Celsius in the safe operating region of flow, will increase significantly as pump flow decreases.

Monitoring pump differential temperature is therefore a practical way of determining parallel pump internal condition and speed (if the pump driver is a variable speed type).

Lessons Learned

Centrifugal pumps operating in parallel are prone to damage from liquid vaporization when one or more pumps have deficient head/flow curves resulting from internal component wear.

It has been my experience that most parallel pump systems are not monitored for individual reduced pump flow, and are not provided with individual pump minimum flow protection. As a result, significant pump components can and will occur when one or more pumps have internal wear or operate at a reduced speed compared to the other pumps.

Benchmarks

This monitoring approach has been used since the mid-1990s to ensure parallel pump optimum safety and reliability (MTBFs greater than 80 months).

B. P. 2.19. Supporting Material

Figure 2.19.1 shows a typical centrifugal pump performance curve and the efficiency islands, which shows how the efficiency of any centrifugal pump varies with flow. As a worn or reduced speed pump is forced to lower flows by the more efficient pump in the parallel pump system, pump efficiency will reduce.

A reduction in pump efficiency will result in an increase in pump fluid differential temperature, which can be measured by contact thermometers on the inlet and discharge pipe. This increase in temperature will therefore indicate reduced pump flow and the condition and or reduced speed of the affected pump. Please refer to Figure 2.19.2 below.

Pump efficiency is low at low flows.

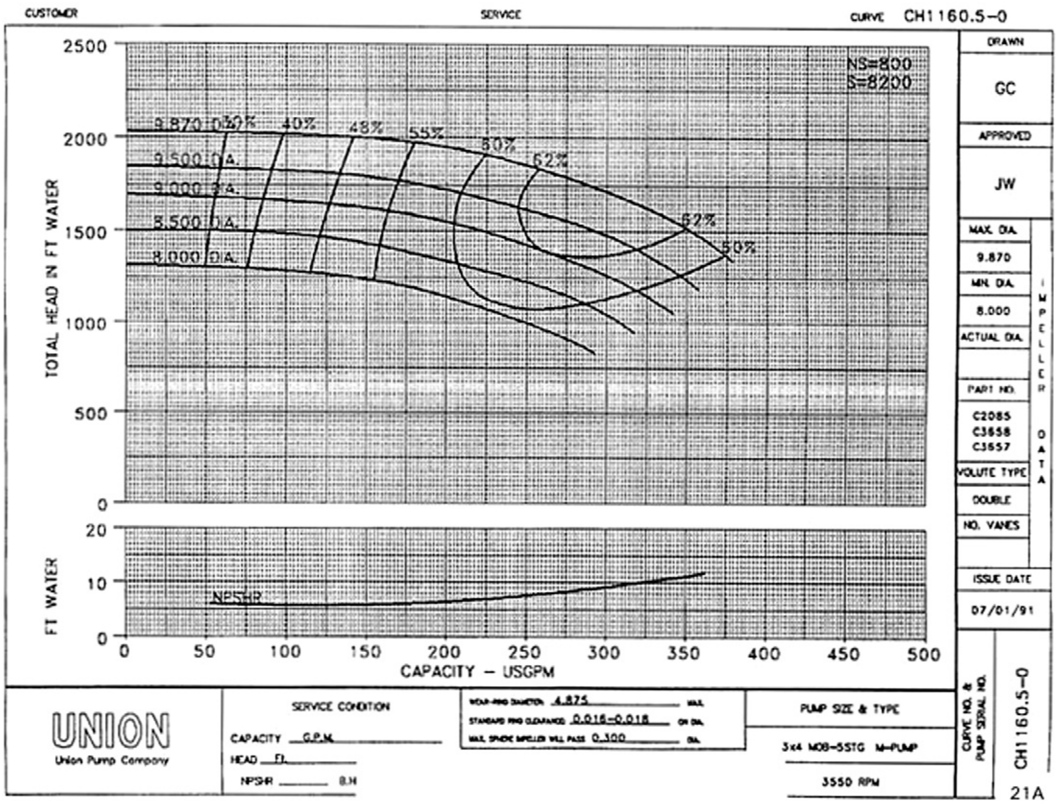


Fig 2.19.1 • A typical centrifugal pump performance curve (Courtesy of Union Pump Co.)

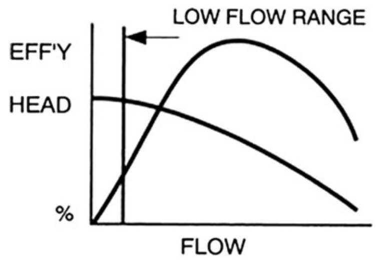


Fig 2.19.2 • Low flow temperature rise can cause vapor formation

Liquid temperature rise increases by:

$$\Delta T = \frac{\text{Pump head}}{337,100 \times C_P} \times \left[\frac{1}{\text{Pump efficiency}} - 1\right]$$

$$\left(\Delta T = \frac{\text{Pump head}}{778 \times C_P} \times \left[\frac{1}{\text{Pump efficiency}} - 1\right]\right)$$

Where: Pump head is calculated from data in $\frac{\text{m-kgf}}{\text{kgM}} \left(\frac{\text{ft-lb}_f}{\text{lb}_M}\right)$

Cp is specific heat of the fluid in $\frac{\text{KJ}}{^\circ\text{C} - \text{kg}} \left(\frac{\text{BTU}}{^\circ\text{F -LBmass}}\right)$

367,100 is conversion factor $\frac{\text{m-kgf}}{\text{kJ}}$

778 is conversion factor $\frac{\text{ft} - \text{lb}_F}{\text{BTU}}$

Pump efficiency is expressed as a decimal.

If the temperature rise increases the fluid's vapor pressure above the surrounding pressure, the fluid will vaporize.

Parallel pump operation

When flow requirements are variable, it may be more cost effective to operate two pumps in parallel, rather than to use a single, large pump. If demand drops off, one pump can be shut down, allowing the remaining pump to operate at or near its peak efficiency. Centrifugal pumps that operate most effectively in parallel are identical ones with steadily rising curves from rated flow to shutoff (refer to Figure 2.19.3). To obtain the overall curve for any pumps operating in parallel, add the flows from each pump at equal heads.

It is advisable to check the performance of pumps operating in parallel before attempting to use them in a process. Frequently, the main pump has been operated for much longer than the auxiliary pump and will have experienced wear, and as a result, will produce a different head vs. flow characteristic. Figure 2.19.4 shows the effect on the head-capacity characteristic of two pumps in parallel if one pump is deficient in head produced. Pumps in parallel operation should be protected by a minimum flow bypass system to prevent operation at shut off (zero flow).

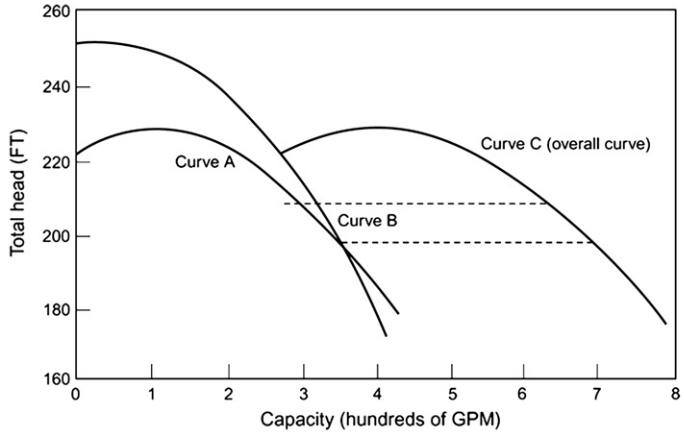


Fig 2.19.4 • Parallel pump operation — non identical pumps

*Note that overall curve C includes part of curve B from 0 flow to 250 gpm since pump A will not pump until its head is equal to the head of B pump.

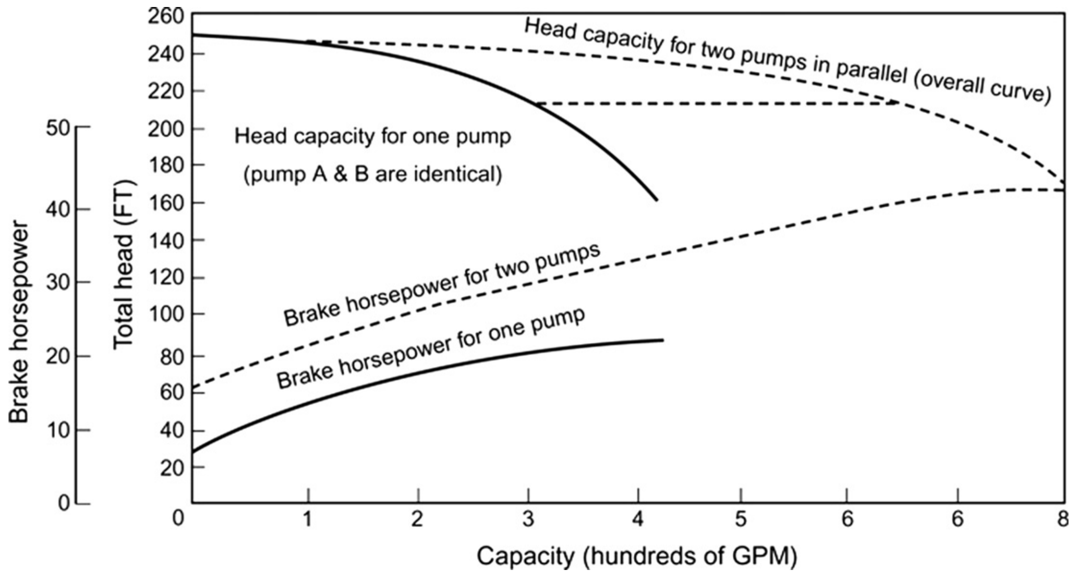


Fig 2.19.3 • Parallel pump operation — identical pumps

Best Practice 2.20

Check the pump flange and foundation forces whenever blinding and/or installing pumps to ensure that the external forces on the bearings are minimum.

Most centrifugal pumps use anti-friction bearings.

The life of any type of anti-friction bearing is inversely proportional to the cube of the forces on the bearing. As an example, if the forces on an anti-friction bearing double, the life of the bearing will decrease by eight times!

Checking the pipe forces and foundation forces (known as soft foot), whenever piping is installed or disturbed will ensure that the forces on bearings are acceptable and yield optimum bearing MTBF.

Lessons Learned

Failure to confirm proper pump flange and foundation forces will lead to low bearing MTBF, and possible safety issues in hot water or hydrocarbon applications.

Many low pump anti-friction MTBF issues (less than 6 months) have been solved by checking and correcting external pipe and foundation forces using the methods noted below.

Benchmarks

This best practice has been used since the mid-1980s to optimize pump safety and reliability. Bearing MTBFs have been increased from less than 6 months to greater than 100 months by implementing these pipe and foundation check methods.

B.P. 2.20. Supporting Material

Have you ever been called into your supervisor's office and questioned on how to properly install equipment or a component? Have you ever had the experience of installing a component (bearing), only to have it fail repeatedly over the following months and became a 'bad actor'? What is the problem? Your assembly procedures, the installation, the equipment, or the process?

The subject of this chapter is equipment pipe stress and soft foot. Without a doubt, these factors are prime contributors to 'bad actors'. They both are relatively easy to prove; however, they can be very difficult to correct. The purpose of this chapter is to present the reasons why excessive pipe stress and soft foot cause bad actors, how to prove these problems exist and the most cost-effective method to correct them.

Before we can understand *how* pipe stress and soft foot can cause equipment component failures, we must know *what* pipe stress and soft foot are! Figure 2.20.1 presents these facts.

Pipe stress and soft foot exert failure producing forces on the equipment casing from:

- Top, side or bottom flanges (pipe loads)
- Support feet (soft foot)

Fig 2.20.1 • Pipe stress and soft foot

Naturally, all equipment cases are designed to accommodate reasonable pipe loads and minimal load due to soft foot. However, Figure 2.20.2 shows what the equipment designer assumes in this regard.

- Under the limit on external pipe force (on assembly dwg)
- Under the limit on external pipe moments (on assembly dwg)
- All support feet are flat and in the same plane
- Foundation under all support feet has been leveled (shimmed if necessary with stainless steel shims)
- All external pipe(s) and support feet are properly connected

Fig 2.20.2 • External force design assumptions

How pipe stress and soft foot can cause component failure

Figures 2.20.3 and 2.20.4 show a typical, single stage, overhung pump and a general purpose steam turbine respectively. In both figures, the process pipes are not connected. If, in addition, both the pump and steam turbine were not coupled or bolted to their bases, what would cause the load (force) on the bearings?

Hopefully your answer was the rotor. Let's use the pump in the following discussion (Figure 2.20.3). However, everything discussed will apply equally to the steam turbine or any other type of equipment.

Please refer to Figure 2.20.5, which shows a typical anti-friction bearing that would be used for the pump radial bearing. Figure 2.20.6 presents the sources of the forces on any radial and/or thrust bearing regardless of the bearing type (anti-friction or sleeve).

For the pump in Figure 2.20.3, please describe the forces that the designer takes into account during the bearing *selection.* (Remember – anti-friction bearings are not custom designed.) Circle the forces in Figure 2.20.6 that should be considered during the bearing selection. Now please refer to Figure 2.20.7 which describes the relationship to determine the life of any anti-friction bearing.

As an exercise, let's determine the bearing life for the following cases: Case 1 – no excessive external load; Case 2 – additional soft foot load; and Case 3 – additional pipe stress load. We will note this information in Figure 2.20.8.

In Figure 2.20.8, Case 1 represents a bearing selected in accordance with industry standards that was installed correctly. That is, the predicted life of the bearing is in excess of 25,000 hours or 3 years' continuous operation.

Cases 2 and 3 are a different story! Observe the dramatic effect of additional forces on the equipment casing from either soft foot or piping forces.

If your manager or an operator had to complete this exercise, he probably would have listed the 'machinist' as the cause of

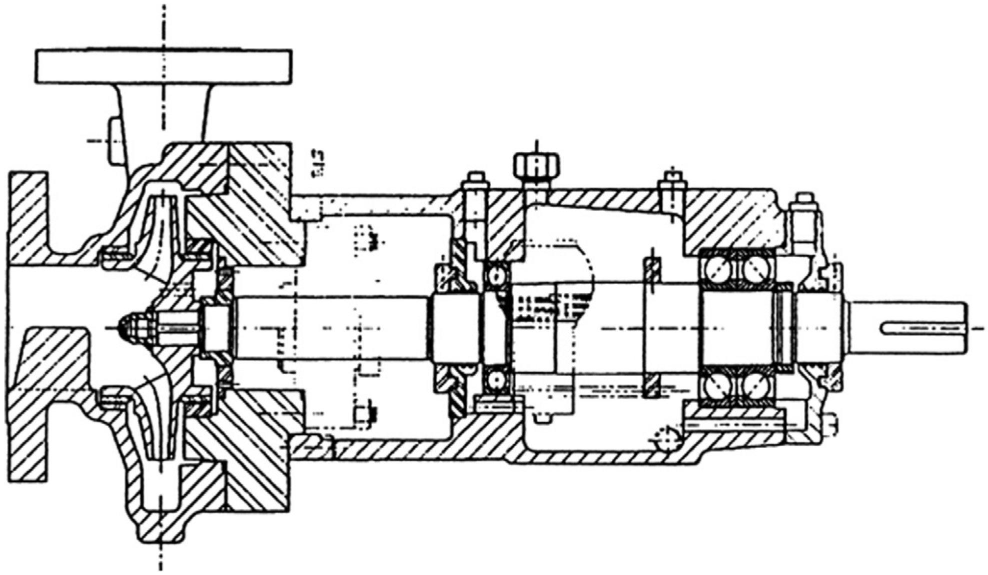


Fig 2.20.3 • Single stage overhung pump (Courtesy of Union Pump Co.)

failure! Hopefully, this exercise has clearly demonstrated why components, especially bearings, can suddenly fail for no apparent reason. These facts are presented in Figure 2.20.14.

Figure 2.20.10 has been modified to show the force path from excessive discharge flange loadings to the bearing bracket. Figures 2.20.11 and 2.20.12 show the orientation of external flange forces and moments that are referred to in Table 2.20.1.

Table 2.20.2 shows that the allowable forces and moments for most pumps are very low!

The root causes of excessive pipe stress and soft foot

Refer to Figure 2.20.13, the machinery environment. An associate of mine has a favorite quote regarding the machinery environment.

'Stand at the equipment unit and rotate yourself 360°. Everything that you see can and will affect the reliability of this piece of machinery'.

As shown in the last section of this chapter, the cause of component failure is the excessive forces exerted on the equipment from:

- The piping
- The foundation (soft foot)

What then are the possible causes? There are many. We will divide the possible causes into the following categories:

- Design
- Construction
- Plant conditions

The possible root causes are presented in Figures 2.20.14, 2.20.15 and 2.20.16.

There have been numerous examples, especially in the Middle East, of poorly prepared foundations and grouting. Careful attention must be paid to the quality of the water used, the type of grout and the method of grouting followed.

It is strongly recommended that an epoxy grout be used for all rotating equipment and a grouting procedure, approved by a reputable epoxy grout manufacturer, be utilized.

Fig 2.20.16 has an important message: 'If you suspect excessive pipe stress and/or soft foot forces, get out and thoroughly walk around the affected machine'.

Condition monitoring indications of excessive pipe stress and soft foot

At this point, we have covered the function of the two most important components in rotating equipment:

- Bearings
- Seals

The components which are most commonly affected by excessive piping and/or foundation (soft foot) forces are the bearings and couplings, although seal reliability can also be affected. Figures 2.20.17 and 2.20.18 present the parameters to monitor, as well as the limits for anti-friction and sleeve type radial bearings.

As previously stated, excessive pipe strain and soft foot exert forces beyond the design limits on equipment components. In the case of bearings, the forces will be transmitted from the source (pipe and/or foundation) through the casing, to the bearing housing, to the bearing. Figure 2.20.19 shows how to determine by condition monitoring if there is a pipe stress and/or soft foot problem.

Once the root causes of machinery component failure are suspected, they must be confirmed. Once confirmed, a cost-effective action plan must be developed that will ensure implementation at the earliest opportunity. This section presents this important information.

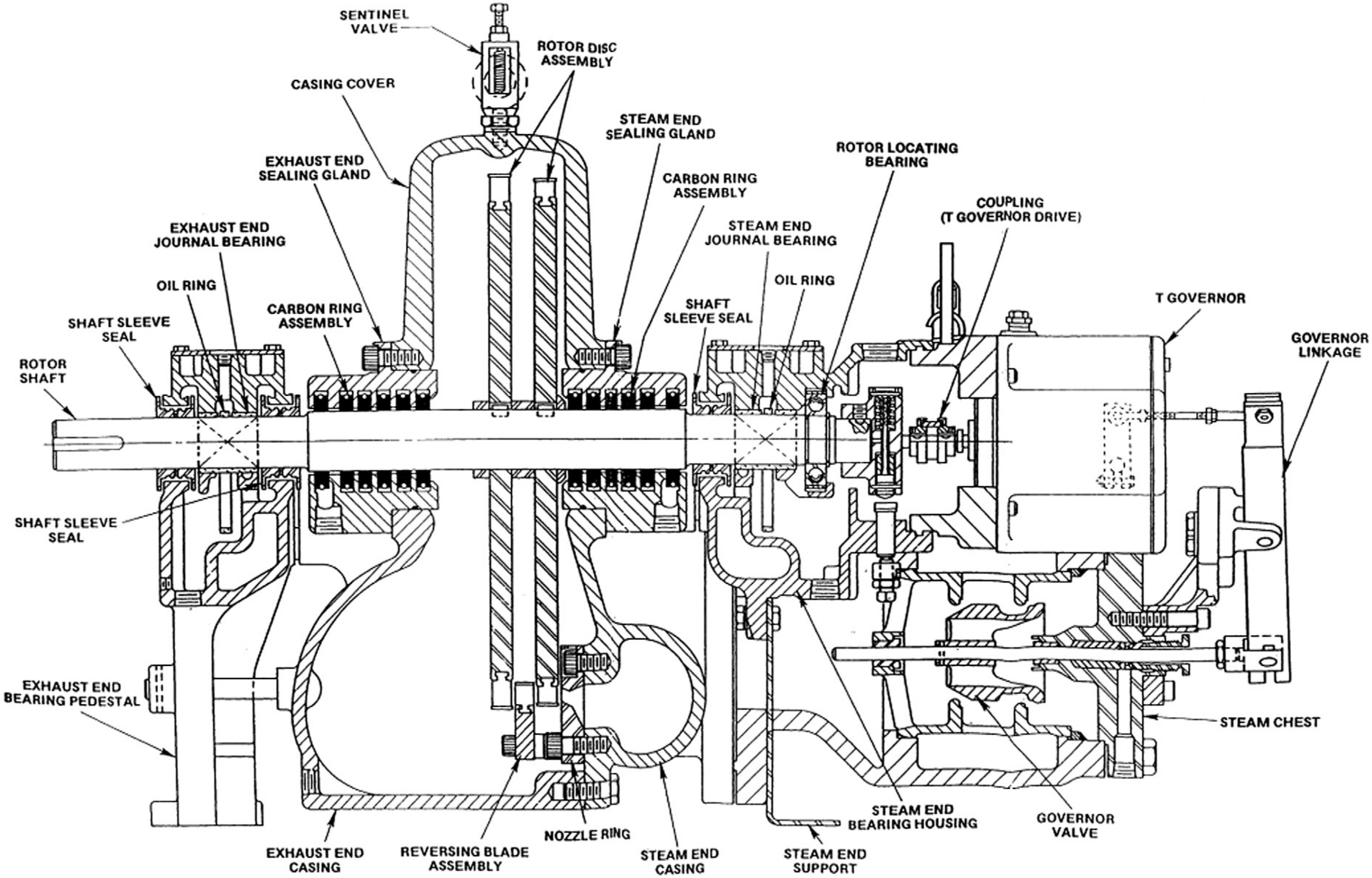


Fig 2.20.4 • General purpose steam turbine (Courtesy of Elliott Co.)

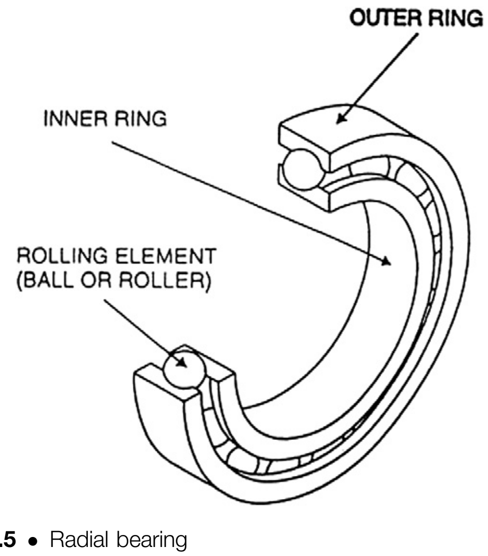


Fig 2.20.5 • Radial bearing

Case	1	2	3
N (speed)	3600 rpm	3600 rpm	3600 rpm
C (bearing dynamic load factor – lbs)	3000	3000	3000
F (total actual bearing load – lbs)	170	500	1000
Condition	As designed	Additional soft foot forces	Additional pipe load forces
L-10 life years	25,495 hours	1002 hours	125 hours
Cause of early failure	No early failure specified minimum life was exceeded	Excessive soft foot forces	Excessive pipe load forces

Note: use the relationship in Figure 2.20.7 to determine the bearing L-10 life.

Fig 2.20.8 • External loads on equipment example (use the relationship in Figure 2.20.7)

- Increased process pipe forces and moments
- Foundation forces ('soft' foot, differential settlement)
- Fouling or plugging of impeller
- Misalignment
- Unbalance
- Rubs
- Improper assembly clearances
- Thermal expansion of components (loss of cooling medium, excessive operating temperature)
- Radial forces (single volute – off design operation)
- Poor piping layouts (causing unequal flow distribution to the pump)

Fig 2.20.6 • Sources of forces

They exert forces in excess of design limits on the components:

- Casing
- Bearings
- Rotor
- Seals
- Wear rings
- The path of the forces is from the external force through the casing bearing bracket to the components

Fig 2.20.9 • How excessive pipe stress and soft foot forces cause equipment component failure

'B' or 'L' - 10 life is defined as the life in hours that 9 out of 10 randomly selected bearings would exceed in a specific application.

$$\text{'B' or 'L'} - 10\ \text{life} = \frac{16700}{N}\left[\frac{C}{F}\right]^3$$

Where: N = RPM
C = Load in lbs that will result in a bearing element life of 1,000,000 revolutions
F = Actual load in lbs

Fig 2.20.7 • 'B' or 'L' - 10 life

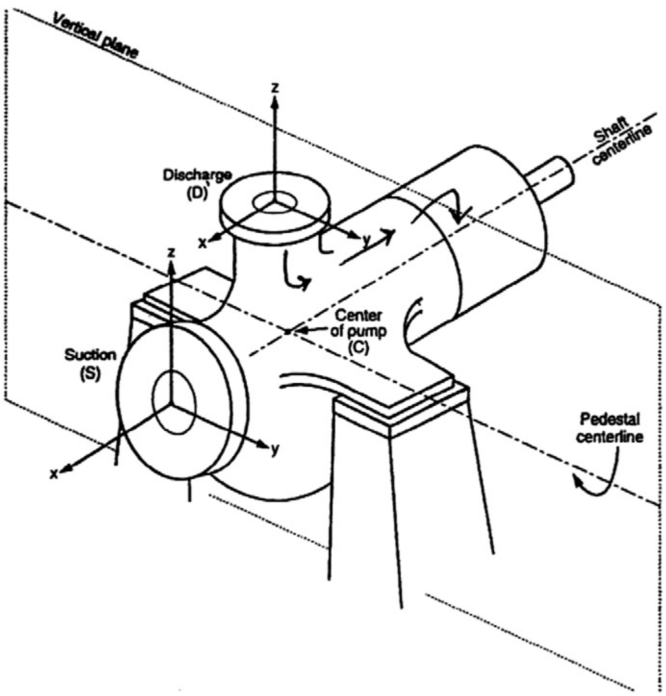


Fig 2.20.10 • Force path from excessive discharge flange loadings to the bearing bracket (Courtesy of API)

Confirming excessive pipe stress and/or foundation forces (soft foot)

In order to implement any action, we had better be sure our suspected root causes are correct. If they are not, we will always have a difficult time obtaining approval for any future recommendation.

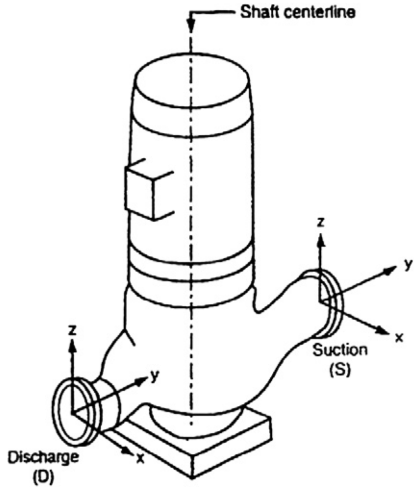


Fig 2.20.11 • Vertical in-line pump (Courtesy of API)

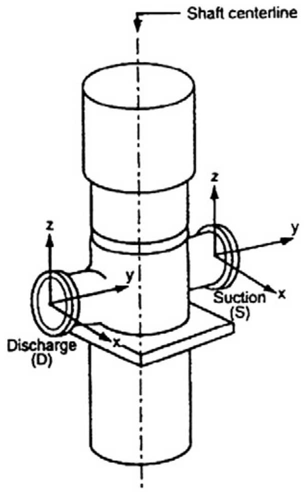


Fig 2.20.12 • Vertically suspended double-casing pump (Courtesy of API)

Table 2.20.1 Nozzle loadings (SI units)

Nominal size of flange (NPS)									
Force/moment	**2**	**3**	**4**	**6**	**8**	**10**	**12**	**14**	**16**
Each top nozzle									
FX	710	1070	1420	2490	3780	5340	6670	7120	8450
FY	580	890	1160	2050	3110	4450	5340	5780	6670
FZ	890	1330	1780	3110	4890	6670	8000	8900	10230
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850
Each side nozzle									
FX	710	1070	1420	2490	3780	5340	6670	7120	8450
FY	890	1330	1780	3110	4890	6670	8000	8900	10230
FZ	580	890	1160	2050	3110	4450	5340	5780	6670
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850
Each end nozzle									
FX	890	1330	1780	3110	4890	6670	8000	8900	10230
FY	710	1070	1420	2490	3780	5340	6670	7120	8450
FZ	580	890	1160	2050	3110	4450	5340	5780	6670
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850
Each nozzle									
MX	460	950	1330	2300	3530	5020	6100	6370	7320
MY	230	470	680	1180	1760	2440	2980	3120	3660
MZ	350	720	1000	1760	2580	3800	4610	4750	5420
MR	620	1280	1800	3130	4710	6750	8210	8540	9820

Note 1: F = force in Newtons; M = moment in Newton meters; R = resultant. See Figures 2.20.11 and 2.20.12 for orientation of nozzle loads (X, Y and Z).
Note 2: Coordinate system has been changed from API Standard 610, 7th edition, convention to ISO 1503 convention.
Note 3: Each value shown below indicates a range from minus that value to plus that value; for example 710 indicates a range from −710 to +710.

Table 2.20.2 Nozzle loadings (Courtesy of API) (US units)

	Nominal size of flange (NPS)								
Force/moment	**2**	**3**	**4**	**6**	**8**	**10**	**12**	**14**	**16**
Each top nozzle									
FX	160	240	320	560	850	1200	1500	1600	1900
FY	130	200	260	460	700	1000	1200	1300	1500
FZ	200	300	400	700	1100	1500	1800	2000	2300
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each side nozzle									
FX	160	240	320	560	850	1200	1500	1600	1900
FY	200	300	400	700	1100	1500	1800	2000	2300
FZ	130	200	260	460	700	1000	1200	1300	1500
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each end nozzle									
FX	200	300	400	700	1100	1500	1800	2000	2300
FY	160	240	320	560	850	1200	1500	1600	1900
FZ	130	200	260	460	700	1000	1200	1300	1500
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each nozzle									
MX	340	700	980	1700	2600	3700	4500	4700	5400
MY	170	350	500	870	1300	1800	2200	2300	2700
MZ	260	530	740	1300	1900	2800	3400	3500	4000
MR	460	950	1330	2310	3500	5000	6100	6300	7200

Note 1: F = force in pounds; M = movement in foot pounds; R = resultant. See Figures 2.20.11 and 2.20.12 for orientation of nozzle loads (X, Y and Z).
Note 2: Coordinate system has been changed from API Standard 610, 7th edition, convention to ISO 1503 convention.
Note 3: Each value shown below indicates a range from minus that value to plus that value; for example 160 indicates a range from −160 to +160.

- Process condition change
- Piping and foundation change
- 'Unit' (driven, driver, transmission, auxiliaries)
- Ambient conditions

Fig 2.20.13 • The rotating equipment environment

- Pipe stress calculation error
- Improper spring hanger selection
- Improper soil analysis assumptions
- Improper foundation design

Fig 2.20.14 • Possible causes for excessive pipe stress and/or soft foot (design)

- Using equipment as a 'pipe support'
- Improper installation of fixed spring supports
- Poor foundation and/or grout preparation
- Poor foundation and/or grout pour

Fig 2.20.15 • Possible causes for excessive pipe stress and/or soft foot (construction)

- Settling pipe support foundations
- Cracked grout and/or foundation (concrete)
- Locked spring supports
- Improperly installed new spring supports
- Shim corrosion under pipe supports
- Shim corrosion between equipment feet and baseplate
- Shims vibrating loose under pipe supports

Fig 2.20.16 • Possible root causes for excessive pipe stress and/or soft foot (plant conditions)

Bearing (anti-friction)

Parameter	Limits
Bearing housing vibration (peak)	10 mm/sec (.4 inch/sec)
Bearing housing temperature	85°C (185°F)
Lube oil viscosity	off spec 50%
Lube oil particle size	
non metallic	25 Microns
metallic	any magnetic particle in the sump
Lube oil water content	below 200 ppm

Fig 2.20.17 • Condition monitoring parameters and their alarm limits

Bearing (hydrodynamic)

Parameter	Limits
Radial vibration (peak to peak)	60 microns (2.5 mils)
Bearing pad temperature	108°C (220°F)
Radial shaft position*	> 30° change and/or 30% position change
Lube oil supply temperature	60°C (140°F)
Lube oil drain temperature	90°C (190°F)
Lube oil viscosity	off spec 50%
Lube oil particle size	> 25 microns
Lube oil water content	below 200 ppm

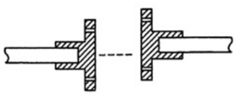
*except for gearboxes where greater values are normal from unloaded to loaded

Fig 2.20.18 • Condition monitoring parameters and their alarm limits

- More than one (1) bearing failure, rotor breakage, or coupling failure per year
- Unexplained high vibration (usually indicating misalignment)
- Unexplained high bearing housing temperature
- Pipe supports close to equipment not vibrating, equipment is vibrating

Fig 2.20.19 • Condition monitoring indications of excessive pipe forces and/or soft foot

Shaft alignment



Preliminary considerations

1. Obtain thermal shafts and machine growth calc's to establish "cold offsets"
2. Coupling hubs installed in accordance with OEM's procedures
3. Set proper B.S.E. dimension
4. Test for "soft foot": 0.05 mm (0.002") maximum allowable differential rise
5. Use only stainless shims – minimize number of shims
6. Shims must straddle hold down bolts

Fig 2.20.20 • Shaft alignment – preliminary considerations

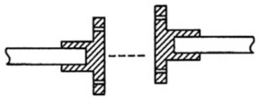
Figures 2.20.21 and 2.20.22 present the guidelines for confirmation of excessive piping stress and/or soft foot.

Correcting excessive pipe stress and foundation forces on equipment

Of all the different problems with rotating equipment, the resolution of excessive pipe stress is the most difficult. Why? Correction can involve extensive work that will require a significant amount of safety permits and may even require process unit shutdown. Figure 2.20.22 shows the suggested excessive pipe stress solution procedure. It is naturally arranged in a cost-effective order (simplest, least costly action first).

Correcting soft foot problems can be extremely simple if equipment support feet are not level to the foundation. In this case, stainless steel shims can be added. However, in some cases, baseplates can become distorted and/or the foundation can experience differential settlement over a period of

Shaft alignment



Alignment change limits when connecting piping

1. Mount dial indicators independent of machinery horizontally and vertically
2. Reading on coupling flanges
3. Zero out indicators
4. Tighten using specified sequence and torque values
5. Maximum shaft movement = 0.05 mm (0.002")
6. Dowell if required by OEM

Fig 2.20.21 • Shaft alignment – alignment change limits when connecting piping

- Confirm excessive pipe stress[1] (refer to Figure 2.12.5). Also confirm pipe bolting can be removed without a 'come along'
- Walk piping system and confirm proper installation per piping isometrics
- Proper pipe support shims
- Spring supports free to move
- No obvious pipe misalignment
- Correct excessive pipe stress by:[2]
  - Attempting rebolting at the next flange
  - Using 'Dutchman' with flexitallic gaskets (each side)
  - Heating of pipe for alignment
  - Pipe modification at next T&I

Notes: [1]Work permits required
[2]All items in III must be confirmed correct per Figure 2.12.5.

Fig 2.20.22 • Suggested excess pipe stress solution procedure

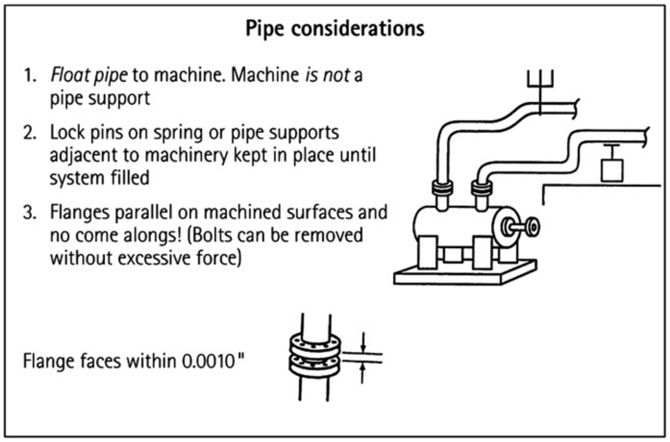


Fig 2.20.23 • Piping considerations

time. This usually requires a complete new foundation be designed and installed at the next T&I. A short-term fix can be to install stainless steel shims to temporarily correct the problem.

Clearly stating impact of problem on plant profit.
Prepare a brief statement of:

- The problem
- Action plan and confirmation of success (past experience)
- Cost of failure to date
- Cost of solution
- The impact on plant profit (loss)

Be confident!
Be professional!
Provide timely updates and final report on completion.

Fig 2.20.24 • Obtain and maintain management support by …

Implementation of the action plan

Correction of pipe stress problems is usually the most difficult problem to obtain action plan implementation for. Why? It is costly, exposes the plant to possible safety problems and can result in a process unit shutdown. Usually, it should be planned for a T&I. I have found that the guidelines in Figure 2.20.23 provide the best probability of implementation.

Best Practice 2.21

Pre-commissioning – ensure that every centrifugal pump operates in the equipment reliability operating envelope (EROE) and change impeller diameter if required.

The hydraulic calculations used to determine the pump head required for centrifugal pumps will only approximate the field conditions, and can be conservative, which will result in lower field head required than noted on the pump data sheet.

Lower pump head required can force centrifugal pumps to operate at greater flow than the design point.

Always confirm that new pumps are operating within the EROE, as previously noted in this chapter, and take corrective action as noted below if required.

Most centrifugal pumps drivers are sized for +10% power and can be overloaded if the pump flow is greater than the design flow.

The most cost effective solution to prevent driver overload is to reduce (cut or trim) the pump impeller diameter, to arrive at the desired pump flow at the actual field process head required conditions.

Lessons Learned

Many new pumps are selected with impeller diameters that are too large for field operation parameters. This can result in driver overload and possible cavitation.

Many new plants commissioned do not confirm that centrifugal pumps are operating in the EROE, which results in pump overload and cavitation. This lack of action has resulted in low MTBFs and frequent tripping of motor driven pumps.

Benchmarks

This best practice has been followed during pre-commissioning of all new centrifugal pump installations since the mid-1980s to optimize centrifugal pump unit safety and reliability. This best practice has ensured pump unit MTBFs in excess of 80 months.

B.P. 2.21. Supporting Material

The affinity laws

Let us examine the effect of speed and/or impeller diameter change on the performance of centrifugal pumps.

The affinity laws – or fan laws, as they are sometimes referred to – play an important role in determining centrifugal pump performance when operating conditions change. They are also used for scale-up purposes when performance parameters

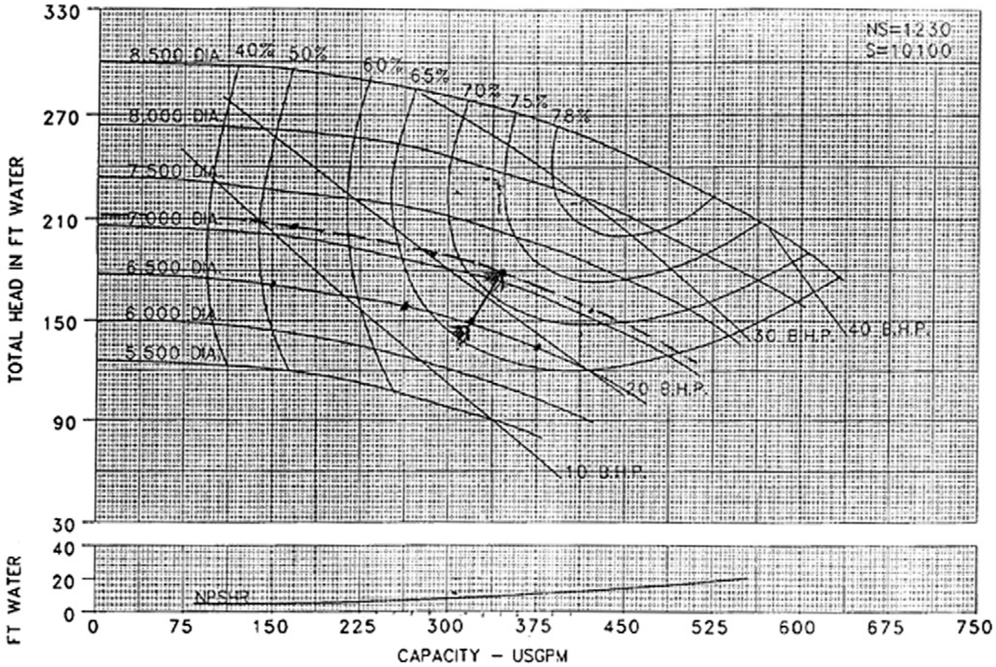
$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} = \frac{D_1}{D_2}$$

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 = \left(\frac{D_1}{D_2}\right)^2$$

$$\frac{HP_1}{HP_1} = \left(\frac{N_1}{N_2}\right)^3 = \left(\frac{D_1}{D_2}\right)^3$$

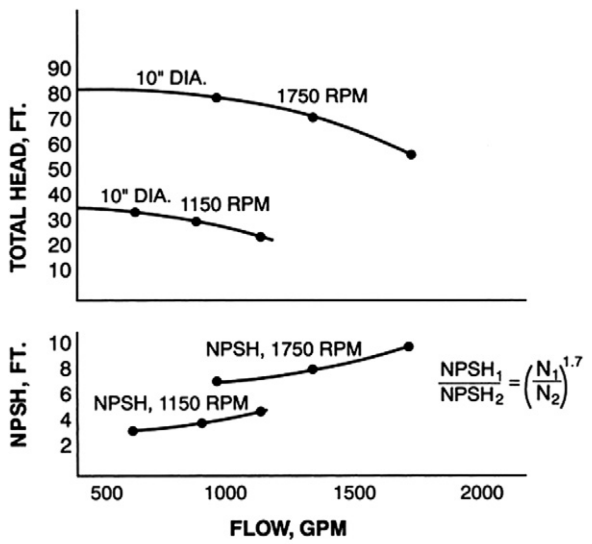
Where: Q = Flow, in m^3/hr or GPM
N = Speed, RPM
H = Head in m-kg force/kg mass or ft-lb force/lb mass
Hp = Power in kW or BHP
D = Impeller diameter in mm or inches

Fig 2.21.1 • Affinity law relationships



Exist flow	New flow	Exist head	New head	Exist DIA	New DIA	Speed	Exist HP	New HP	Exist EFF %	New EFF %
315	345	150	180	6.5	7.12	3550	17.6	23.2	67.5	72.5
262	287	159	190	6.5	7.12	3550	16.4	22	64	68.0
150	164	171	205	6.5	7.12	3550	13	17	51	53.5
375	411	132	158	6.5	7.12	3550	16.3	22	67.5	71.5

Fig 2.21.2 • Performance change vs. impeller diameter change (Courtesy of Union Pump Company)



$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2};\ \frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2;\ \frac{HP_1}{HP_2} = \left(\frac{N_1}{N_2}\right)^3$$

EXIST FLOW	EXIST HEAD	EXIST NPSH	EXIST HP	EXIST RPM	EXIST EFF	NEW FLOW	NEW HEAD	NEW RPM	NEW HP	NEW NPSH
612	35	3.5	9	1150	60	931	81	1750	32	7.14
875	32	4.0	10	1150	70	1331	74	1750	36	8.16
1137	26	5.0	10.5	1150	71	1730	60	1750	37	10.2

Fig 2.21.3 • Performance change versus speed change

exceed that of existing pumps (refer to Figure 2.21.1 for affinity law relationships).

Once a pump has been selected and the impeller diameter to deliver a defined flow rate for a required level of head (energy) has been determined, the affinity laws can be used to determine what new speed or impeller diameter is required to satisfy the alternative operating conditions. A revised flow versus head (energy) versus horsepower curve can be developed and plotted from these relationships. Refer to Figure 2.21.2 for an example of an impeller diameter change.

The affinity law relationships for changing impeller diameter usually work pretty well for relatively small changes; of the order of 10%. If the change exceeds 10%, the relationship between the impeller and casing can change significantly, which potentially can alter the design configuration of the pump. It is always good practice to check the pump rating curves to determine whether the pump has been tested with that particular impeller diameter.

The variation of pump performance with changes in speed also follows the affinity laws, but with a higher level of accuracy than do changes in impeller diameter.

When the performance is known for a pump operating at a lower speed (e.g. 1150 rpm) and it is desired to operate at a higher speed (e.g. 1750 rpm), the affinity laws can be used to calculate performance at the new speed, and the converse is also

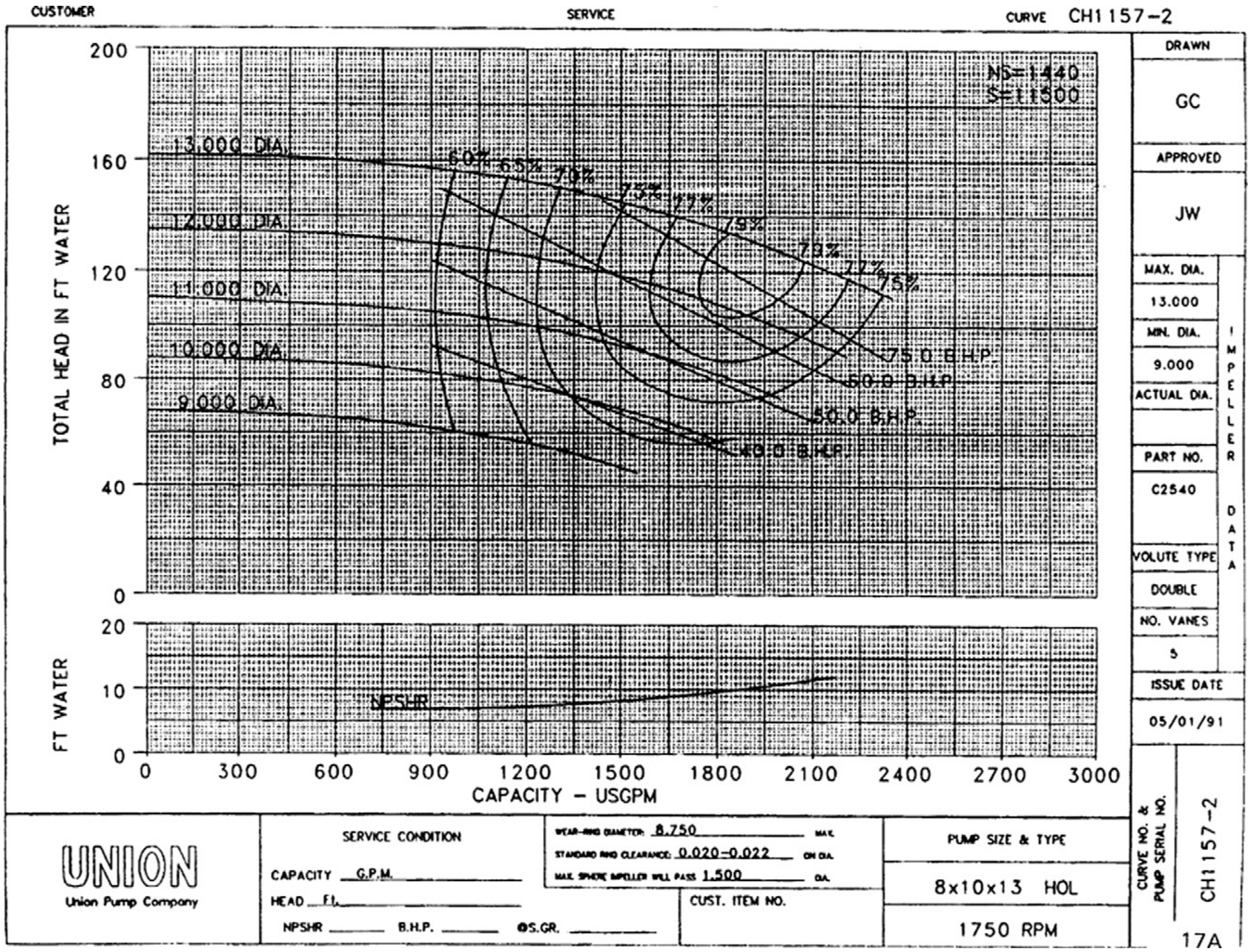


Fig 2.21.4 • Speed change performance verification (Courtesy of Union Pump Company)

true. However, it is good practice to verify the mechanical integrity of the pump coupling and driver with the vendor before proceeding with the change. The mechanical design of the process system must also be confirmed for the revised pressures and flow rates (refer to Figure 2.21.3).

The data used for this example are taken from an actual pump operating at 1150 rpm. The affinity law calculation results compare favorably with the same pump operating at 1750 rpm (refer to Figure 2.21.4).

Best Practice 2.22

Require a plant pump spare changeover philosophy to ensure optimum pump MTBF.

All pumps are subjected to the highest component forces and changes during transient (start-up and shut down) operation.

Whilst it is true that spare pumps must be periodically checked to ensure safe and reliable operation when required, the periods between checks should be optimized.

We have found that the changeover period which results in the highest possible pump MTBF is between 3 and 6 months in most pump applications. Notable exceptions are firewater pumps (weekly – for safety) and seawater pumps (to eliminate corrosion).

Lessons Learned

Lack of a plant spare changeover philosophy has been responsible for low pump MTBFs, safety issues and process unit shutdowns.

Many plants either do not have a stated change over philosophy, or do not implement the stated policy. In cases where pumps operate on fluids with a high vapor pressure, too frequent changeover (less than 3 months) has resulted in low mechanical seal MTBFs (less than 12 months).

Benchmarks

This best practice has been followed since the mid-1990s to optimize pump safety and maximize pump MTBFs (in excess of 80 months).

B.P. 2.22. Supporting Material

As stated in this best practice material above, we have found that most plants do not practice a consistent pump changeover philosophy. It is usually practiced in firewater pump applications but not many others. Where it is practiced, the periods are inconsistent and not always followed.

We have taken the following approaches in this regard:

- Review pump maintenance site and company records to discover what changeover periods have resulted in the highest pump MTBFs.
- If the history records are either not available or inconclusive, we recommend the following action:
- Start with a period of 3 months and extend based on condition monitoring results
- Determine the desired operating period for the spare pump (Equal periods or a limited period to monitor spare pump performance – minimum time should be 4 hours).

Best Practice 2.23

Consider the drive system limitation whenever considering impeller diameter increase, to ensure new operating requirements can be met without driver overload.

Driver power requirements increase by the cube of the flow increase.

If additional pump flow can be accommodated in the existing process unit, pump impeller diameter or the number of impellers in a multistage pump can be increased by a limited amount, provided that the driver has sufficient power.

Most pump applications are based on industry and end user specifications that only require 10% additional driver power above the rated power of the pump.

Since driver power requirement is a cube power function of flow, this means that the diameter increase has to be limited to an approximate 2.5% increase if there is only 10% excess driver power available.

If the electrical system and the pump unit baseplate permit, a larger motor may be able to be used, along with a larger coupling for additional pump flow above 2.5% and corresponding power requirements.

Lessons Learned

Many centrifugal pump impeller diameter increases have resulted in driver and/or coupling overloads when not properly checked.

It has been my experience that there are many applications where larger drivers were not originally used and impeller diameter was still increased, resulting in driver overload (mostly motor drivers). Eventually, the baseplate was modified for a larger driver.

Benchmarks

This best practice guideline has been used since the mid-1970s, and I have always inquired during the pre-FEED or FEED project phases regarding possible pre-investment in a larger motor, in process units where the original design pump capacity will be a bottleneck for larger future process unit rates.

B.P. 2.23. Supporting Material

The affinity laws

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H = Head in m-kg force/kg mass or ft-lb force/lb mass
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Fig 2.23.1 • Affinity law relationships

Once a pump has been selected and the impeller diameter has been determined to deliver a defined flow rate for a required level of head (energy), the affinity laws can be used to determine what new speed or impeller diameter is required to satisfy the alternative operating conditions. A revised flow versus head (energy) versus horsepower curve can be developed and plotted from these relationships. Refer to Figure 2.23.2 for the example of an impeller diameter change.

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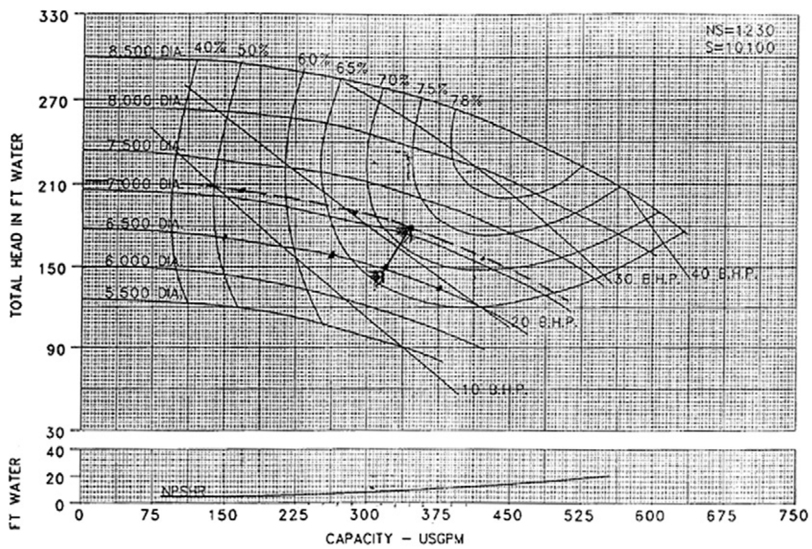

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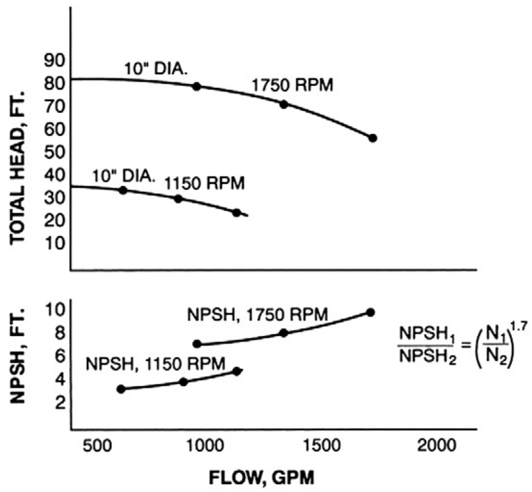

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Fig 2.23.3 • Performance change versus speed change

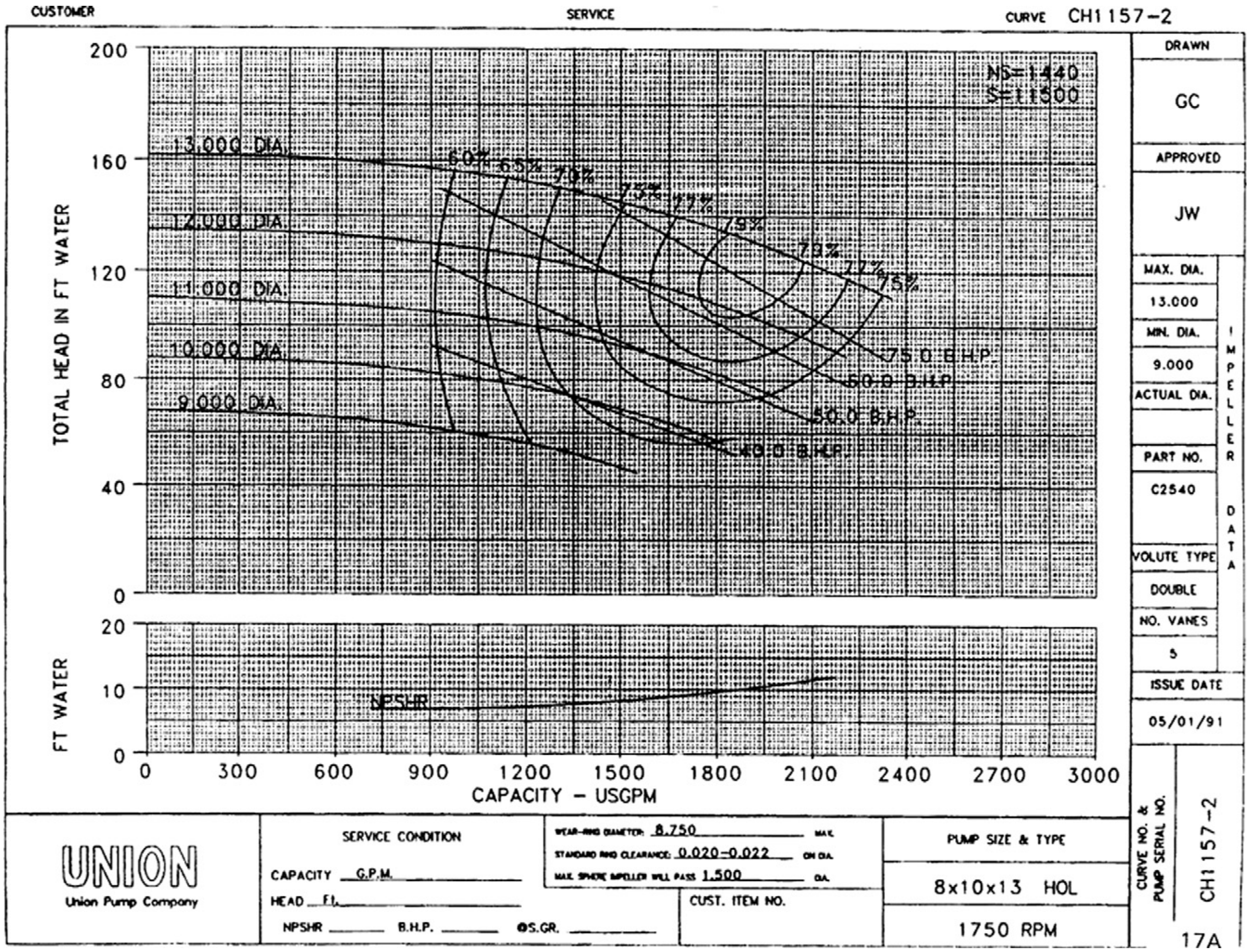


Fig 2.23.4 • Speed change performance verification (Courtesy of Union Pump Company)

Best Practice 2.24

The use of variable speed motors in low voltage services is effective in terms of energy and reliability.

Consider a low voltage motor best practice of always using variable frequency driver motors to provide just the amount of power required for process requirements.

Install a control system (level, pressure or flow) that will regulate pump motor driver speed to meet set point requirements without the use of a process control valve.

This approach will reduce required power in low voltage applications (less than 600 volts) by as much as 25%.

At the present time, the additional installed cost of low voltage VFDs can be paid for in less than three years by the energy savings produced.

Lessons Learned

Controlling centrifugal pumps with control valves is not energy efficient, and can expose centrifugal pumps to reliability issues due to control valve instability.

The writer has experienced large savings by using VFDs and not using process control valves which typically have a pressure drop of 23 meters (75 feet) of head or more and expose the pump process control system to valve instability and maintenance (valve packing and diaphragm replacement).

Benchmarks

This best practice has been used since 2005, and has resulted in significant energy savings and increased pump control system safety and reliability.

B.P. 2.24. Supporting Material

See B.P. 2.2.6 and 2.2.21 for supporting material.

Best Practice 2.25

Have every centrifugal pump curve available in the control room and instruct all operators on their use to optimize centrifugal pump safety and MTBF.

Centrifugal pumps produce flow inversely proportional to the process head required.

The safety and reliability of all centrifugal pumps is optimized if pumps are operated within the equipment reliability operating envelope known as the EROE.

This flow range is obtained by having operations aware of the centrifugal pump characteristic, providing process targets and having the pump test curves available for each pump for operator use and understanding.

Lessons Learned

Unnecessary centrifugal pump maintenance and pump failures result from operators not checking the pump test curves, or confirming that the pump operates within its EROE and understanding their use.

We have found that approximately 80% of the root causes of centrifugal low MTBFs (below 36 months) are due to process changes, not operator error, which can be significantly avoided by incorporating this best practice.

Benchmarks

I have employed this best practice since 1990 in upstream oil and gas, refinery and chemical plants to optimize plant safety and to produce pump MTBFs in excess of 80 months.

3

Compressor Best Practices

Compressors are the highest producers of revenue among driven equipment, in upstream and down-stream industries. They are usually un-spared, and are considered as critical equipment items. Compressor reliability is therefore a high priority, and is directly proportional to company profit. Many compressor trains are continuously condition-monitored for predictive maintenance purposes to obtain optimum reliability – which can exceed 99.7% for compressor types that are properly selected, designed, installed and monitored.

This chapter will address best practices that will optimize site compressor safety, reliability, and minimize repair time and maintenance costs.

Best Practice 3.1

Use site, company and industry lessons learned to determine if positive displacement or dynamic (turbo) type compressors are required.

First, accurately determine all of the required operating conditions. Process conditions not anticipated in the selection phase account for 80% or more of compressor reliability problems. Then use company guidelines for selection of the proper compressor type if available. Whether company guidelines are available or not, determine the type by site and industry lessons learned to avoid plant safety and reliability issues. Finally, indicate the selected compressor type on the data sheet with all best practices dictated by plant, company and industry lessons learned.

Lessons Learned

Improperly selected compressor type has led to field safety issues, revenue and legal costs as noted by the following examples:

- Lubricated screw compressor in sour gas service – never operated for more than 24 hours continuously.
- Reciprocating compressors, used when screw compressors should be used, resulted in extensive maintenance costs and sour gas leakage.
- Rotary lobe compressors, used instead of centrifugal compressors, resulted in the need for multiple compressors when a single centrifugal compressor could have been used.

Benchmarks

I have used this best practice since the mid-1980s in upstream, refining and chemical plants, giving the following optimum compressor reliabilities:

- Centrifugal greater than 99.5%
- Dry screw greater than 99%
- Lubricated screw greater than 97.5%
- Dry reciprocating greater than 94%

B.P. 3.1. Supporting Material

This section gives an overview of compressor types and their typical applications.

The two basic types of compressors are positive displacement and dynamic compressors.

Positive displacement compressors are constant volume, variable energy (head) machines that are not affected by gas characteristics.

Dynamic compressors are variable volume, constant energy (head) machines that are significantly affected by gas characteristics.

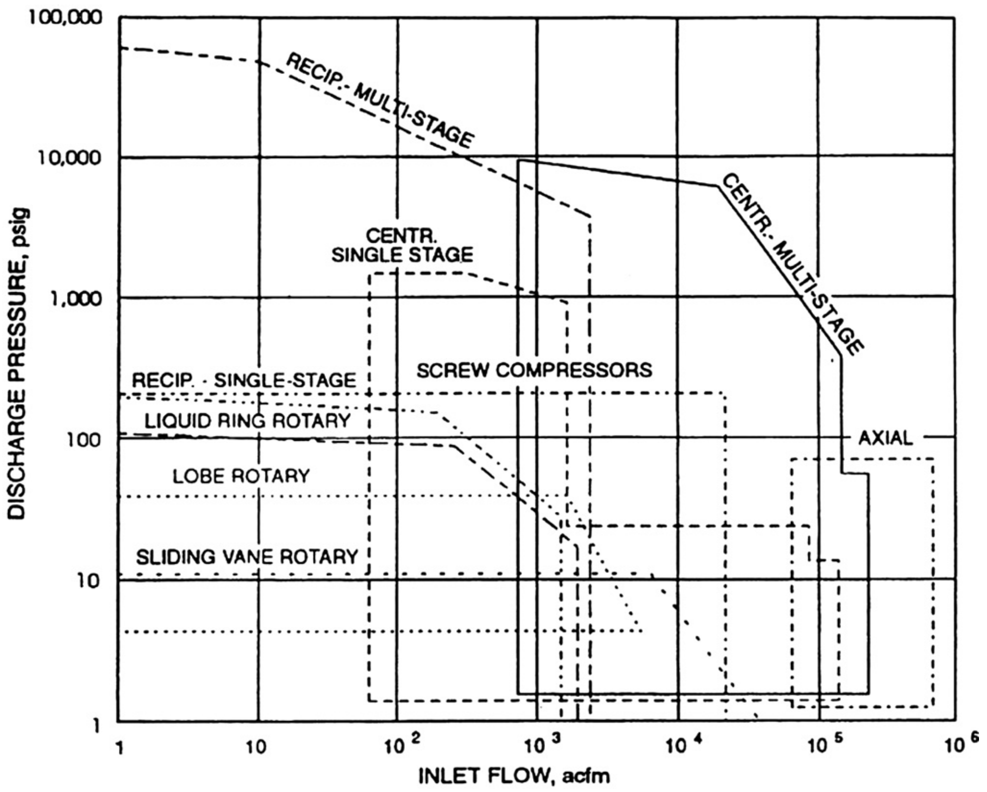


Fig 3.1.1 • Compressor application range chart

The type of compressor that will be used for a specific application therefore depends on the flow rate and pressure required, and the characteristics of the gas to be compressed.

In general, dynamic compressors are the first preference, since they have the lowest maintenance requirements. The next choice is the rotary type, with positive displacement compressor since they do not contain valves and are gas pulsation-free. The last choice is a reciprocating compressor, since this type has the highest maintenance requirements and produces gas pulsations. However, the type that is finally chosen depends upon the specific requirements of the application, as discussed below.

Figure 3.1.1 presents a flow range chart showing the various types of compressor applications as a function of flow (acfm) and discharge pressure (psig).

Table 3.1.1 shows the typical operating ranges for the various types of compressors used in the refining, chemical and gas processing industries.

Table 3.1.2 describes the typical applications for the various types of compressors presented in Figure 3.1.1 and Table 3.1.1.

Positive displacement compressors

Positive displacement compressors are used for low flow and/or low molecular weight (hydrogen mixture) applications. The various types are presented below.

Rotary lobe

A rotary lobe compressor consists of identically synchronized rotors. The rotors are synchronized through use of an external, oil-lubricated, timing gear, which positively prevents rotor contact, and which minimizes meshing rotor clearance to optimize efficiency. This feature also allows the compressor to be oil free in the gas path. The rotors of the two-lobe compressor each have two lobes. When the rotor rotates, gas is trapped between the rotor lobes and the compressor casing. The rotating rotor forces the gas from the gas inlet port, along the casing, to the gas discharge port. Discharge begins as the edge of the leading lobe passes the edge of the discharge port. The trailing lobe pushes the entrapped gas into the discharge port, which compresses the gas against the backpressure of the system. Rotary lobe compressors are usually supplied with noise

Table 3.1.1 Typical operating range of various types of gas compressors

Machine Type	Capacity m^3/hr (icfm) Min	Max	T_2 Max °C (°F)	P_1 Max kPa (psia)	P_2 Max kPa (psia)	P/R Min	P/R Max
Rotary lobe	1 to 68,000 (1 to 40,000)		177 (350)	240 (35)	380 (55)	1.0+	2.4
Rotary vane	75 to 5,500 (45 to 3,300)		177 (350)	340 (45)	450 (65)	1.3	3.2
Rotary screw	80 to 34,000 (50 to 20,000)		177 (350)	1,000 b (150)	4,250 () (615)	2.0	6.0
Recip	1 to 17,000 (1 to 10,000)		427 (800)	6,900 (1,000)	69,000 (10,000)	3.0	50.0
Liquid ring	17 to 17,000 (10 to 10,000)		N/A	690 (100)	965 (140)	1.0+	10.0
Centrifugal	1,200 to 250,000 (700 to 150,000)		260 (500)	6,900 (1,000)	9,650 (1,400)	1.0+	3.4
Single stage							
Centrifugal	500 to 250,000 (300 to 150,000)		427 (800)	13,800 (2,000)	41,400 (6,000)	2.0	10.0
Multi stage							
Axial	125,000 to 600,000 (75,000 to 350,000)		427 (800)	210 (30)	1,030 (150)	1.0	10.0

Table 3.1.2 Typical compressor applications

Compressor Type	Application
Rotary lobe	Conveying — powder, polyethylene
Rotary vane air	Air blowers (low volume). Also used as gas turbine starters
Rotary screw	Plant and instrument air, low flow process — off gas, recycle, sulfur blowers
Rotary liquid ring	Crude unit vacuum, various saturated gas applications
Reciprocating	Plant and instrument air off gas (low flow) recycle (low flow) H_2 make-up, gas reinjection (low flow)
Centrifugal single stage	Air blowers, recycle
Centrifugal single stage — integral gear	Low flow recycle, off gas, plant air (can replace a recip. in low flow, medium to high molecular weight applications)
Centrifugal multi-stage side load (horizontal split or barrel)	(Propane, propylene, ethylene, Freon, mixed gas) refrigeration
Centrifugal multi-stage barrel	Recycle, reinjection, syn gas
Centrifugal multi-stage — integral gear	Plant and instrument air *Process applications require proven field experience
Axial compressor	FCC blower, MTBE effluent, gas turbine air

enclosures or silencers to reduce their characteristically high noise level. A schematic of a two-lobe rotary compressor is shown in Figure 3.1.2.

Rotary vane

A sliding-vane rotary compressor uses a series of vanes that slide freely, in longitudinal slots that are cut into the rotor. Centrifugal force causes the vanes to move outward against the casing wall. The chamber that is formed between the rotor, between any two vanes, and the casing is referred to as a cell. As the rotor turns, an individual vane passes the inlet port to form a cell between itself and the vane that precedes it. As an individual vane rotates toward the end of the inlet port, the volume of this cell increases. This increase in volume creates a partial vacuum in the cell, which draws the gas in through the inlet port. When a vane passes the inlet port, the cell is closed, and the gas is trapped between the two vanes, the rotor and the casing. As rotation continues toward the discharge port, the volume of the cell decreases. The vanes ride against the casing and slide back into the rotor. The decrease in volume increases the gas pressure. The high pressure gas is discharged out of the compressor through the gas discharge port. Sliding-vane rotary compressors have a high noise level that results from the vane motion. A schematic of a sliding-vane rotary compressor is shown in Figure 3.1.3.

Rotary screw

The single-stage design consists of a pair of rotors that mesh in a one-piece, dual-bore cylinder. The male rotor usually consists of four helical threads that are spaced 90 degrees apart. The female rotor usually consists of six helical grooves that are spaced 60 degrees apart. The rotor speed ratio is inversely

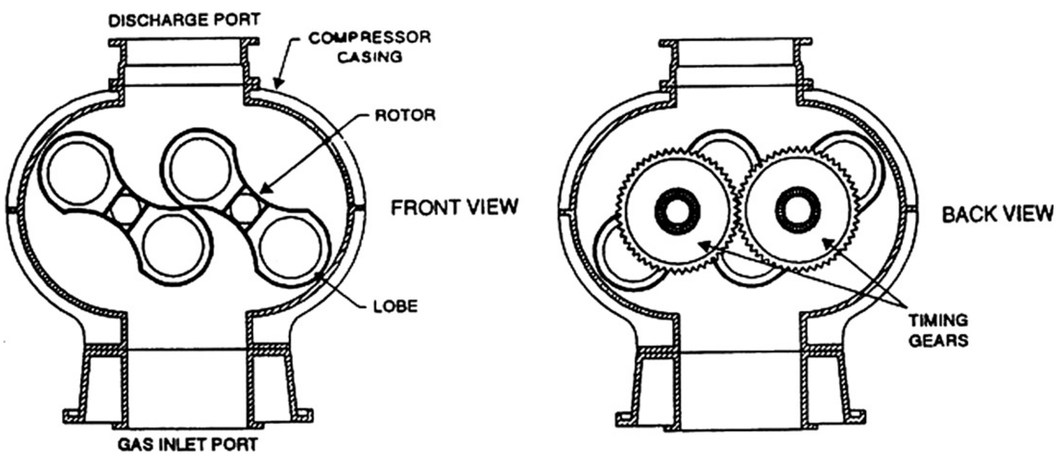


Fig 3.1.2 • Rotary lobe

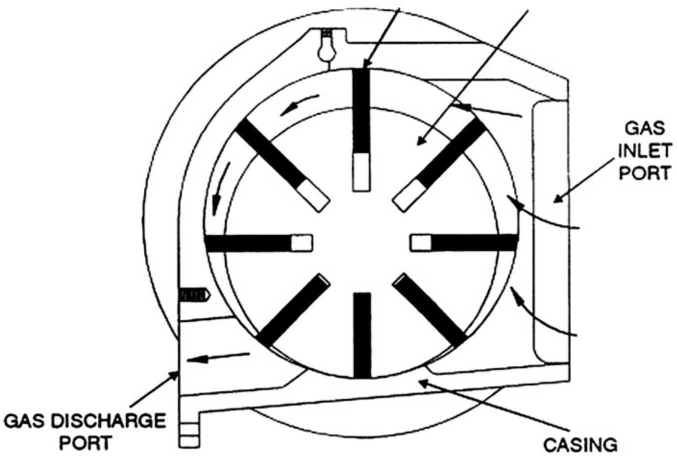


Fig 3.1.3 • Rotary vane

proportional to the thread-groove ratio. In the four-thread, six-groove, screw compressor, when the male rotor rotates at 1800 rpm, the female rotor rotates at 1200 rpm.

The male rotor is usually the driven rotor, and the female rotor is usually driven by the male rotor. A film of foil is normally injected between the rotors to provide a seal, and to prevent metal-to-metal contact. An oil-mist eliminator, installed immediately downstream of the compressor, is required for plant and instrument air service. However, designs are available that do not require lubrication. Screw compressors that do not require lubrication are commonly referred to as 'dry screw-type compressors'.

The inlet port is located at the drive-shaft end of the cylinder. The discharge port is located at the opposite end of the cylinder. Compression begins as the rotors enmesh at the inlet port. Gas is drawn into the cavity between the male rotor threads and female rotor grooves. As rotation continues, the rotor threads pass the edges of the inlet ports and trap the gas in the cell that is formed by the rotor cavities and the cylinder wall. Further rotation causes the male rotor thread to roll into the female rotor groove and to decrease the volume of the cell. The decrease in the volume increases the cell pressure. Oil is normally injected after the cell is closed to the inlet port. The oil seals the clearances between the threads and the grooves, and it absorbs the heat of compression. Compression continues until the rotor threads pass the edge of the discharge port and release the compressed gas and oil mixture. A typical single stage screw compressor is shown in Figure 3.1.4.

Rotary liquid ring

Liquid ring rotary compressors consist of a round, multi-blade rotor that revolves in an elliptical casing. The elliptical casing is partially filled with a liquid, which is usually water. As the rotor turns, the blades form a series of buckets which carry the liquid around with the rotor. Because the liquid follows the contour of the casing, it alternately leaves and returns to the space between the blades. The space between the blades serves as a rotor chamber. The gas inlet and discharge ports are located at the inner diameter of the rotor chamber. As the liquid leaves the rotor chamber, gas is drawn into it through the inlet ports. As the rotor continues to rotate, the liquid returns to the rotor chamber and decreases the volume in the chamber, hence the gas pressure increases. As the rotor chamber passes the discharge port, the compressed gas is discharged into a gas/liquid separator and then to the process. A typical liquid ring rotary compressor is shown in Figure 3.1.5.

Reciprocating

The basic components of a reciprocating compressor are a crankshaft, crossheads, piston rod packing, cylinders, pistons,

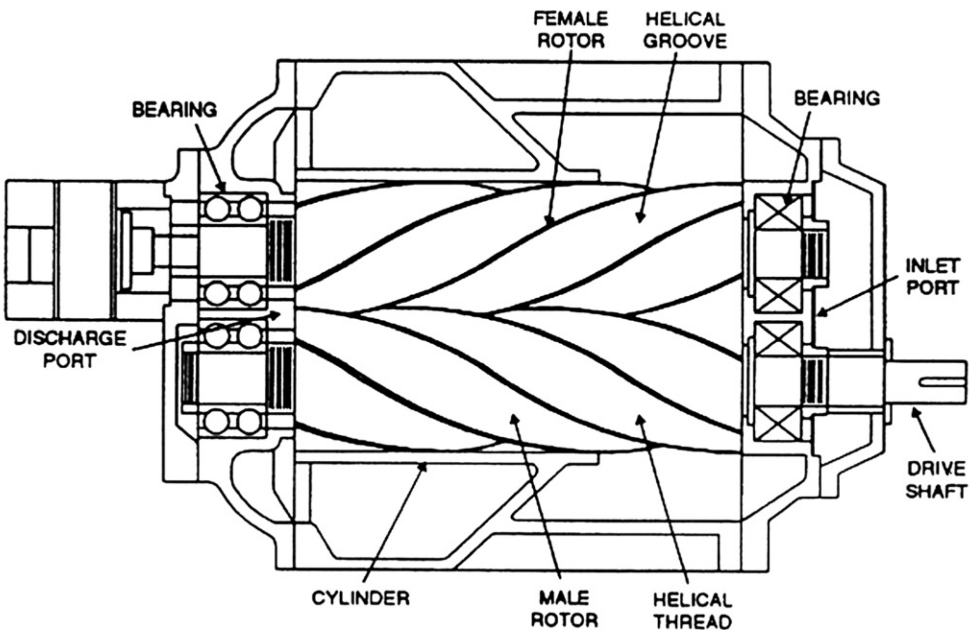


Fig 3.1.4 • Rotary screw

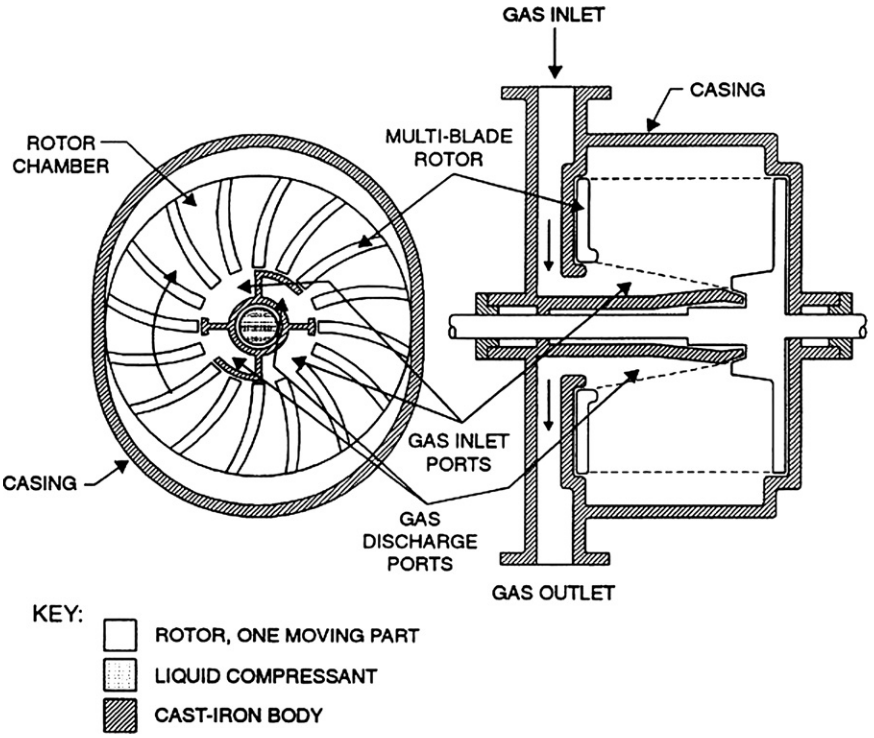


Fig 3.1.5 • Rotary liquid ring (Courtesy of Nash Engineering Company)

suction valves, and discharge valves. Figure 3.1.6 is an illustration of a three-stage reciprocating compressor. Note that the third stage piston and cylinder are mounted on top of the second stage piston and cylinder. A prime mover (not shown) rotates the crankshaft. The crankshaft converts the rotary motion of the prime mover into reciprocating motion of the pistons.

The compression cycle of a reciprocating compressor consists of two strokes of the piston: the suction stroke and the compression stroke. The suction stroke begins when the piston moves away from the inlet port of the cylinder. The gas in the space between the piston and the inlet port expands rapidly until the pressure decreases below that on the opposite side of

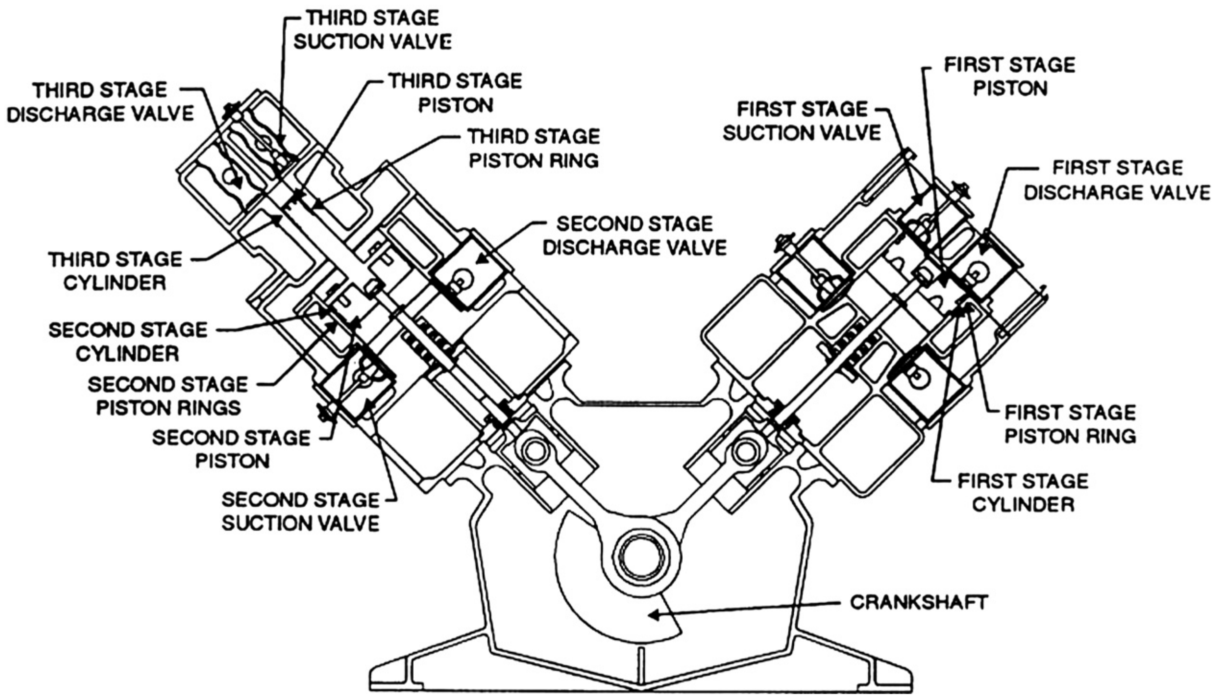


Fig 3.1.6 • Reciprocating compressor

the suction valve. The pressure difference across the suction valve causes the suction valve to open and admit gas into the cylinder. The gas flows into the cylinder until the piston reaches the end of its stroke. The compressor stroke starts when the piston starts its return movement. When the pressure in the cylinder increases above the pressure on the opposite side of the suction valve, the suction valve closes to trap the gas inside the cylinder. As the piston continues to move toward the end of the cylinder, the volume of the gas in the cylinder decreases, and the pressure of the gas increases. When the pressure inside the cylinder reaches the design pressure of the stage, the discharge valve opens and discharges the contents of the cylinder to the suction of the second stage. The second stage takes a suction on the discharge of the first stage, further compresses the gas and discharges to the third stage. The third stage takes a suction on the discharge of the second stage and compresses the gas to the final discharge pressure.

Figure 3.1.7 shows a balanced opposed, four throw reciprocating compressor typical of the type that would be used in refinery hydrogen make-up service.

Dynamic compressors

Dynamic compressors are used wherever possible because of their low maintenance requirements. The single stage integral gear centrifugal compressor allows the use of a dynamic compressor in many applications where positive displacement compressors have previously been used. The two types of dynamic compressors are centrifugal and axial.

Centrifugal compressors – general principles of operation

Centrifugal compressors increase the energy of a gas by increasing the tangential velocity (V_T) of a gas, as shown in Figure 3.1.8. The principle of operation of a centrifugal compressor is very similar to that of a centrifugal pump. The gas enters through the inlet nozzle – which is proportioned so that the gas enters the impeller with minimum shock or turbulence. The impeller, which consists of a hub and blades, is mounted on a rotating shaft. This receives the gas from the inlet nozzle and dynamically compresses it, by increasing the energy of gas proportional to the product of the blade tip speed velocity (U_T) and the gas tangential velocity change (V_T) in the impeller. V_R represents the velocity of the gas relative to the blade. The resultant velocity (V) is the vector sum of the relative velocity (V_R) and the blade tip speed velocity (U_T).

A diffuser surrounds the impeller, and it gradually reduces the velocity of the gas as it leaves the impeller. The diffuser converts the velocity energy to a higher pressure level. In a single stage compressor, the gas exits the diffuser through a volute casing that surrounds the diffuser. The volute casing collects the gas to further reduce its velocity, and to recover additional velocity energy. The gas exits through the discharge nozzle. In a multi-stage compressor, the gas exits the diffuser and enters return vanes, which direct the gas into the impeller of the next stage.

Centrifugal single stage (low ratio)

A typical single stage centrifugal low ratio compressor is shown in Figure 3.1.9. These types are known as single stage overhung compressors, since the impeller is outboard of the radial bearings the cases are radially split.

Centrifugal single stage integral gear

Figure 3.1.10 shows a “Sundyne” single stage integral gear compressor section. This type of compressor is an in-line type (similar to a pump), usually driven by motor through an integrally mounted gear box (not shown). These compressors are used for low flow, high energy (head) applications, and are now used in many applications that were previously serviced by positive displacement compressors. These compressors operate at high speeds (8,000 – 34,000 rpm), and are limited to approximately 300 kW (400 horsepower).

Centrifugal multi-stage horizontal split

A typical, multi-stage, horizontally split, centrifugal compressor is shown in Figure 3.1.11. The casing is divided into upper and lower halves along the horizontal centerline. This allows access to the internal components of the compressor without disturbing the rotor to casing clearances or bearing alignment. If possible, piping nozzles should be mounted on the lower half of the compressor casing, to allow disassembly of the compressor without removal of the process piping.

Centrifugal multi-stage with side loads

This type of compressor is used exclusively for refrigeration services. The only difference from the previous type is that gas is induced or removed from the compressor via side load nozzles. A typical refrigeration compressor is shown in Figure 3.1.12. Note that this type of compressor can be either horizontally or radially split.

Centrifugal multi-stage (barrel)

A typical multi-stage, radially-split, centrifugal compressor is shown in Figure 3.1.13. The compressor casing is constructed as a complete cylinder, with one end of the compressor being removable to allow access to the internal components. Multi-stage, radially-slit centrifugal compressors are commonly called barrel compressors.

Barrel compressors are used for the same types of applications as the multi-stage, horizontally-split centrifugal compressors. Because of the barrel design, however, they are normally used for higher pressure applications or certain low molecular weight gas compositions (hydrogen gas mixtures).

Centrifugal multi-stage integral gear

A typical four-stage, integrally-geared, centrifugal compressor is shown in Figure 3.1.14. Integrally-geared, centrifugal compressors have a low-speed (bull) gear that drives two or more high-speed gears (pinions). Impellers are mounted at one or both ends of each pinion. Each impeller has its own casing that is bolted to the gear casing. The gear casing may be horizontally split to allow access to the gears.

The gas enters the compressor through the first stage inlet nozzle to the impeller. The impeller receives the gas from the inlet nozzle, dynamically compresses it, and discharges to the diffuser. The diffuser surrounds the impeller, which gradually

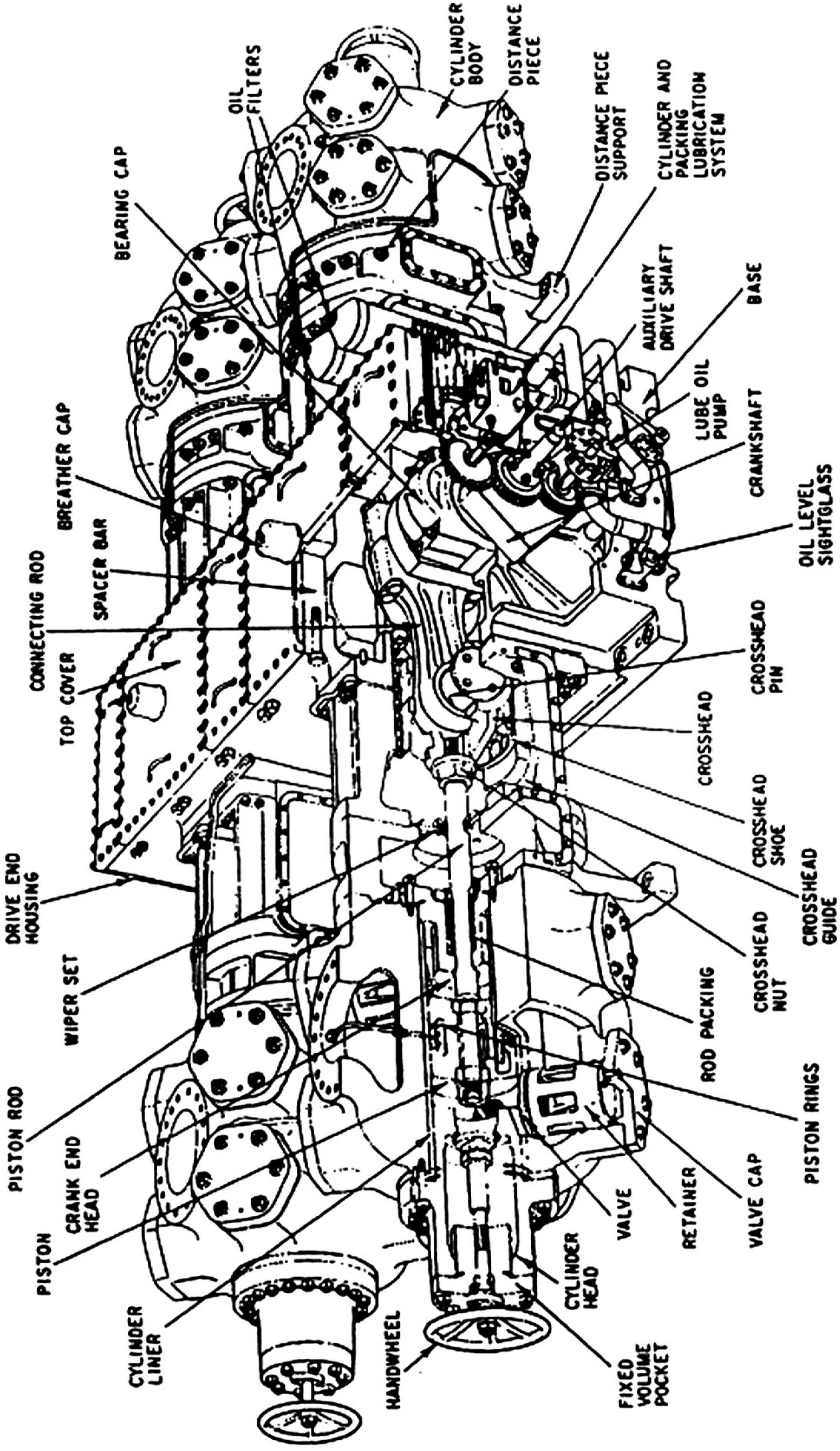


Fig 3.1.7 • Reciprocating compressor

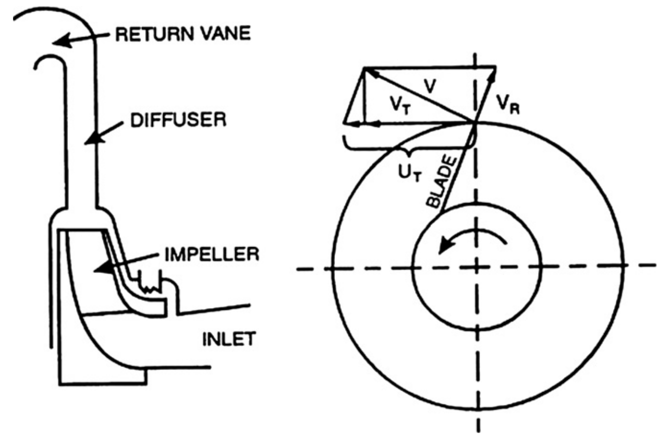


Fig 3.1.8 • Centrifugal compressor

reduces the velocity of the gas as it leaves the impeller. The gas exits the diffuser through a volute casing, which collects the gas, further reduces its velocity, and recovers additional velocity energy. The gas exits the first stage through the first stage discharge nozzle, enters an intercooler, and is then piped to the second stage. The discharge from the second stage enters an intercooler, and then it is piped to the third stage.

Axial horizontal split

A typical axial compressor is shown in Figure 3.1.15. An axial compressor consists of a rotor shaft with a series of rotating blades, and a tapered cylindrical casing with fixed stator vanes. Each set of blades is followed by a set of stator vanes. The gas enters the inlet nozzle, which guides it to the inlet volute. The inlet volute guides and accelerates the gas stream into the stator vanes. The stator vanes turn the gas stream to properly align the gas with the blades. The blades increase the energy of the gas by increasing its velocity. The stator vanes act as diffusers to provide resistance to the gas flow, and they cause the gas stream to decrease in velocity and to increase in pressure. Since blades and stator vanes alternate down the length of the casing, the gas is accelerated then decelerated several times before it leaves the compressor. Pressure is increased each time the gas flow meets a set of stator vanes. The gas exits the compressor through the discharge volute and discharge nozzle.

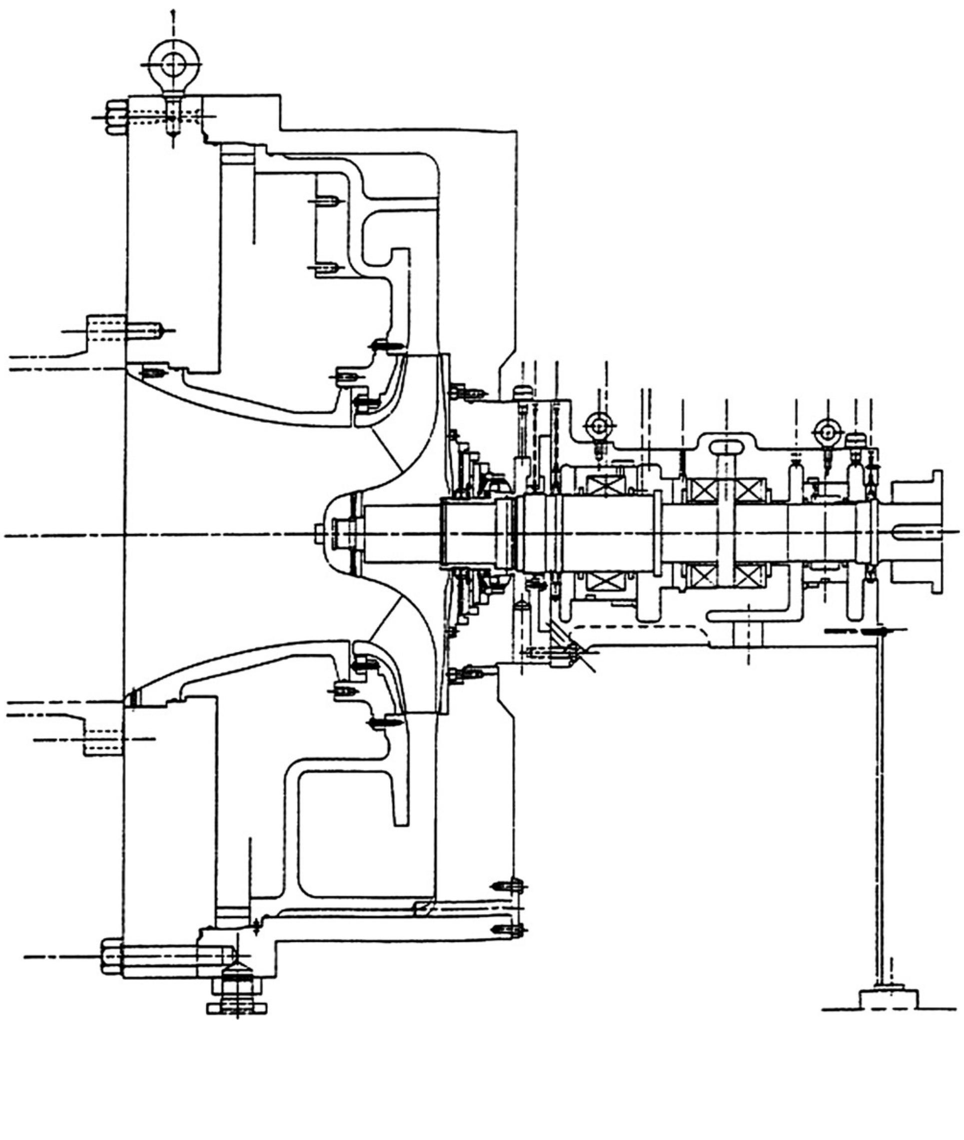


Fig 3.1.9 • Single stage overhung turbo compressor (Courtesy of A-C Compressor Corp.)

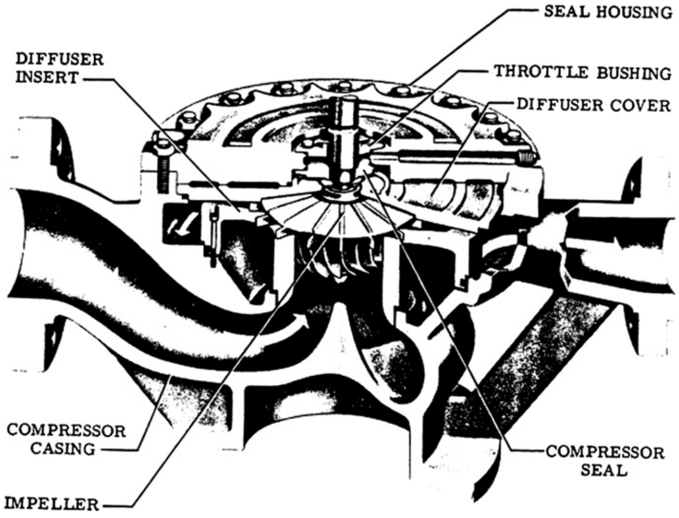


Fig 3.1.10 • Single stage high speed compressor (Courtesy of Sundstrand Corp.)

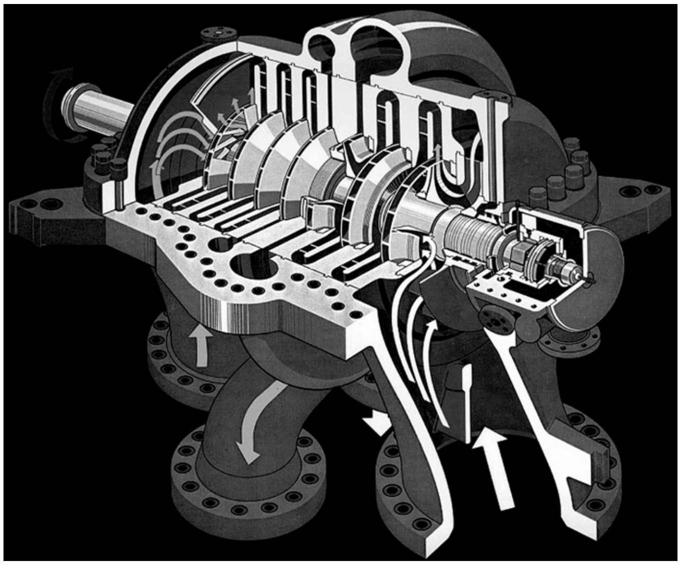


Fig 3.1.11 • Centrifugal multi-stage horizontal split (Courtesy of Mannesmann Demag)

Compressor characteristics

In this section we will discuss the two principle compressor characteristics: positive displacement and dynamic compression. In addition, the concepts of volume flow, mass flow and standard flow will be covered. Although this chapter covers compressors, the characteristics of positive displacement and dynamic are equally applicable to pumps.

Positive displacement compression is defined as the increase in pressure of a gas caused by operating on a fixed volume in a confined space. Types include reciprocating, rotary liquid piston, rotary lobe and screw compressors. This concept can best be envisioned by using a simple syringe. As one moves the plunger into the syringe, the volume inside changes. It will be displaced regardless of the resistance under which the compressor operates, provided sufficient power is available and the design of the compressor can meet the pressure requirements. Looking at a schematic of a positive displacement reciprocating compressor shows that gas will not enter the cylinder until the pressure inside it is lower than the suction pressure. Conversely, gas will not exit the cylinder until the pressure inside the cylinder is greater than the discharge pressure. The valves shown in this figure are merely check valves. The suction valves act as check valves, preventing the compressed gas from escaping back into the suction line. The characteristics, then, of a positive displacement compressor are fixed volume, variable pressure capability (energy or head) and not self-limiting. By this, we mean the compressor will stall or damage itself unless a pressure or power limiting device is included in its design. This is usually achieved by using a relief valve.

Before proceeding to the concept of dynamic compression, actual flow, mass flow and standard flow will be discussed. In the design of any compressor, actual volume flow must be used. This is necessary since the design is based on an optimal gas velocity. Gas velocity is the result of a given volume flow acting in a specific area. Think of any compressor, dynamic or positive displacement, compressing a volume of one actual cubic foot per minute, and let us assume that the temperature of compression remains constant. If the compressor in question has a compression ratio of two, one actual cubic foot per minute entering the compressor will be compressed to a discharge volume of exactly one half of a cubic foot per minute assuming that the gas is dry.

Standard volume is defined as one volume, always referenced to the same pressure and temperature conditions. In customary units, standard pressure is defined as 14.7 pounds per square inch and standard temperature is defined as 60°F. A measurement in standard cubic feet then is the ratio of the actual pressure to the referenced standard pressure and the referenced standard temperature to the actual temperature multiplied by the actual volume. Referring back to the previous example of a compressor with a compression ratio of two and no compression temperature increase, one can see that the standard cubic feet per minute in this compressor would remain the same, assuming a dry gas. This is because even though the actual volume of the gas does decrease by one half, the discharge standard volume is the ratio of the discharge pressure to the standard atmospheric pressure multiplied by the discharge volume. This will result in the same exit standard volume as the inlet.

Mass flow is the product of the actual volume flow and the density of the specific gas. The concept of mass flow and standard volume flow are the same. That is, the mass flow into the compressor example cited above will be exactly equal to the exit mass flow provided the gas is dry.

Both standard volume flow and mass flow are used to describe process capacities and are used in power calculations. After all, it would be very difficult to charge customers for produced products (gas) unless either a standard volume measure or weight measure were used.

We now consider dynamic compressors. These increase the pressure of a gas by using rotating blades to increase its velocity. They can be either axial or centrifugal compressors.

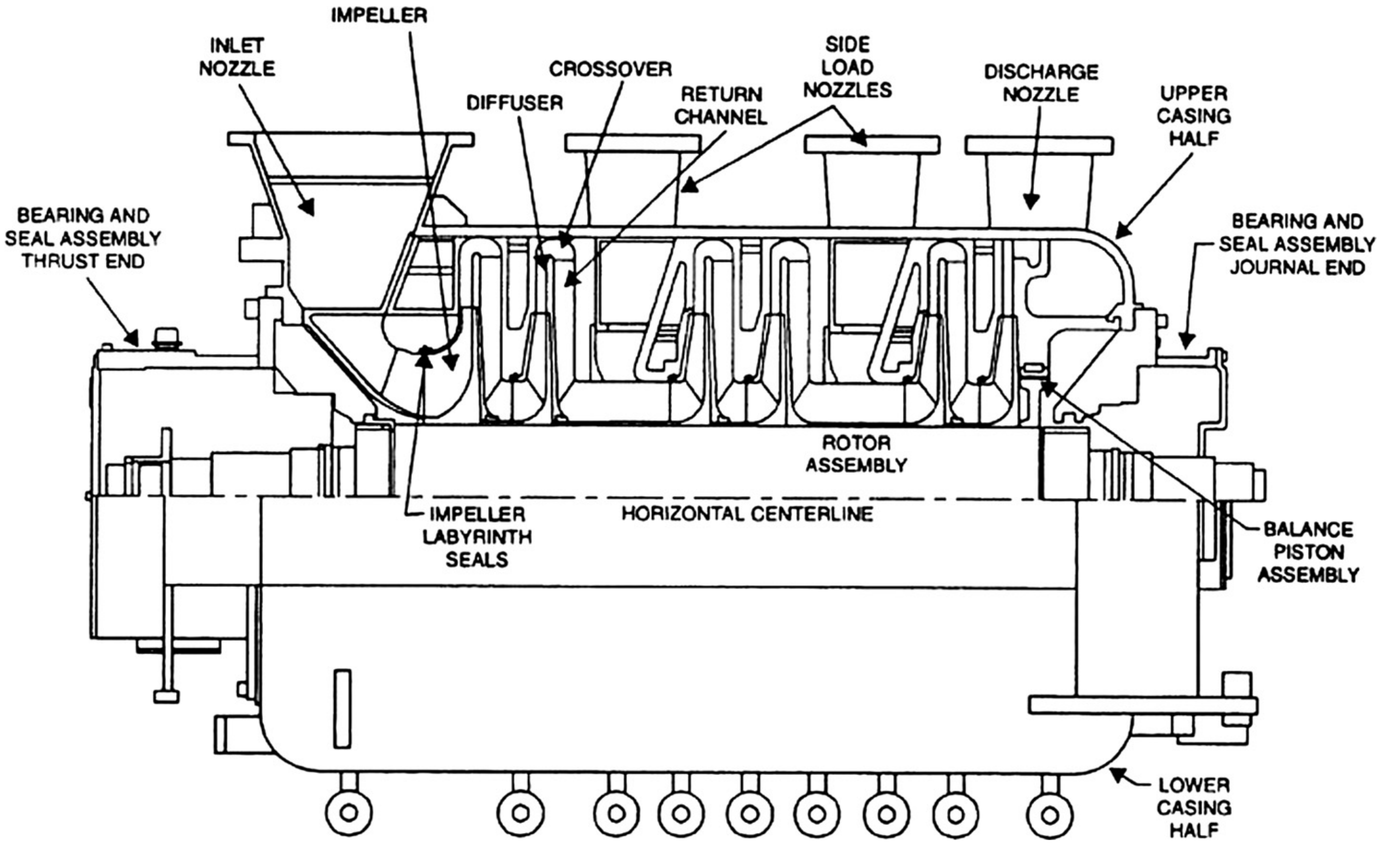


Fig 3.1.12 • Typical multi-stage refrigeration compressor

As we will see, there are basically two velocities which determine the performance of a dynamic compressor. Both of these occur at the exit of the moving blade. They are tip speed (the product of blade tip diameter and RPM) and relative velocity (velocity of the gas between two blades). The curve of a centrifugal compressor is significantly different from that of a positive displacement compressor. The

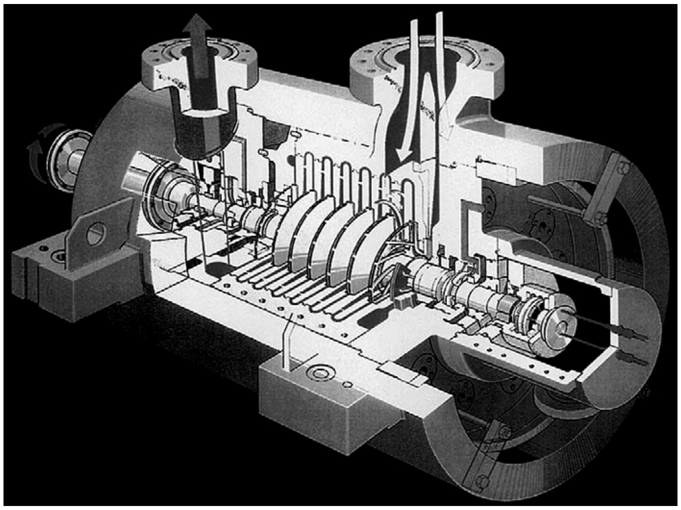


Fig 3.1.13 • Typical multi-stage, radially-split centrifugal compressor (Courtesy of Mannesmann Demag)

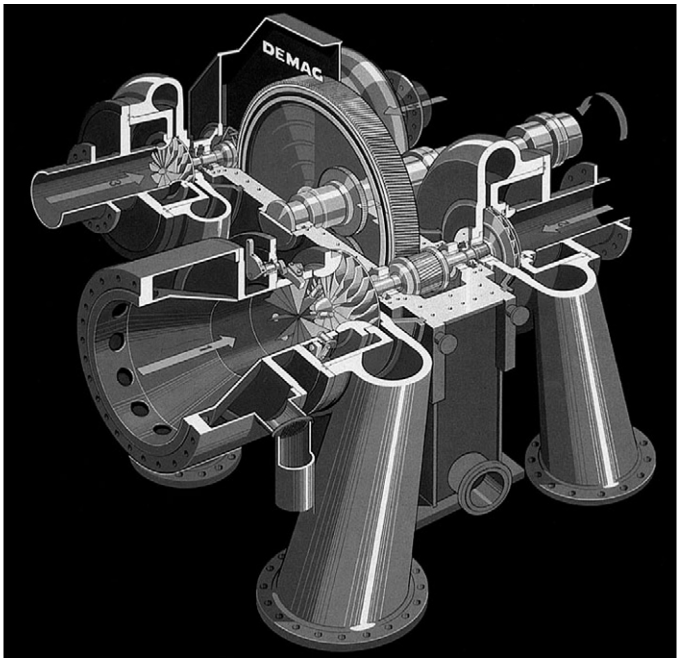


Fig 3.1.14 • Typical integrally-geared, centrifugal compressor (Courtesy of Mannesmann Demag)

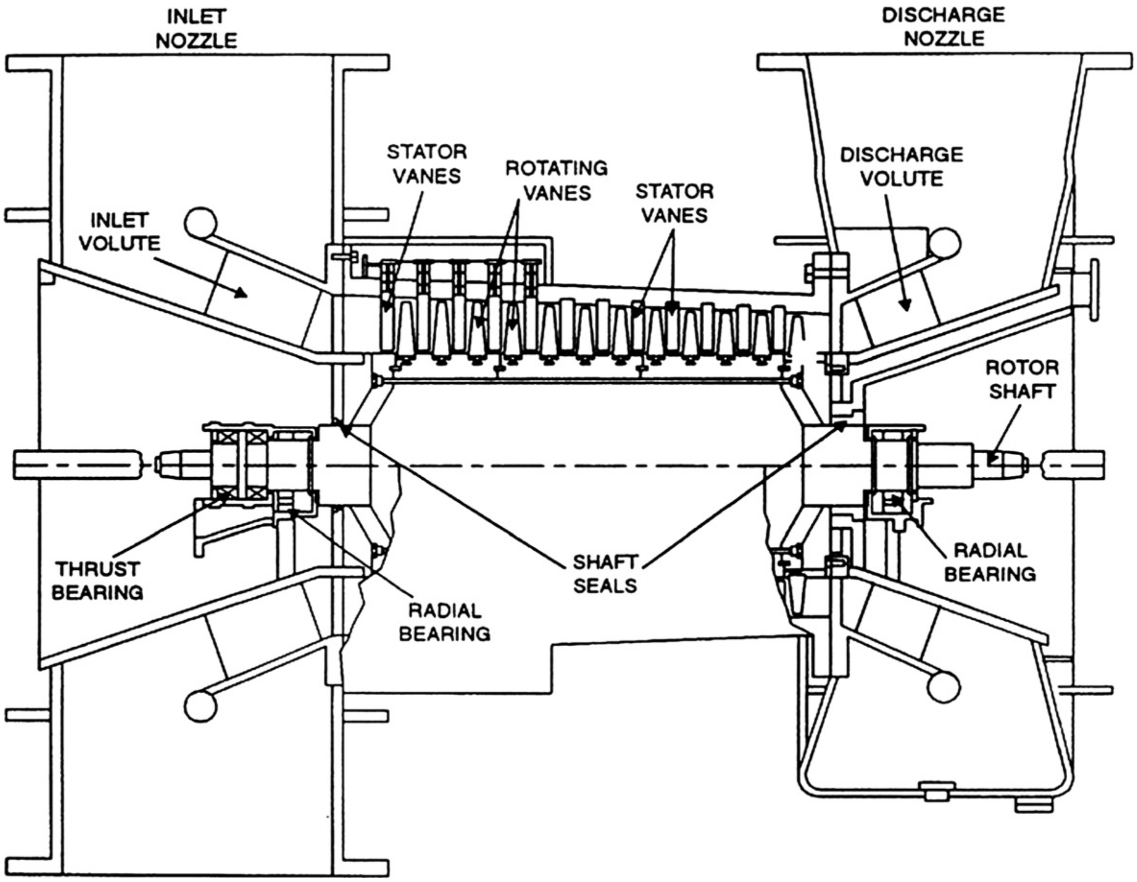


Fig 3.1.15 • Typical axial compressor

centrifugal compressor is a machine of variable capacity, fixed energy or head for a given flow and is self-limiting. That is, there is a maximum pressure which the compressor can produce and a maximum horsepower which it requires. Observing the curve of a typical dynamic compressor, one can see that the only place that a dynamic compressor can develop higher energy or head is at a lower flow for a given speed. We will see in the next chapter how the system combined with the compressor operating curve will determine the operating point. It should be noted that the operating curve of an axial compressor can sometimes approach that of a positive displacement compressor, in that its volume range is small and almost approximates the positive displacement curve.

The concept of an equivalent orifice is introduced in this section. Any typical axial compressor blade row or radial compressor impeller can be reduced to a series of equivalent orifices. The inlet of the blade row or impeller being one orifice, the exit being another and the eye and hub seals being additional equivalent orifices.

It should also be noted that both the suction and discharge process system for a given point in time and flow rate can be thought of as equivalent orifices placed at the inlet and discharge of the compressor flanges respectively.

In summary, the characteristics of positive displacement and dynamic compressors are as follows:

Flow – fixed for positive displacement, variable for dynamic.
Head – variable for positive displacement, fixed for dynamic.
Limiting characteristics for dynamic compressors – not self limiting for positive displacement (requires relieving device).

Best Practice 3.2

Define all process conditions and specific requirements on the compressor data sheet.

Confirm all process conditions on the data sheet, while accurately defining the gas analysis, the highest gas head required and all expected operating points (head and volume flows).

Meet with the EP&C and/or process licensor process engineers to ensure that all operating and upset process conditions are accurately defined.

Lessons Learned

Inaccurate process conditions have led to scope changes during the project and increased delivery time by 6 months or longer.

One recent example was a lubricated screw compressor installation where the compressor could not operate for more than 24 continuous hours due to the actual field gas analysis (sour) being different from the data sheet gas analysis (sweet).

Benchmarks

Since the mid-1970s I have met with process engineering and senior operators in the process design phase to ensure that all conditions were defined. This best practice has resulted in compressor reliability of the highest levels (greater than 99.7%).

B.P. 3.2. Supporting Material

In this section we will cover the relationships that the compressor vendor uses to determine the head produced, efficiency, horsepower required and overall design for a particular compressor application. The end user's or the purchaser's objective is to deliver a specified amount of a given gas to the process. Therefore, the data that the compressor vendor obtains is required mass flow, inlet pressure, temperature conditions and gas composition. With this data a compressor manufacturer will calculate actual flow, the ideal energy and the horsepower that is required to achieve that objective. The calculation for horsepower will require a specific compressor efficiency, as well as compressor mechanical losses; i.e., friction losses from bearings and discs, and seal losses.

Gas characteristics are defined in this chapter. and useful relationships are presented to enable the reader to calculate various compressor requirements. Once the vendor obtains the data, the gas head can be calculated. Once the head and required flow are known, the impeller can be selected.

The principle of impeller design is chiefly based on that of specific speed. This is defined as the ratio of speed times the square root of the actual flow divided by head raised to the three quarters power. It can be shown that increasing values of specific speed will result in increasing impeller efficiencies. Therefore, having been given the required flow and energy (head), the only source of obtaining higher specific speed for the vendor is to increase the compressor speed. This fact is very significant, because while compressors have increased in efficiency over the years, their mechanical requirements have also increased significantly, i.e., higher bore impeller stresses, etc. resulting in potential reliability problems. Therefore, the design of the impeller is a very fine balance between the performance requirements and the mechanical constraints of the components used in the compressor design.

Efficiency is presented as a ratio of ideal to actual energy, as depicted on a typical Mollier Diagram. In addition, the Fan Laws are presented, showing how increased impeller energy can be obtained via speed change in a compressor application.

Satisfying the objective

The objective of the end user is to deliver a specified amount of a given gas. Refer to Figure 3.2.1 and note that his objective can best be stated by the relationship:

$$\text{Gas Flow Produced} = \text{Gas Flow Delivered.}$$

This, incidentally, is the reason why most process control systems monitor pressure in the process system and install a controller to either modulate flow via a control valve (change the head required by the process) or vary the speed of the compressor (change the head produced by the compressor).

The vendor then determines the head required by the process, on the basis of the parameters given by the contractor and end user on the equipment data sheet. It is very important to note that all possible sources should be used to confirm that the conditions stated on the data sheet are correct and realistic. This fact is especially true for dynamic compressors, since erroneous process conditions will impact the throughput of the compressor.

Gas characteristics

Figure 3.2.2 presents the relationships which are used to calculate the design parameters for the compressor. Note that the same relationships are used regardless of the type of compressor (positive displacement or dynamic) under consideration.

The gas characteristics that are used in the determination of design parameters are defined in Table 3.2.1.

Table 3.2.2 shows some useful relationships that are used in compressor calculations, together with the definitions of the constants used.

Compression head

The ideal gas head equations are again defined in Figure 3.2.3. As previously stated, polytropic head is the usual choice among compressor vendors.

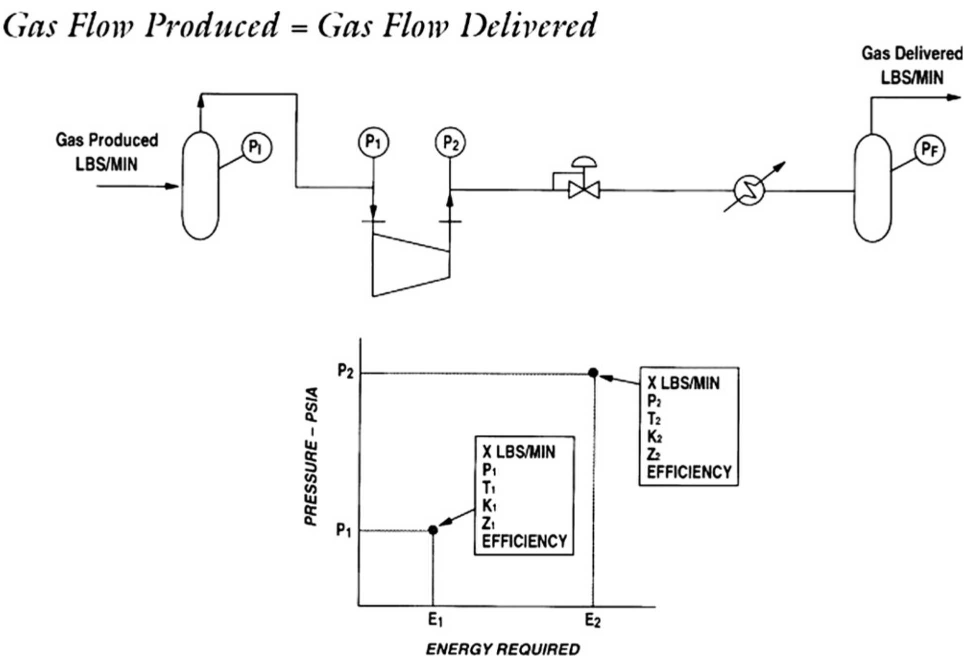


Fig 3.2.1 • The objective: to deliver a specified amount of a given gas

Impeller types and specific speed

Various types of radial (centrifugal) impellers are shown in Figures 3.2.4 and 3.2.5.

Open impellers

Some open impellers are shown in Figure 3.2.4. They can operate at higher tip speeds, and thus produce greater head than closed impellers. Open impellers can produce approximately 4,500 – 7,500 m-kg force/kg mass (15,000 – 25,000 ft-lbs force/lb mass) of head per stage. This is because a side plate is not attached to the inlet side of the vanes, which results in significantly lower blade stresses. The disadvantages of open impellers are their lower efficiency, due to increased shroud (front side) leakage and increased number of blade natural frequencies resulting from the cantilevered attachment of the blades to the hub. Most end users restrict the use of open impellers to plant and instrument air applications, since the high speeds and intercooling offset the efficiency penalties caused by shroud leakage. Older designs of multistage centrifugal compressors frequently used open impellers in the first stages, since the high flows caused unacceptable side plate stresses in closed impeller design. Modern calculation (finite element) methods and manufacturing methods (attachment techniques – machine welding, brazing, etc.) now make possible the use of enclosed first stage impellers for all multistage compressor applications. Finally, radial bladed impellers (whether open or enclosed) produce an extremely flat (almost horizontal) head curve.

This characteristic renders these impellers unstable in process systems that do not contain much system resistance. Therefore, radial impellers are to be avoided under these circumstances (plant and instrument air compressors, charge

To achieve the client's objective the compressor vendor must calculate the actual flow to the compressor inlet, the actual energy and work required.

Actual flow

Volume flow rate m^3/hr (ft^3/min) = mass flow rate kg/hr (lb/min) · density kg/m^3 (lb/ft^3) $\left\{ \text{gas density depends on } P_1, T_1, Z, MW \right.$

Energy (ideal) Compression HEAD$_{\text{POLYTROPIC}}$ = m-kgf kgm $\left(\frac{ft-lbs}{lb\ mass}\right)$ Energy (ideal) to compress and deliver one lb of gas from P_1 to P_2 $\left\{ \text{Depends on } P_1, T_1, Z_{avg}, K_{avg}, MW\ P_2, \text{efficiency} \right.$

Work

$$\text{Power kW (hp)} = \frac{\text{ideal energy } \frac{m-kgf}{kgm}\left(\frac{ft-lbs}{lb\ mass}\right) \text{ mass flow } \frac{kg}{hr}\left(\frac{lb}{min}\right)}{3600\,\frac{m-kgf}{hr-kW}\left(33{,}000\left(\frac{ft-lbs}{min\text{-}hp}\right)\right) \text{ x efficiency (\%)}}$$

Fig 3.2.2 •

Table 3.2.1 Gas characteristics

Compressibility (Z)	– Accounts for the deviation from an ideal gas
Specific heat (C)	– The amount of heat required to raise one mass of gas one degree
C_P and C_V	– Specific heat at constant pressure and volume respectively
Specific heat ratio (K)	– C_P/C_V
MW	– Molecular weight
Polytropic exponent (n)	– Used in polytropic head calculation $\frac{n-1}{n} = \frac{k-1}{k} \times \frac{1}{\eta \text{ polytropic}}$

Table 3.2.2 Useful relationships

Actual flow – m^3/hr (ft^3/min)	where:
m^3/hr (acfm) = $\frac{\text{mass flow kg/hr (lbs/min)}}{\text{density kg/m}^3\text{ (lbs/ft}^3\text{)}}$	$C = 3{,}600 = \frac{m - kgf}{hr - kW}\left(\frac{\text{ft-lbs}}{\text{Min-H.P.}}\right)$
Density kg/m^3 (lbs/t^3) $= \frac{(P)}{ZRT}$	$HD = \frac{HEADm - kgf}{kgm}\left(\frac{ft - lbs}{lb}\right)$
acfm = m3/hr Nm3/hr x (101) (T) P 289 (scfm x (14.7) (T)) P 520	Massflow $= \frac{kg}{hr}\left(\frac{lb}{min}\right)$
Energy (ideal) – $\frac{m - kgf}{kgm}$(ft – lb/lb mass)	Eff' y = corresponding efficiency (polytropic, isentropic, etc)
Use head equation, polytropic is usually used	P = pressure – kPaa (psia)
Efficiency – %	T = temperature – K (R*)
Derived from impeller test results – does not include mechanical losses	K = °C + 273.1 (*R = °F + 460) Z = compressibility
Work – kW (horsepower)	R = 1545/mol. wgt
Brake power = gas power + mech. losses	Nm^3/hr = Normal m^3/hr referenced to 17°C and 101 kPA (scfm = standard FT^3/min referenced to 60°F and 14.7 psia)
Gas power $= \frac{(HD)(\text{mass flow})}{(C)(\text{eff'y})}$	

IDEAL GAS EQUATIONS

Isothermal Head

$$\frac{HD}{\begin{array}{c}M - Kgf/kgm\\(FT - Lbf/Lbm)\end{array}} = \frac{847.4}{MW}\left(\frac{1545}{MW}\right)(T_1)\left(Z_{AVG}\right)\left[L_N\left(\frac{P_2}{P_1}\right)\right]$$

Isentropic (Adiabatic) Head

$$\frac{HD}{\begin{array}{c}M - Kgf/kgm\\(FT - Lbf/Lbm)\end{array}} = \frac{847.4}{MW}\left(\frac{1545}{MW}\right)(T_1)\left(\frac{K}{K-1}\right)\left(Z_{AVG}\right)\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{K-1}{K}\right)} - 1\right]$$

Polytropic Head

$$\frac{HD}{\begin{array}{c}M - Kgf/kgm\\(FT - Lbf/Lbm)\end{array}} = \frac{847.4}{MW}\left(\frac{1545}{MW}\right)(T_1)\left(\frac{n}{n-1}\right)\left(Z_{AVG}\right)\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]$$

Where:

- $\frac{847.4}{MW}$ = *Metric Gas Constant "R"*
- $\frac{1545}{MW}$ = *Customory Gas Constant "R"*
- MW = *Molecular weight*
- T_1 = *Inlet Temperature °K or °R*
- °K = *273.1 + °C*
- °R = *460.0 + °F*
- Z_{AVG} = *Average Compressibility* $\left(\frac{Z_1 + Z_2}{2}\right)$
- K = *Ratio of Specific Heats* C_p / C_v
- $\frac{n-1}{n}$ = *Polytropic Exponent* $= \left(\frac{K-1}{K}\right)\left(\frac{1}{\eta\, Poly}\right)$
- Poly η = *Polytropic Efficiency*
- Ln = *Log to base A*
- P_1 = *Suction Pressure KPaa (PSIA)*
- P_2 = *Dischrage Pressure KPaa (PSIA)*

Fig 3.2.3 • Ideal gas head equations

gas compressors and refrigeration applications with side loads).

Enclosed impellers

Enclosed impellers are shown in Figure 3.2.5.

Note that the first stage impeller in any multistage configuration is always the widest. That is, it has the largest flow passage. As a result, the first stage impeller will usually be the highest stressed impeller. The exception is a refrigeration compressor with side loads (economizers).

Dynamic compressor vendors use a specific speed to select impellers, based on the data given by the contractors and end user. The vendor is given the total head required by the process and the inlet volume flow. As previously discussed, at the stated inlet flow (rated flow) the head required by the process is in equilibrium with the head produced by compressor. Vendor calculation methods then determine how many compressor impellers are required, on the basis of the mechanical limitations (stresses) and performance requirements (quoted overall efficiency). Once the head required per stage is determined, the compressor speed is optimized for highest possible overall efficiency using the concept of specific speed as shown in Figure 3.2.6.

It is a proven fact that the larger the specific speed, the higher the attainable efficiency. As shown, specific speed is a direct function of shaft speed and volume flow and an inverse function of produced head. Since the vendor knows, at this point in the design, the volume flow and head produced for each impeller, increasing the shaft speed will increase the specific speed and the compressor efficiency.

However, the reader is cautioned that all mechanical design aspects (impeller stress, critical speeds, rotor stability, design of bearings and seals) must be confirmed prior to acceptance of impeller selection. Often, too great an emphasis on

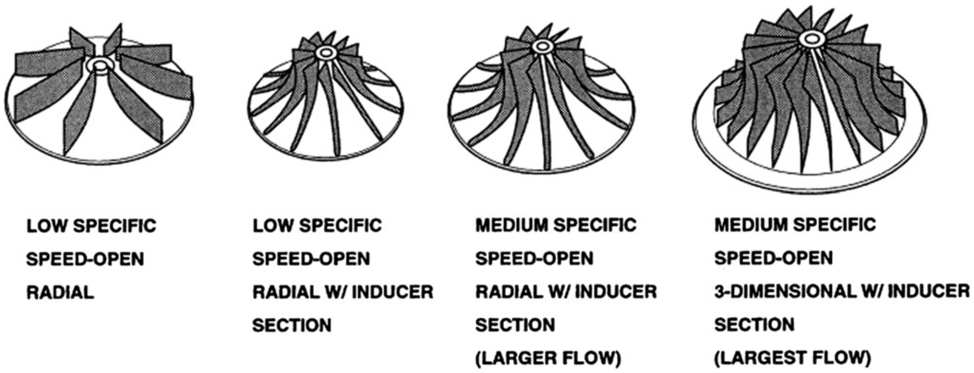


Fig 3.2.4 • Compressor impellers

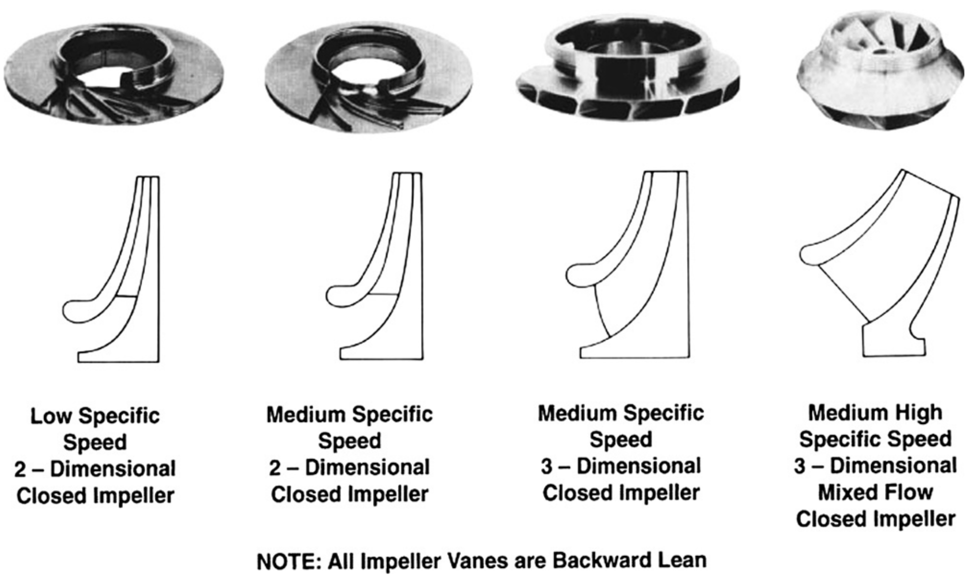


Fig 3.2.5 • Enclosed impellers (Courtesy of IMO Industries, Inc.)

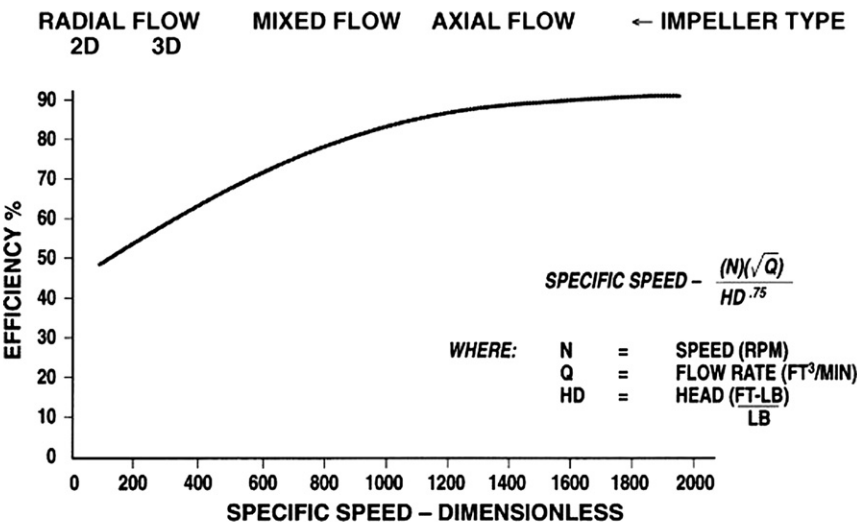


Fig 3.2.6 • Impeller geometry vs. specific speed

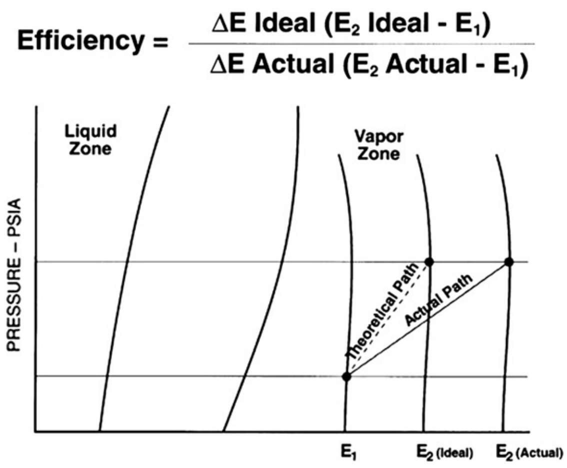


Fig 3.2.7 • Efficiency

performance (efficiency) results in decreased compressor reliability. One mechanical design problem can quickly offset any power savings realized by designing a compressor for a higher efficiency.

Referring back to Figure 3.2.6, calculation of specific speed for the first impeller by the contractor or end user will give an indication of the type of dynamic compressor blading to be used. One other comment; Sundstrand Corporation successfully employs an integral high speed gear box design for low flow, high head applications or for low specific speed applications.

The use of a speed increasing gear box (for speeds up to 34,000 RPM) enables the specific speed to be increased, and therefore gives higher efficiency and less complexity than would be obtained with a multistage compressor design approach.

Efficiency

Compressor efficiency, regardless of the type of compressor, can best be understood by referring to a typical Mollier Diagram as depicted in Figure 3.2.7.

All produced heads shown on performance curves (isothermal, isentropic and polytropic) represent the ideal reversible head produced to compress a given gas from P_1 to P_2. This then is the theoretical compression path of the gas. That is, the energy required to compress a gas if the efficiency is 100%.

However due to friction, sudden expansion etc., the efficiency is less than 100%. Therefore the actual compression path requires more head (energy) to compress the gas from P_1 to P_2. The efficiency then is equal to:

$$\text{Efficiency} = \frac{\Delta E\ \text{Ideal}\ (E_2\ \text{Ideal-}E_1)}{\Delta E\ \text{Actual}\ (E_2\ \text{Actual-}E_1)}$$

Note that () is used to represent any ideal reversible path (isothermal, isentropic, polytropic).

Horsepower

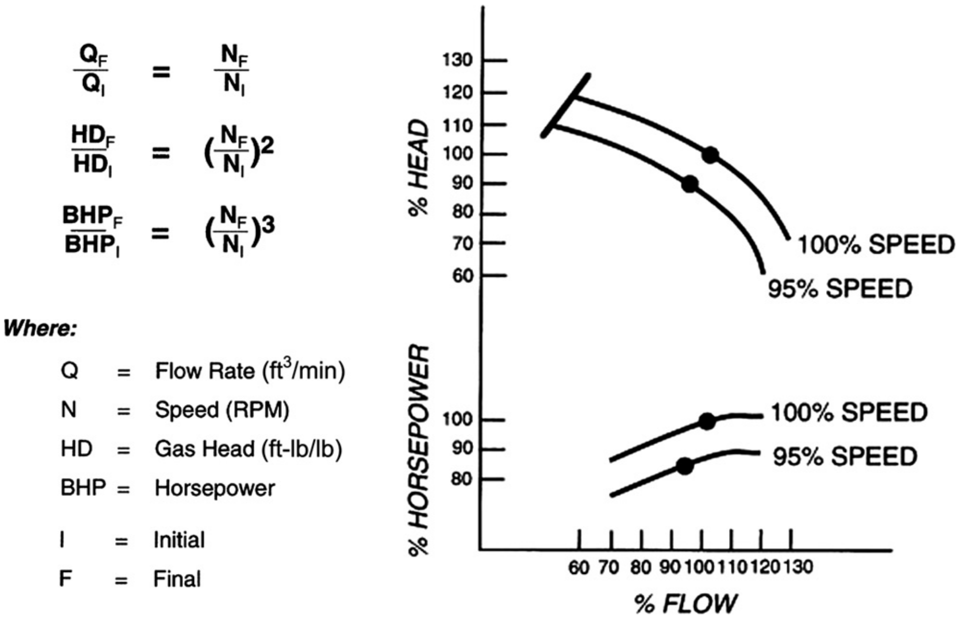
Gas Horsepower is defined as the total actual energy (work) required to compress a given gas from P_1 to P_2 when compressing a given mass flow:

$$\text{Head}()\frac{\text{m}-\text{kgf}}{\text{kgm}}\left(\frac{\text{ft}-\text{lbf}}{\text{lbm}}\right)(\text{MassFlow}-\text{kg/hr(lb/min)})$$

3,600 (33,000) (Efficiency) ()

Note: () must be for the same ideal reversible compression path. The brake horsepower is the sum of the gas horsepower and the mechanical losses of the compressor.

Fig 3.2.8 • The Fan Laws



$$B.H.P. = G.H.P. + \text{Mechanical losses}$$

The mechanical losses are the total of bearing, seal and windage (disc friction) losses and are provided by the compressor vendor. For estimating purposes, a conservative value of mechanical losses for one centrifugal or axial compressor case would be 112 kW (150 H.P.).

The Fan Laws

These familiar relationships, sometimes called the affinity laws for pumps were originally derived for a single stage fan which is a low pressure compressor. The Fan Laws are presented in Figure 3.2.8.

As shown, if the speed is changed, the flow, head and horsepower vary by the first, second and third power of speed ratio respectively.

The reader must be cautioned however that the Fan Laws are only an approximation; hence they should only be used as an estimating tool. Their accuracy decreases significantly with increasing gas molecular weight and increase in the number of compression stages.

Best Practice 3.3

Pre-select centrifugal compressor casing type, impeller type, the number of compressor cases and impellers in each casing.

Pre-selection of centrifugal compressor casing type and impellers ensures optimum safety and reliability.

Determine if a horizontal split casing or vertical (barrel) type is required based on process conditions and vendor/company/industry guidelines and plant lessons learned.

Determine the impeller type (opened or closed) base on company/industry guidelines and plant lessons learned.

Determine the number of impellers allowed in each casing based on head per impeller stage limits and shaft stiffness (see B.P.s 3.9 and 3.18).

Lessons Learned

There are many case histories of failure because centrifugal compressors were not selected using the proper case design and limiting the number of impellers per stage.

Benchmarks

I have used this best practice since the mid-1970s to achieve success in all centrifugal compressor installations resulting in plant installations of greater than 99.7% centrifugal compressor reliability, without:

- Critical speed issues
- Gas instabilities (gas whirl and whip)
- Impeller failures
- Excessive factory acceptance test (FAT) time.

B.P. 3.3. Supporting Material

Centrifugal multi-stage horizontal split

A typical multi-stage horizontally split centrifugal compressor is shown in Figure 3.3.1. The casing is divided into upper and lower halves along the horizontal centerline of the compressor. The horizontal split casing allows access to the internal components of the compressor without disturbing the rotor to casing clearances or bearing alignment. If possible, piping nozzles should be mounted on the lower half of the compressor casing to allow disassembly of the compressor without removal of the process piping.

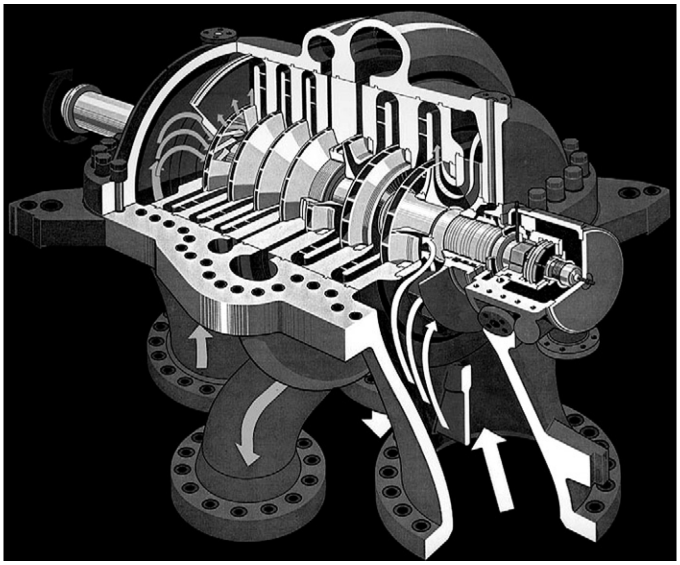


Fig 3.3.1 • Centrifugal multi-stage horizontal split (Courtesy of Mannesmann Demag)

Centrifugal multi-stage with side loads

This type of compressor is used exclusively for refrigeration services. The only difference from the compressor shown in Figure 3.3.1 is that gas is induced or removed from the compressor via side load nozzles. A typical refrigeration compressor

is shown in Figure 3.3.2. Note that this type of compressor can be either horizontally or radially split.

Centrifugal multi-stage (barrel)

A typical multi-stage, radially-split, centrifugal compressor is shown in Figure 3.3.3. The compressor casing is constructed as a complete cylinder with one end of the compressor removable to allow access to the internal components. Multi-stage, radially-slit centrifugal compressors are commonly called barrel compressors.

Impeller types and specific speed

Various types of radial (centrifugal) impellers are shown in Figures 3.3.4 and 3.3.5.

Open impellers

Open impellers are shown in Figure 3.3.4. They have the advantage of being able to operate at higher tip speeds and thus produce greater head than closed impellers. Open impellers can produce 15,000 – 25,000 ft-lbs/lb of head per stage. This is because a side plate is not attached to the inlet side of the vanes, which results in significantly lower blade stresses. The disadvantages of open impellers are their lower efficiency due to increased shroud (front side) leakage, and increased number of blade natural frequencies resulting from the cantilevered attachment of the blades to the hub.

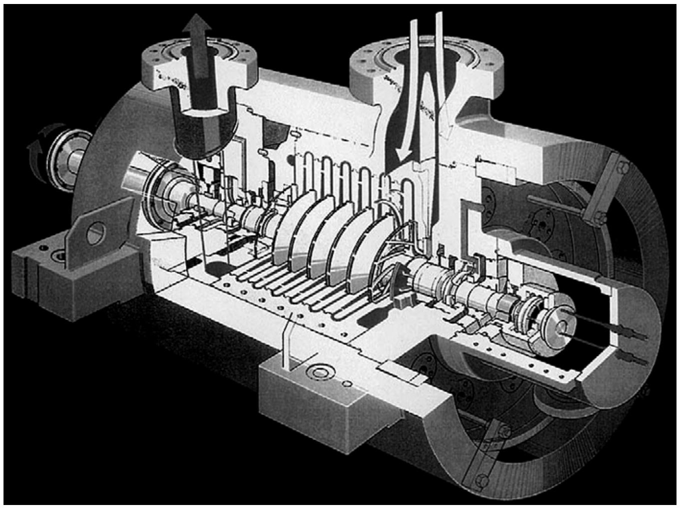


Fig 3.3.3 • Typical multi-stage, radially split centrifugal compressor (Courtesy of Mannesmann Demag)

Most end users restrict the use of open impellers to plant and instrument air applications since the high speeds and intercooling offset the efficiency penalties caused by shroud leakage. Older design multistage centrifugal compressors frequently used open impellers in the first stages since the high flows caused unacceptable side plate stresses in closed impeller design. Modern calculation (finite element) methods and manufacturing methods (attachment techniques – machine

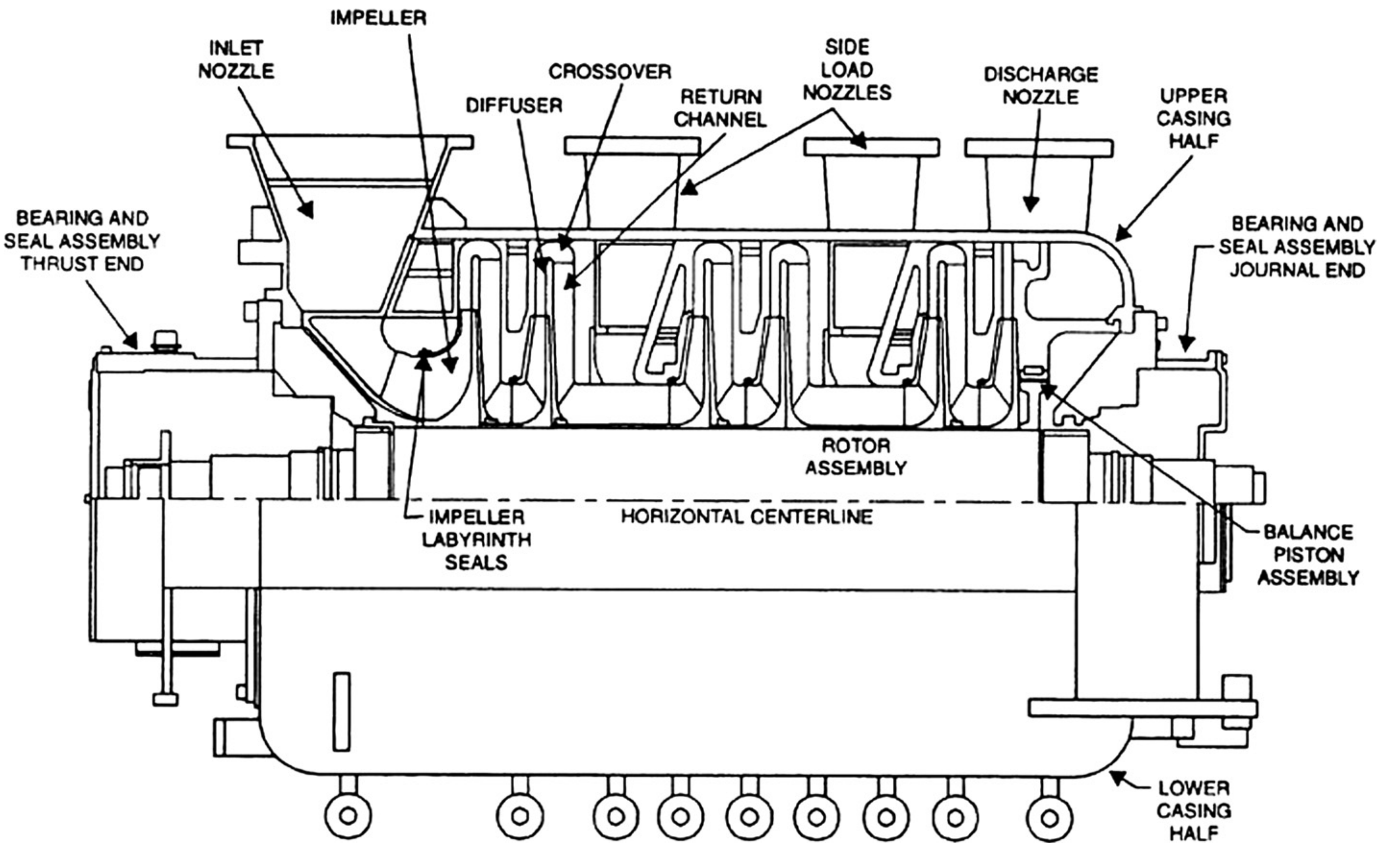


Fig 3.3.2 • Typical multi-stage refrigeration compressor

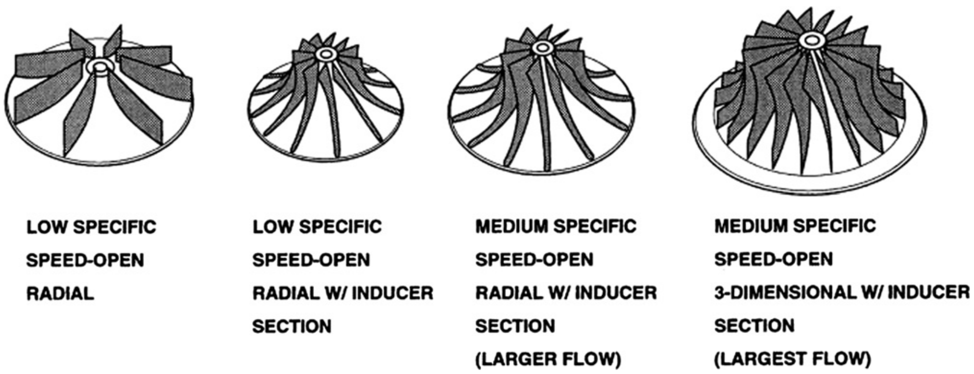


Fig 3.3.4 • Compressor impellers

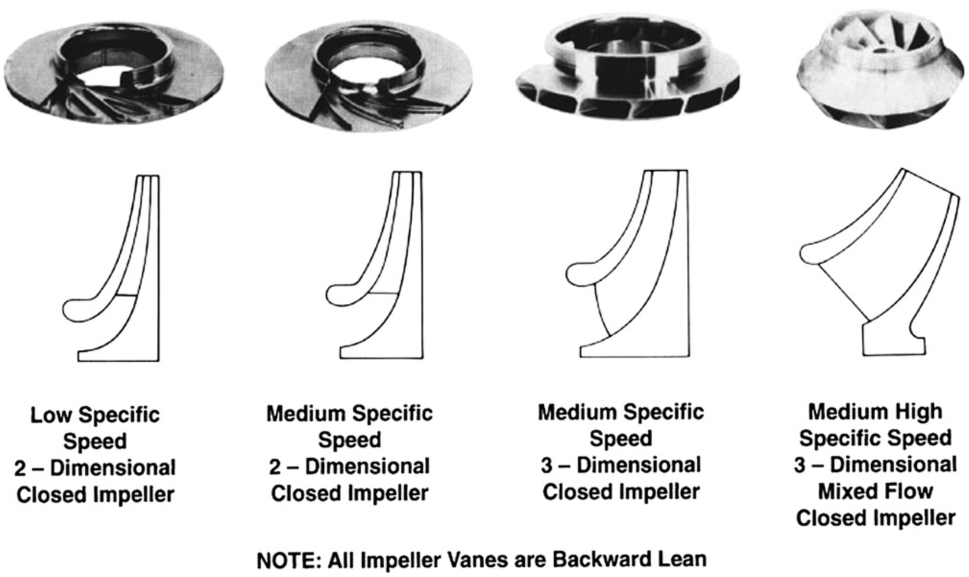


Fig 3.3.5 • Enclosed impellers (Courtesy of IMO Industries, Inc.)

welding, brazing, etc.) today make possible the use of enclosed first stage impellers for all multistage compressor applications. Finally, radial bladed impellers (whether open or enclosed) produce an extremely flat (almost horizontal) head curve. This characteristic renders these impellers unstable in process systems that do not contain much system resistance. Therefore, radial impellers are to be avoided in process systems that do not contain much system resistance (plant and instrument air compressors, charge gas compressors and refrigeration applications with side loads).

Enclosed impellers

Enclosed impellers are shown in Figure 3.3.5.

Note that the first stage impeller in any multistage configuration is always the widest. That is, it has the largest flow passage. As a result, the first stage impeller will usually be the highest stressed impeller. The exception is a refrigeration compressor with side loads (economizers).

Dynamic compressor vendors use a specific speed to select impellers based on the data given by the contractors and end user. The vendor is given the total head required by the process and the inlet volume flow. As previously discussed, at the stated inlet flow (rated flow) the head required by the process is in equilibrium with the head produced by the compressor.

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However, the reader is cautioned that all mechanical design aspects (impeller stress, critical speeds, rotor stability, bearing and seal design) must be confirmed prior to acceptance of impeller selection. Often, too great an emphasis on performance (efficiency) results in decreased compressor reliability. One mechanical design problem can quickly offset any power

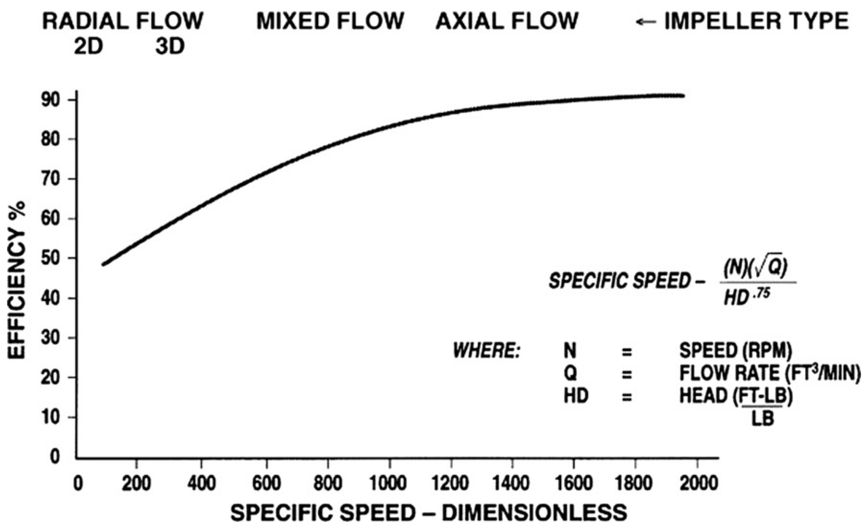


Fig 3.3.6 • Impeller geometry vs. specific speed

savings realized by designing a compressor for a higher efficiency.

Referring back to Figure 3.3.6, the calculation of specific speed for the first impeller by the contractor or end user will give an indication of the type of dynamic compressor blading to be used. One other comment; Sundstrand Corporation successfully employs an integral high speed gear box design for low flow, high head applications or for low specific speed applications. The use of a speed increasing gear box (for speeds up to 34,000 RPM) enables the specific speed to be increased, and therefore resulting in higher efficiency and less complexity than would be obtained with a multistage compressor design approach.

Critical speeds and rotor response

The term 'critical speed' is often misunderstood. In nature, all things exhibit a natural frequency, which is defined as that frequency at which a body will vibrate if excited by an external force. The natural frequency of any body is a function of the stiffness and the mass of that body. As mentioned, for a body to vibrate, it must be excited. A classical example of natural frequency excitation is the famous bridge 'Galloping Gerty' in the state of Washington, which vibrated to destruction when its natural frequency was excited by prevailing winds.

In the case of turbo-compressor rotors, their natural frequency must be excited by some external force to produce a response that will result in increased amplitude of vibration. One excitation force that could produce this result is the speed of the rotor itself, which gives rise to the term 'critical speeds'. The term 'critical speed' defines the operating speed at which a natural frequency of a rotor system will be excited. All rotor systems have both lateral (horizontal and vertical) and torsional (twist about the central shaft axis) natural frequencies. Only lateral critical speeds will be discussed in this section.

In the early days of rotor design, it was thought that the rotor system consisted primarily of the rotor supported by the bearings. This led to the assumption that only the stiffness of the rotor supported by rigid bearings needed to be considered in the analysis of the natural frequency. Countless machinery problems have proven this assumption to be false over the years. The concept of the 'rotor system' must be thoroughly understood. The rotor system consists of the rotor itself, the characteristics of the oil film that support the rotor, the bearing, the bearing housing, the compressor case that supports the bearing, compressor support (base plate), and the foundation. The stiffness and damping characteristics of all of these components together result in the total rotor system that produces the rotor response to excitation forces.

We will examine a typical rotor response case in this section and note the various assumptions, the procedure modeling, the placement of unbalance, the response calculation output, and discuss the correlation of these calculations to actual test results.

Critical speeds

The natural frequency of any object is defined by the relationship:

$$F_{NATURAL} = \sqrt{\frac{K}{M}}$$

Where: K = Stiffness

M = Mass

When excited by an external force, any object will vibrate at its natural frequency. If the frequency of the exciting force is equal to the natural frequency of the object, and no damping is present, the object can vibrate to destruction. Therefore, if the frequency of an exciting force equals the natural frequency of an object, the exciting force is operating at the 'critical frequency'.

Rotor speed is one of the most common external forces in turbo-machinery. When the rotor operates at any rotor system natural frequency, it is said that the rotor is operating at its critical speed. The critical speed of a rotor is commonly designated as NC and the corresponding natural frequencies or critical speeds are: NC_1, NC_2, NC_3, etc.

Every turbo-compressor must have its rotor system's critical speeds determined prior to manufacture. In this section, we will follow the procedure for the determination of the necessary parameters to define a rotor system's critical speed. The procedure is commonly known as determination of rotor response. Figure 3.3.7 is a representation of a critical speed map for a rotor system.

It should be understood that all stiffness values are 'calculated' and will vary under conditions of actual use. As an exercise, determine NC_1, NC_2 and NC_3 for the horizontal and vertical directions for each bearing in Figure 3.3.7 (assume bearing 1 and 2 stiffness are the same).

Critical speed	Horizontal (X)	Vertical (Y)
NC_1	3,300 rpm	3,000 rpm
NC_2	9,700 rpm	8,000 rpm
NC_3	16,000 rpm	15,000 rpm

Based on a separation margin of ± 20% from a critical speed, what would be the maximum allowable speed range between NC_1 and NC_2 in Figure 3.3.7?

- Maximum speed 6,600 rpm
- Minimum speed 4,000 rpm

Remember, changing of any value of support stiffness will change the critical speed. Support stiffness in lbs/inch is plotted on the x axis. The primary components of support stiffness, in order of decreasing increasing influence are:

- Oil support stiffness
- Bearing pad or shell
- Bearing housing
- Bearing bracket
- Casing support foot
- Baseplate
- Foundation

Note that this analysis of the critical speed does not include oil film damping. It is common practice to first determine the 'undamped critical speeds' to allow for necessary modifications to the rotor or support system. This is because the effects of stiffness on the location of critical speed are significantly greater than damping. Figure 3.3.7 shows four (4) distinct critical speeds. Operation within ± 20% of actual critical speeds is to be

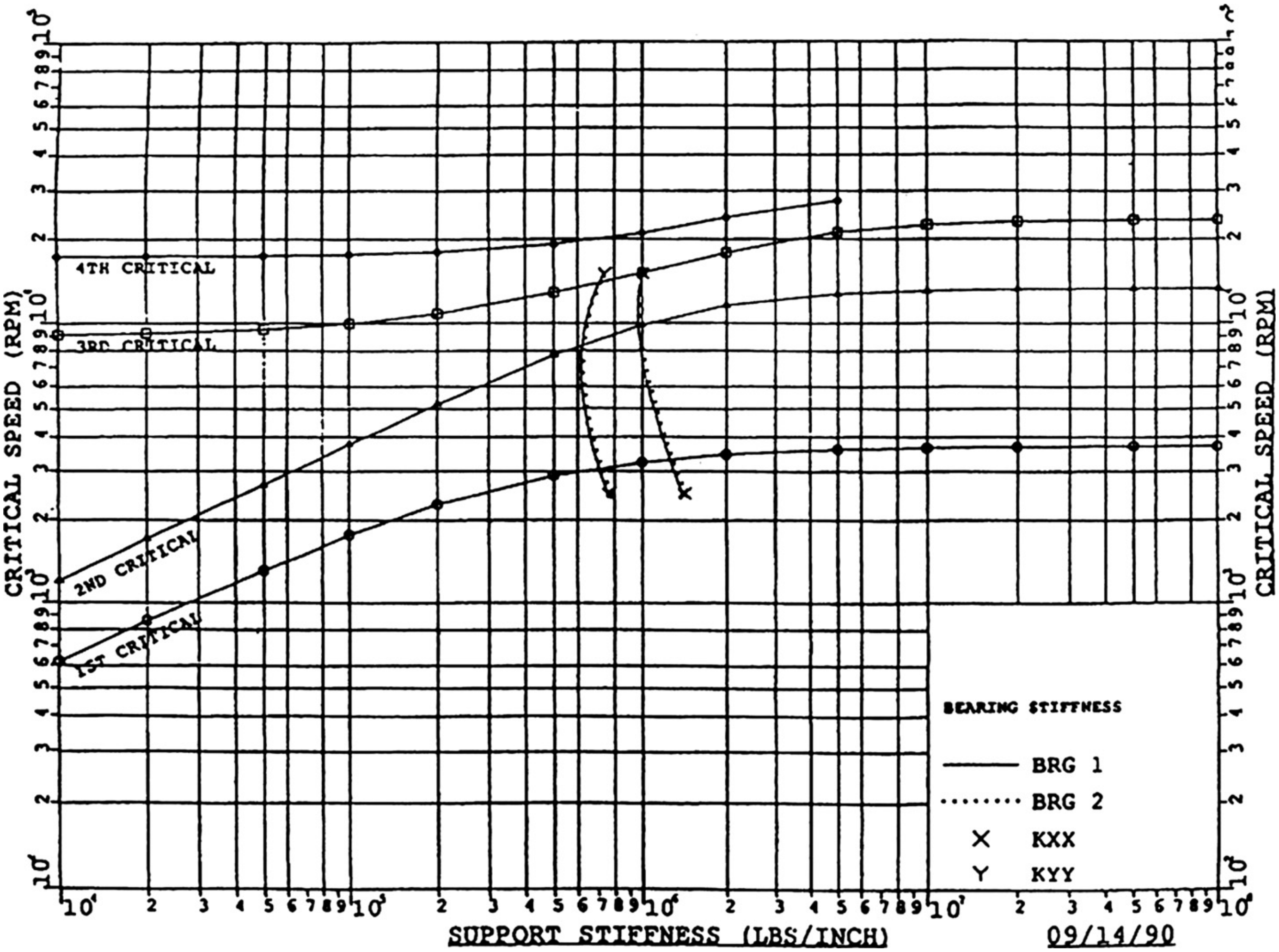


Fig 3.3.7 • Compressor rotor critical speed map – no damping (Courtesy of Elliott Company)

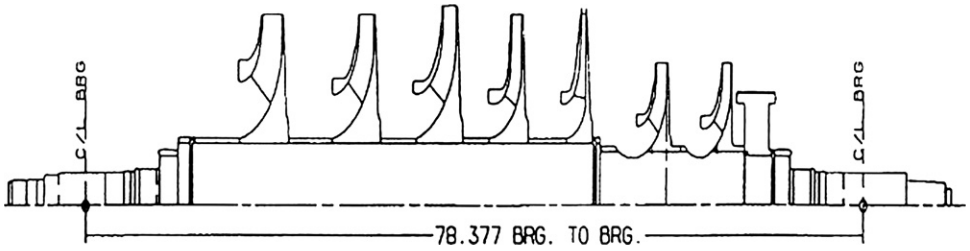


Fig 3.3.8 • Rotor response modeling – rotor (Courtesy of Elliott Co.)

avoided. Also plotted are the horizontal (x) and vertical (y) bearing stiffness for each bearing. Note that these values vary with speed and are the result of changes in the oil stiffness. Therefore, a change in any of the support stiffness components noted above can change the rotor critical speed. Experience has shown that critical speed values seldom change from ± 5% of their original installed values.

If a turbo-compressor with oil seals experiences a significant change in critical speeds, it is usually an indication of seal lock-up. That is, the seal does not have the required degrees of freedom and supports the shaft acting like a bearing. Since the seal span is less than the bearing span, the rotor stiffness 'K' increases and the critical speeds will increase in this case.

The rotor system (input)

Figure 3.3.8 shows a typical turbo-compressor rotor before modeling for critical speed or rotor response analysis.

Since the natural frequency or critical speed is a function of shaft stiffness and mass, Figure 3.3.9 presents the rotor in Figure 3.3.8 modeled for input to the computer rotor response program. Figure 3.3.9 is an example of a modeled rotor and only includes the rotor stiffness (K) and mass (M).

In order to accurately calculate the rotor critical speeds, the entire rotor system stiffness, masses and damping must be considered. Table 3.3.1 models the oil film stiffness and damping of the journal bearings at different shaft speeds.

Note that it is essential that viscosity characteristics of the type of oil to be used in the field must be known. End users should consult with the OEM before changing the oil type as this will affect the rotor response. In addition to modeling the rotor and bearings, most rotor response calculations also include the following additional inputs:

- Bearing support stiffness
- Oil film seal damping effects

Of all the input parameters, the effects of bearing and seal oil film parameters are the most difficult to calculate and measure. Therefore, a correlation difference will always exist between the predicted and actual values of critical speed. Historically, predicted values of NC_1 (first critical speed) generally agree within ± 5%. However, wide variations between predicted and actual values above the first critical speed (NC_1) exist for NC_2, NC_3, etc.

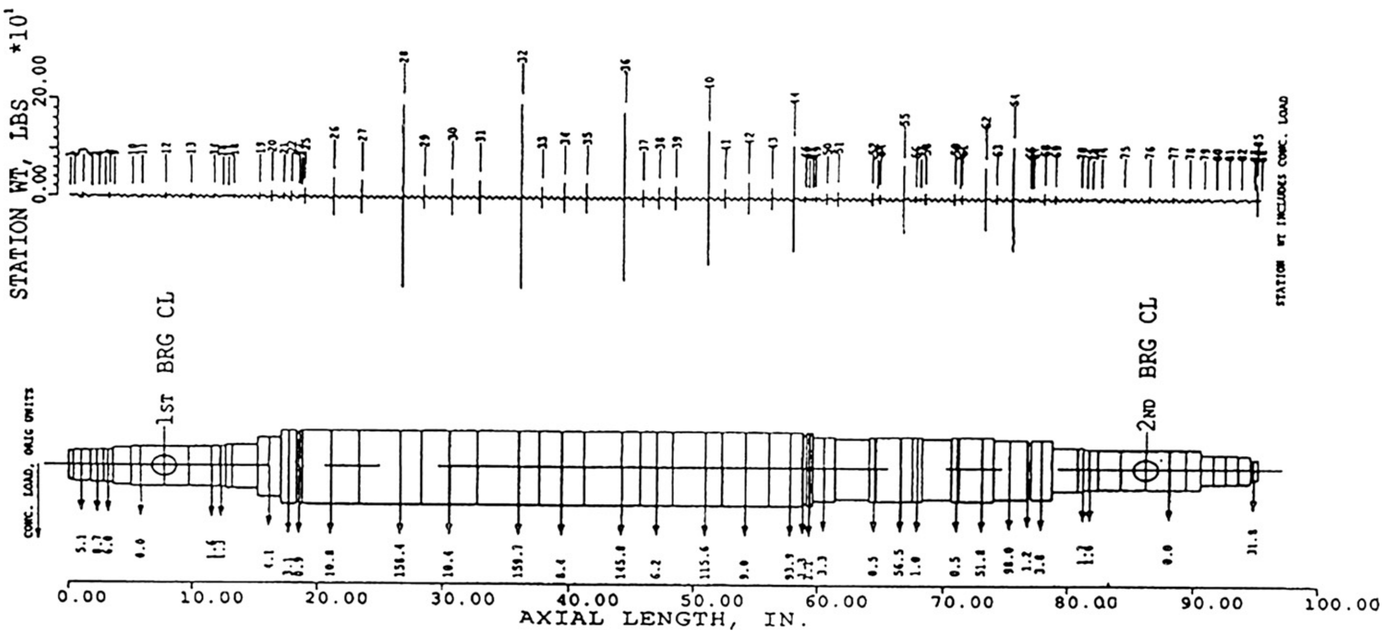


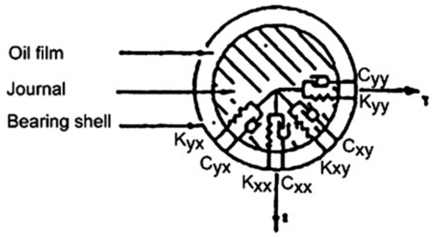
Fig 3.3.9 • Rotor response input data – dimensions, masses and unbalances (Courtesy of Elliott Co.)

Table 3.3.1 Typical compressor oil film bearing parameters (Courtesy of Elliott Co.)

4 × 1.6"tilt 20.5" TB 3.0" shaftend 7.5–6.5" shaft Bendix coupling

Static bearing load (lbs)	897	diameter (inches)	4.00
Bearing station	12	length (inches)	1.60
Bearing location	thrust	diam assembly clearance (inches)	5.7487E–03
Bearing type	tilt pad	diam machined clearance (inches)	8.7500E–03
Location of load	between pads	inlet oil temperature (deg F)	120.0
Preload	0.343	type of oil	DTE–light (150SSU @100°F)

Speed (rpm)	50mm No	Fluid film stiffness KXX (lb/in)	KYY (lb/in)	Damping WCXX (lb/in)	WCYY (lb/in)
2500	0.114	1.3871E 06	7.5446E 05	7.7995E 05	4.6249E 05
3000	0.137	1.2984E 06	7.1330E 05	7.8487E 05	4.7587E 05
4000	0.183	1.1769E 06	6.6147E 05	8.0311E 05	5.0825E 05
4500	0.206	1.1341E 06	6.4543E 05	8.1400E 05	5.2564E 05
5500	0.252	1.0703E 06	6.2556E 05	8.3686E 05	5.6116E 05
6613	0.303	1.0230E 06	6.1679E 05	8.6656E 05	6.0354E 05
7000	0.321	1.0109E 06	6.1616E 05	6.7775E 05	6.1885E 05
8000	0.366	9.8751E 05	6.1898E 05	9.0798E 05	6.5935E 05
9000	0.412	9.7305E 05	6.2684E 05	9.4015E 05	7.0111E 05
10000	0.458	9.6556E 05	6.3864E 05	9.7461E 05	7.4430E 05
11000	0.504	9.6360E 05	6.5354E 05	1.0110E 06	7.8878E 05
12000	0.549	9.6610E 05	6.7094E 05	1.0490E 06	8.3434E 05
13000	0.595	9.7225E 05	6.9037E 05	1.0881E 06	8.8080E 05
14000	0.641	9.8144E 05	7.1149E 05	1.1283E 06	9.2801E 05
15000	0.687	9.9317E 05	7.3403E 05	1.1696E 06	9.7586E 05



When selecting machinery, the best practice is to request specific vendor experience references for installed equipment with similar design parameters as follows:

- Bear span ÷ major shaft diameter
- Speeds
- Bearing design
- Seal design
- Operating conditions (if possible)

Once the rotor system is adequately modeled, the remaining input parameter is the amount and location of unbalance. Since the objective of the rotor response study is to accurately predict the critical speed values and responses, an assumed value and location of unbalances must be defined. Other than bearing and seal parameters, unbalance amount and location is the other parameter with a 'correlation factor'. There is no way to accurately predict the amount and location of residual unbalance on the rotor. Presently, the accepted

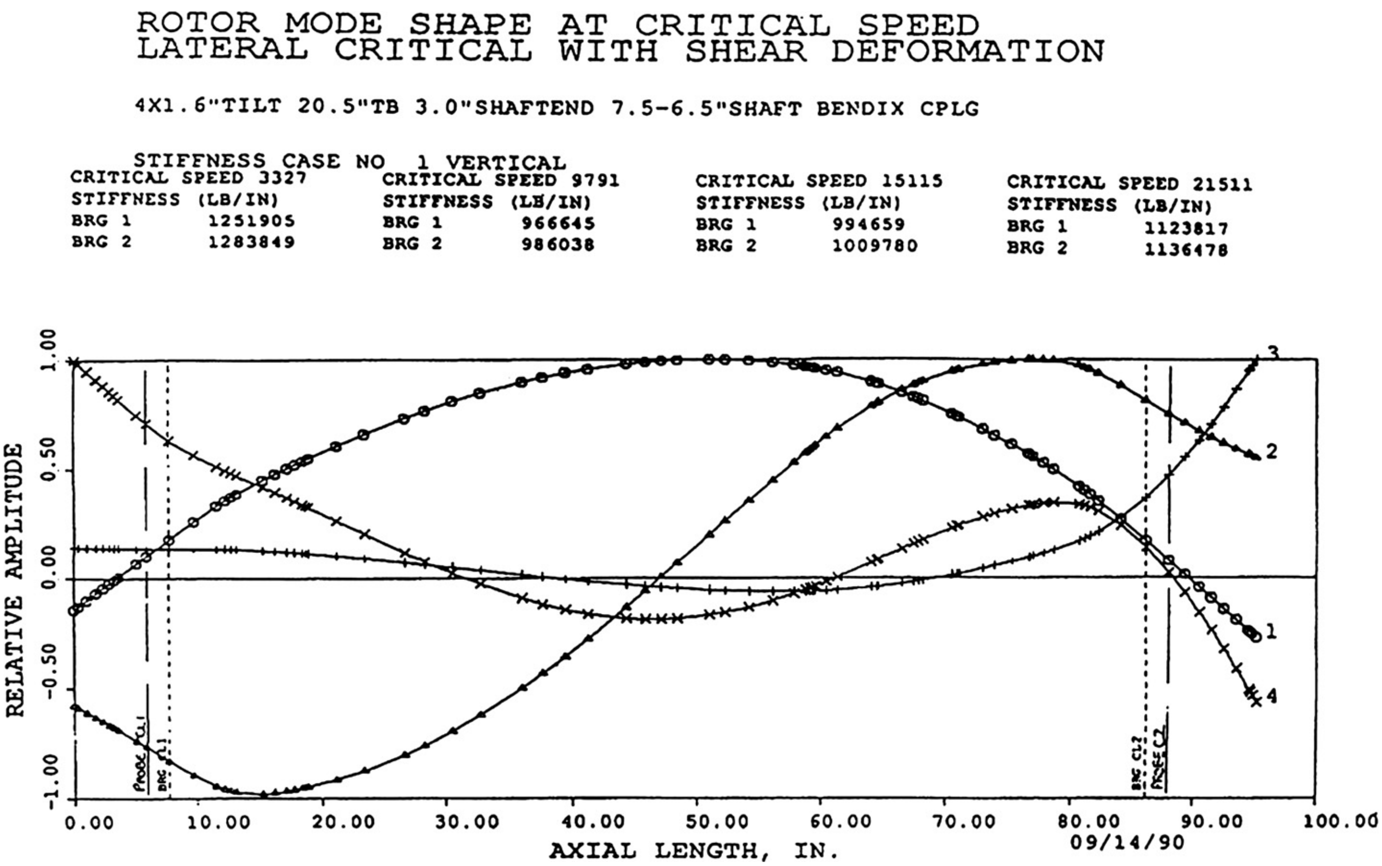


Fig 3.3.10 • Rotor natural frequency mode shapes (Courtesy of Elliott Co.)

method is to input a value of 8 × A.P.I. acceptable unbalance limit $\frac{(4W)}{N}$.

This results in a rotor response input unbalance of $\frac{32W}{N}$.

The location of the unbalance is placed to excite the various critical speeds. Typically the unbalances are placed as noted below:

Location	To excite
Mid span	NC_1
Quarter span (2 identical unbalances)	NC_2
At coupling	NC_2, NC_3

Failure to accurately determine the value and location of residual rotor unbalance is one of the major causes of correlation differences between predicted and actual critical speeds.

Rotor response (output)

The output from the rotor response study yields the following:

- Relative rotor mode shapes
- Rotor response for a given unbalance

Figure 3.3.10 shows the relative rotor mode shapes for NC_1, NC_2, NC_3 and NC_4. Usually, the rotor will operate between NC_1 and NC_2.

Rotor mode shape data is important to the designer because it allows determination of modifications to change critical speed values.

For the end user, this data provides an approximation of the vibration at any point along the shaft as a ratio of the measured vibration data. As an example in Figure 3.3.10, determine the vibration at the shaft mid span if the vibration measured by the probe C_2 when operating at NC_1 is 2.00 mils. From Figure 3.3.10, the vibration at the shaft mid span when operating at the first critical speed of 3327 RPM (50 in location) is:

$$\frac{1.00}{.1} \text{ or } 10 \ \times \text{the bearing vibration}$$

Ten (10) times the value at C_2 or 20.0 mils!

Mode shape data should always be referred to when vibration at operating speed starts to increase and your supervisor asks

'When do we have to shut down the unit?'

or

'Can we raise the radial vibration trip setting?'

In this example, the bearing clearance may be 0.006 or 6 mils, and an honest request would be 'We'll replace the bearing at the turnaround, please run to 7.0 mils vibration'.

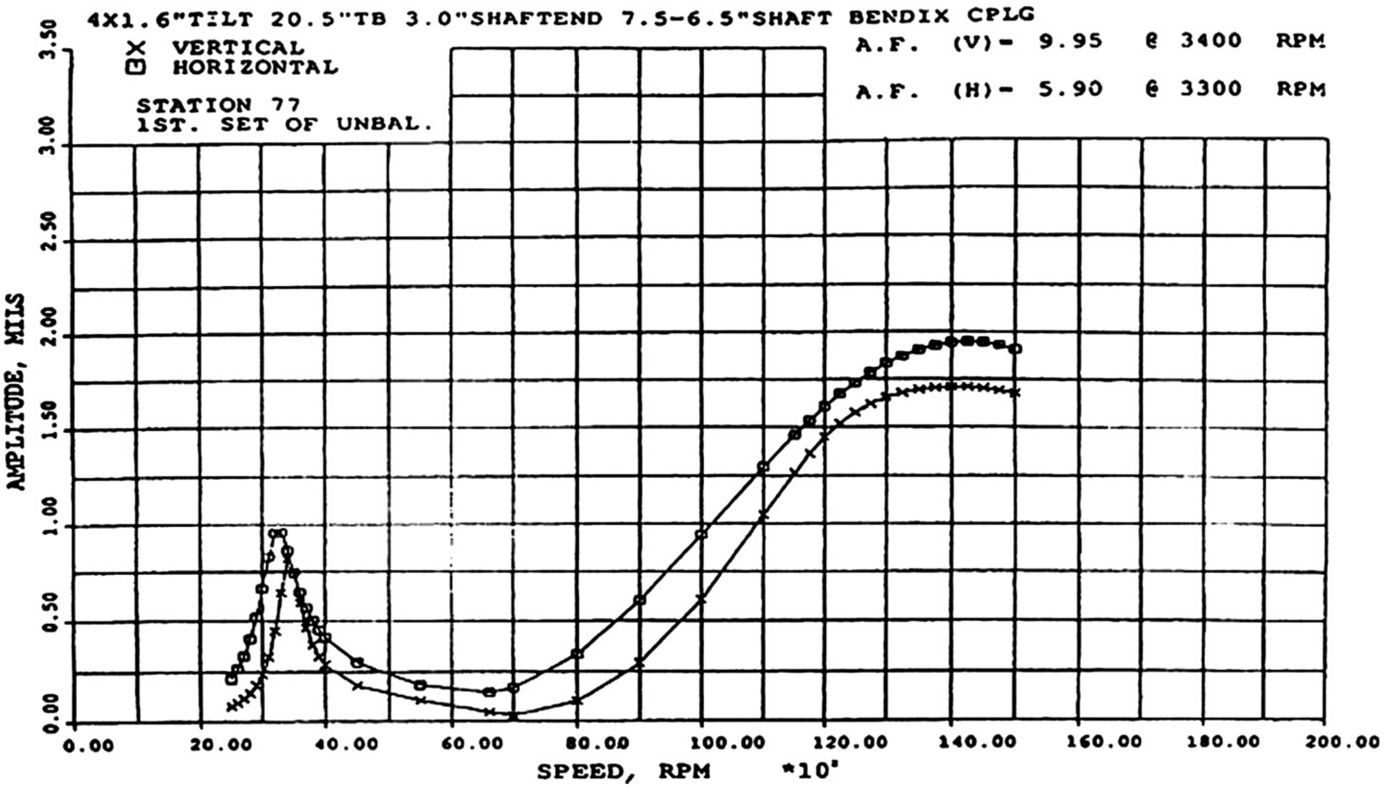


Fig 3.3.11 • Rotor response output at non-drive end bearing (N.D.E) (Courtesy of Elliott Co.)

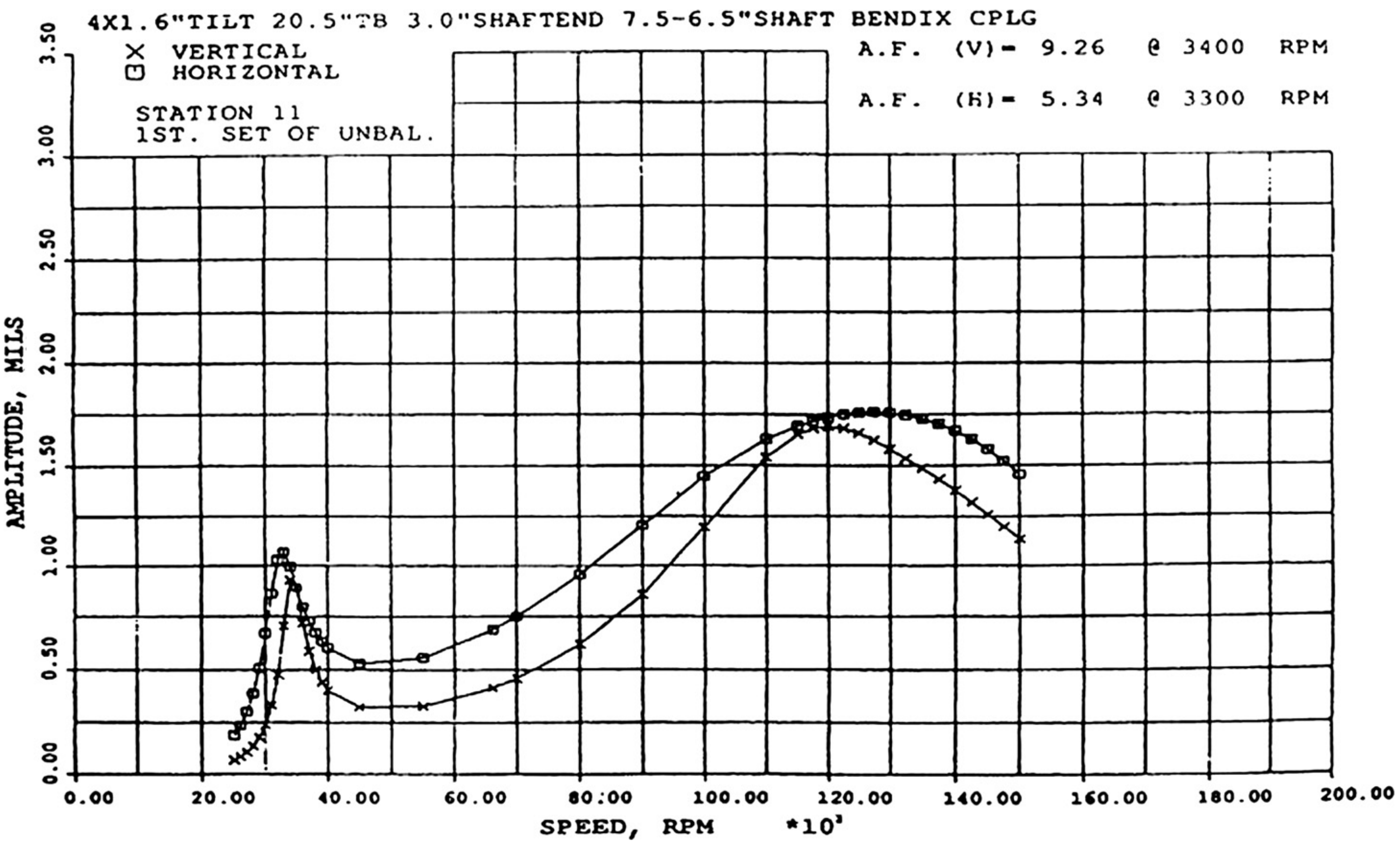


Fig 3.3.12 • Rotor response output drive end bearing (D.E.) (Courtesy of Elliott Co.)

Refer to Figure 3.3.10 and remember:

- The compressor must go through NC_1
- The shaft vibration **increases** at NC_1 (usually 2×, 3× or more)
- The vibration at center span is approximately **10** × the probe vibration

Therefore:

Vibration at the mid span during the first critical speed will be:

= (7.0 mils) × (2.0) × (10)

Probe value NC_1 amplification Mode shape difference
= 140 mils!!

Normal clearance between the rotor and interstage labyrinths is typically 40 mils! This vibration exposes the diaphragms, which are usually cast iron, to breakage. One final comment; during shutdown, the rate of rotor speed decrease **CANNOT** be controlled as in the case of start-up. It depends on rotor inertia, load in the compressor, the process system characteristics and the control and protection system. If the vibration at the probe locations is high, the best advice is to stop the compressor whilst fully loaded, which will reduce the time in the critical speed range as much as possible. Yes, the compressor will surge, but the short duration will not normally damage the compressor.

Figures 3.3.11 and 3.3.12 present the primary output of a rotor response study.

Rotor response plots display vibration amplitude, measured at the probes, vs. shaft speed for the horizontal and vertical probes. Note that a response curve must be plotted for each set of unbalance locations and unbalance amount.

Figure 3.3.11 shows the rotor response for the non-drive end (N.D.E.) set of probes with the first set of unbalance. Figure 3.3.12 shows the rotor response for the drive end set of probes (D.E.). The operating speed range of this example is 6,000 − 8,000 rpm.

Measured rotor response

During a shop test, the rotor response of every turbo-compressor rotor is measured during acceleration to maximum speed and deceleration to minimum speed. Values are plotted on the same coordinates as for the rotor response analysis. The plot of shaft vibration and phase angle of unbalance vs. shaft speed is known as a **bode plot**.

Bode plots represent the actual signature (rotor response) of a rotor for a given condition of unbalance and support stiffness. They indicate the location of critical speeds, the change of shaft vibration with speed and the phase angle of unbalance at any speed. A bode plot is a dynamic or transient signature of vibration for a rotor system, and is unique to that system for the recorded time frame. Bode plots should be recorded during every planned start-up and shutdown of every turbo-compressor. As discussed in this section, the bode plot will provide valuable information concerning shaft vibration and phase angle at any shaft speed.

Best Practice 3.4

Limit non-lubricated reciprocating compressor piston speed to below 600 ft/minute for optimum compressor non-lubricated compressor reliability (greater than 94%).

Non-lubricated reciprocating compressors have the lowest reliabilities; only around 92%.

Invest in an extra cylinder if necessary to keep piston speeds below 600 ft/minute for maximum packing, valve and piston ring/rider band life.

Lessons Learned

Failure to limit piston speed in non-lubricated reciprocating compressor applications has led to valve, packing and ring MTBFs of less than 6 months.

Benchmarks

I have used this best practice since the mid-1970s to result in non-lubricated reciprocating compressor reliabilities greater than 94%. Packing, valve, piston ring and rider band MTBFs have exceeded 24 months.

B.P. 3.4. Supporting Material

Non-lubricated reciprocating compressors have the lowest reliability (92% or lower) amongst compressor types because they, like all reciprocating compressors, have a number of wear parts but also do not have any directed packing and cylinder lubrication. Packing MTBFs can be typically less than 6 months. Reducing the piston rod speeds below 600 ft/minute has proven to optimize packing, valve, piston ring and rider band MTBFs.

In this section the functions of each major component of a reciprocating compressor are defined. That is, what the purpose of each component is or "What It Does". With this understanding, you will be in a better position to know if the compressor is performing its duty correctly. We will present each major component, state its function, operating limits and what to look for. After presenting general information for each component, we will present specific information concerning site compressors. We start with the crank shaft.

Frame and running gear

Figure 3.4.1 presents a picture of a seven-throw crank shaft arrangement, along with a sectional view of two throws. The crankcase supports the crank shaft bearings, provides a sump for the bearing and crosshead lube oil and provides support for the crosshead assembly.

Typical crank case condition monitoring and safety devices are:

- Relief device – To prevent crank case breakage in the event of explosion (caused by entrance of process gas into the crank case).
- Breather vent – To allow removal of entrained air from the lube oil.
- Crank case oil level gauge – Allows continuous monitoring of crank case lube oil level.
- Crank case oil temperature gauge – Allows continuous monitoring of crank case lube oil temperature.
- Crank case vibration detector (optional) – Provides information concerning crank case vibration useful in detecting dynamic changes in running gear.
- Crank case low oil level switch (optional) – Provides alarm signal on low crank case oil level.
- Main lube oil pump – shaft driven (optional) – Directly connected to crank shaft and usually discharges oil directly to crank shaft bearings, connecting rod bearing, crosshead shoes and crosshead pin bushing via precision bore in crank shaft and connecting rod bearing.
- Main lube oil pump discharge pressure gauge (when supplied) –Allows continuous monitoring of main lube oil pump discharge pressure.

An important reliability consideration is to make sure that the crank case is securely mounted, and is level. This requires proper grouting and maintaining a crack free (continuous) crank case base support. Since the dynamic forces on the crank case and crosshead mounting feet can be very large, it is usual to use an epoxy grout, since they provide high bond strengths and are oil resistant. All reciprocating baseplates should be continuously checked for any evidence of grout foundation cracks (discontinuities) and repaired at the first opportunity.

Figures 3.4.2 and 3.4.3 show plan, elevation and side views of a two-throw, balanced, opposed crank case assembly.

The crosshead assembly shown has the function of continuously assuring vibration-free reciprocating motion of the piston and piston rod. The crosshead pads (or shoes) and supports are usually made from Babbitt or aluminum (smaller size units). Crosshead assembly lubrication is supplied via a pressure-drilled hole (rifle drilled) in the connecting rod, which in turn lubricates the crosshead pin bushing and crosshead shoes (see Figure 3.4.3).

Cylinder distance piece

Figure 3.4.4 presents the functions of the cylinder distance piece.

The proper operation of the distance piece baffles and seals is essential for maintaining the safety and reliability of a reciprocating compressor in process gas applications. In most refinery process gas applications, a double compartment distance piece is used, to ensure contamination of the crank case or cylinders does not occur. Usually, the cylinder end compartment contains a partial N_2 atmosphere, since the packing rings are usually N_2 purged.

Reliability considerations concerning this assembly are ensuring proper packing, partition packing and wiper ring clearances.

Piston rod packing

Figure 3.4.5 depicts a sectional and exterior view of a cartridge packing assembly.

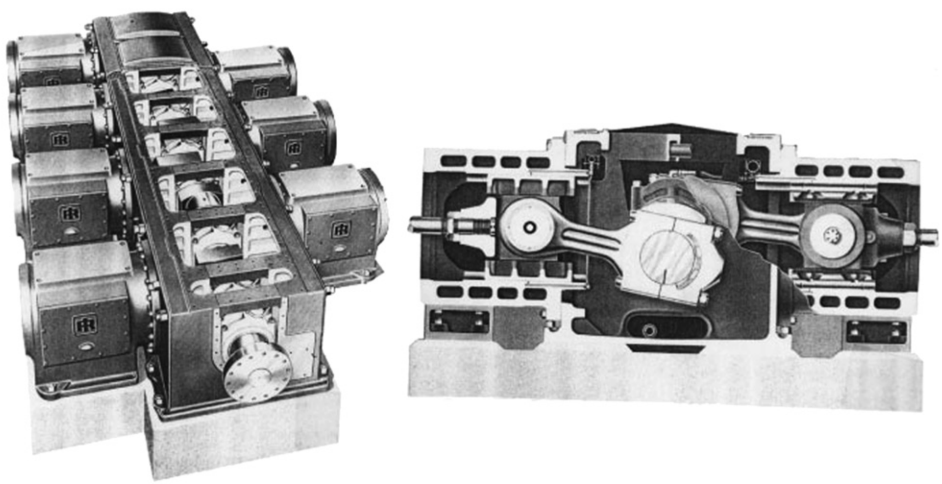


Fig 3.4.1 • Frame and running gear (crank case and crosshead) (Courtesy of Dresser Rand)

FUNCTIONS:
- TRANSMITS POWER
- CONTAINS LUBE OIL
- CONVERTS ROTARY TO RECIPROCATING MOTION

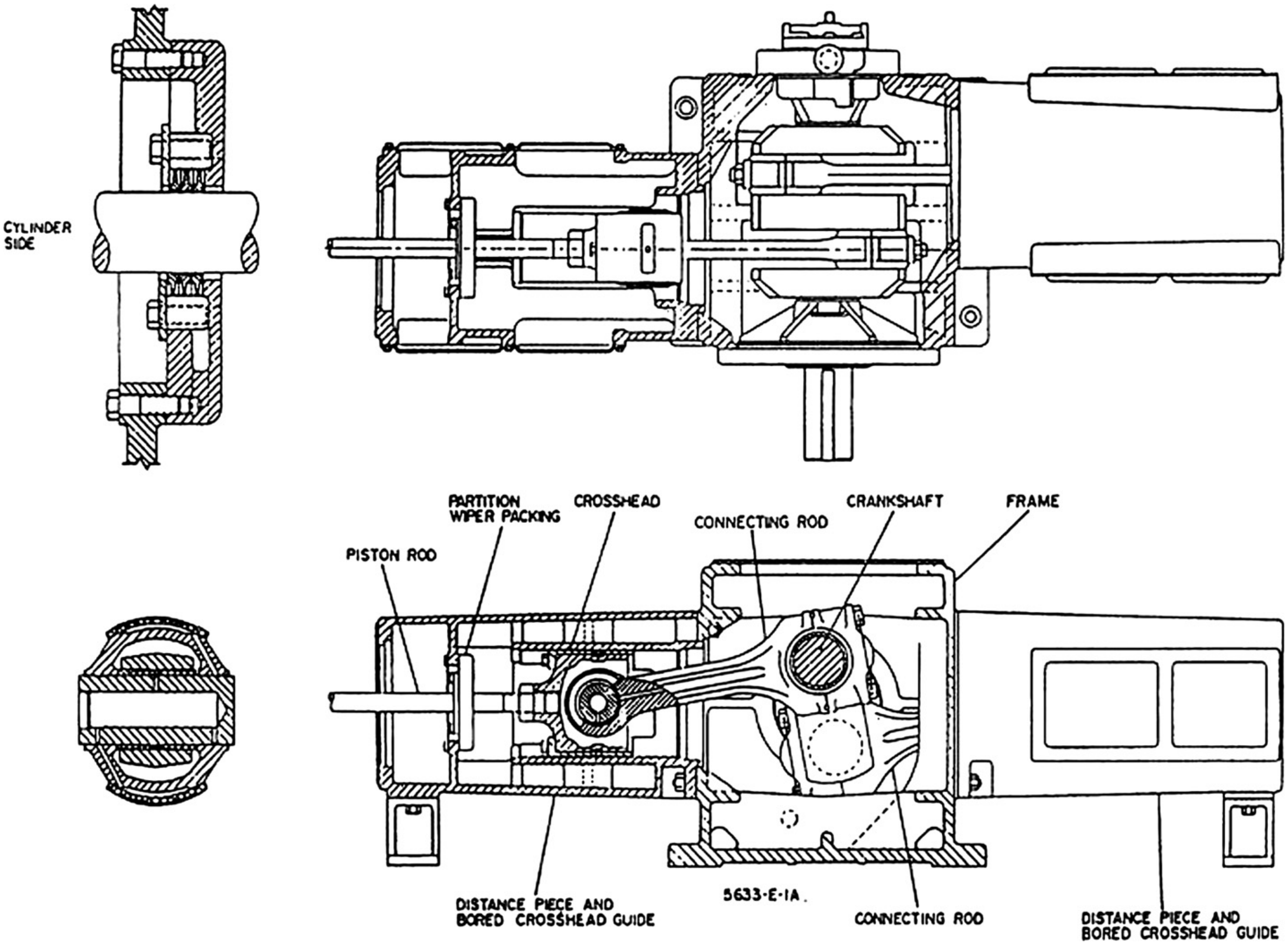


Fig 3.4.2 • HDS off-gas running gear (Courtesy of Dresser Rand)

The number of packing rings and type of arrangement is varied according to the cylinder maximum operating pressures. It is important to note that the packing does not provide an absolute seal, but only minimizes the leakage from the cylinder. The vent port is shown in the upper portion of the section drawing in Figure 3.4.5, which carries the leakage gas either to a safe vent location (atmosphere, flare or fuel gas system) or back to the cylinder suction.

A means should be available to provide easy detection of excessive packing clearances. Alternatives are:

- Packing line flow switch
- Packing line orifice and pressure switch (only if compressor pressures are controlled to be constant)
- Visual detection of gas flow (vent). Note: Flammable or toxic process gas must be purged with N_2 to attain a non-flammable mixture if the gas is to be vented to atmosphere.

Figure 3.4.6 shows additional packing assembly details.

The figure on the left side of the drawing is typical of a packing arrangement used between sections of a distance piece. Mounted horizontally, the assembly is equipped with a gravity drain and top vent.

The figure on the right side of Figure 3.4.6 shows a four (4) ring piston rod packing assembly. The upper part of the drawing shows the lubrication connections that are used when lubricated packing is required. Lube packing is normally used if the lubricant is compatible with the process stream.

If dry packing is used, piston rod speeds are usually slower and PTFE materials are usually employed. In the lower half of the drawing, the vent connections are shown and perform as previously discussed. The cup supports and positions an individual packing ring. Not shown is a purge connection which is inserted between the last and next to last packing ring (rings closest to the distance piece).

Cylinder and liner

Shown in Figure 3.4.7 are the two most common cylinder arrangements; double acting and single acting.

Most process reciprocating compressors are supplied with a replaceable cylinder liner. All cylinders are either jacked for

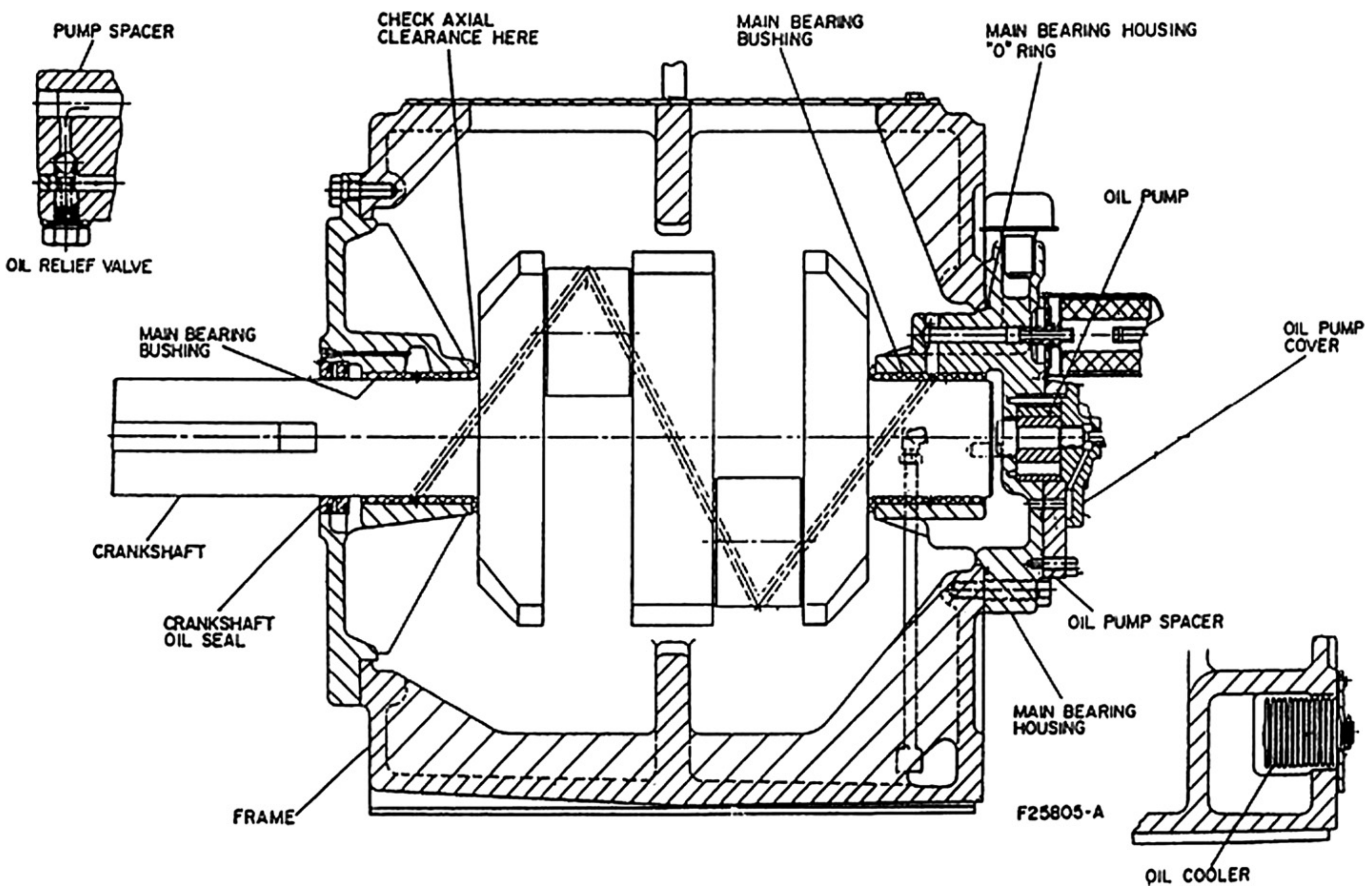


Fig 3.4.3 • HDS off-gas running gear (Courtesy of Dresser Rand)

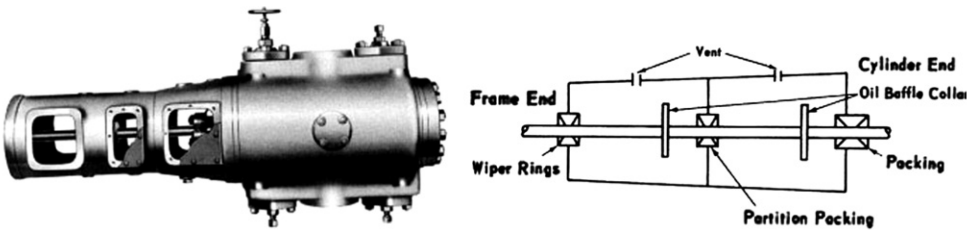


Fig 3.4.4 • Cylinder distance piece

DOUBLE COMPARTMENT DISTANCE PIECE

DOUBLE COMPARTMENT SCHEMATIC

FUNCTIONS:
- PREVENTS CONTAMINATION OF PROCESS GAS
- PREVENTS CONTAMINATION OF CRANK CASE OIL

FUNCTION: MINIMIZES GAS LEAKAGE FROM CYLINDER

Fig 3.4.5 • Cylinder packing

cooling water or finned for air cooling. Some older design cylinders use gaskets to isolate cooling water jackets from the cylinder. This design exposes the user to breakage from excessive cylinder water entrainment if the gasket fails. Most current reciprocating specifications do not allow gaskets to be used in the cylinder. A double acting cylinder is designed to compress gas on both ends of the cylinder (crank end and cylinder head end), while a single acting cylinder is designed for compression only on one end of the cylinder.

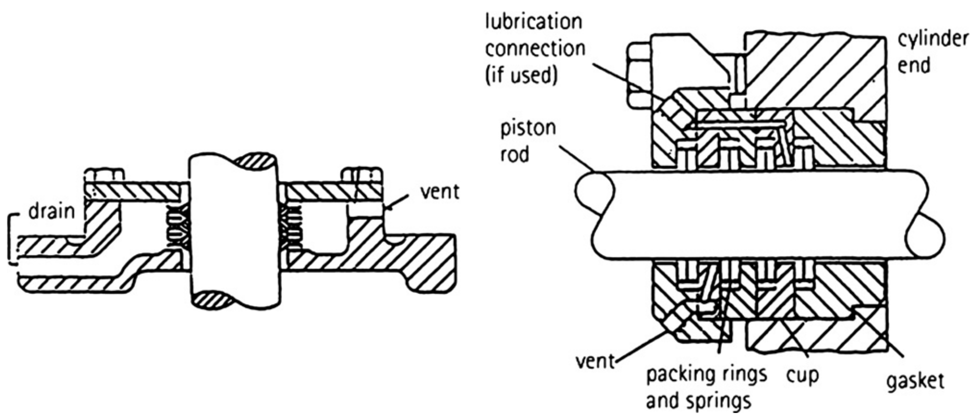


Fig 3.4.6 • Left: Oil scraper ring arrangement for 9" & 11" ESH/V, HSE units; Right: Piston rod packing (Courtesy of Dresser Rand)

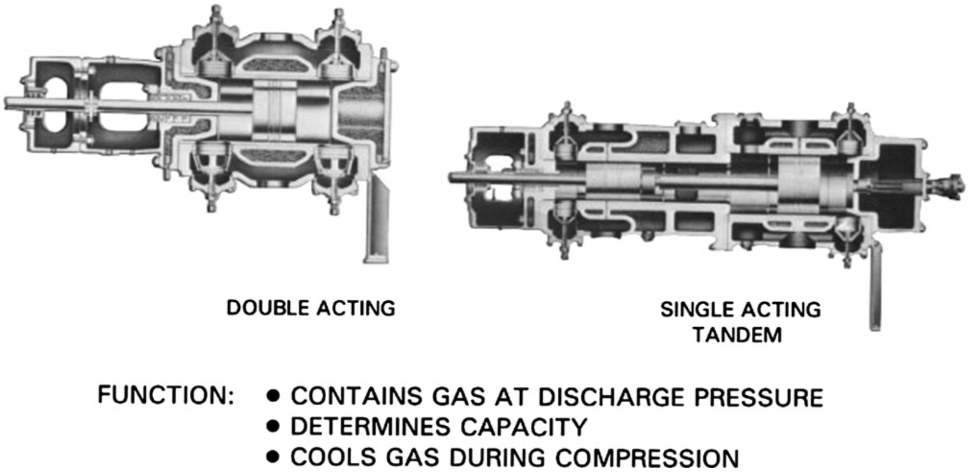


Fig 3.4.7 • Cylinder and liner (Courtesy of Dresser Rand)

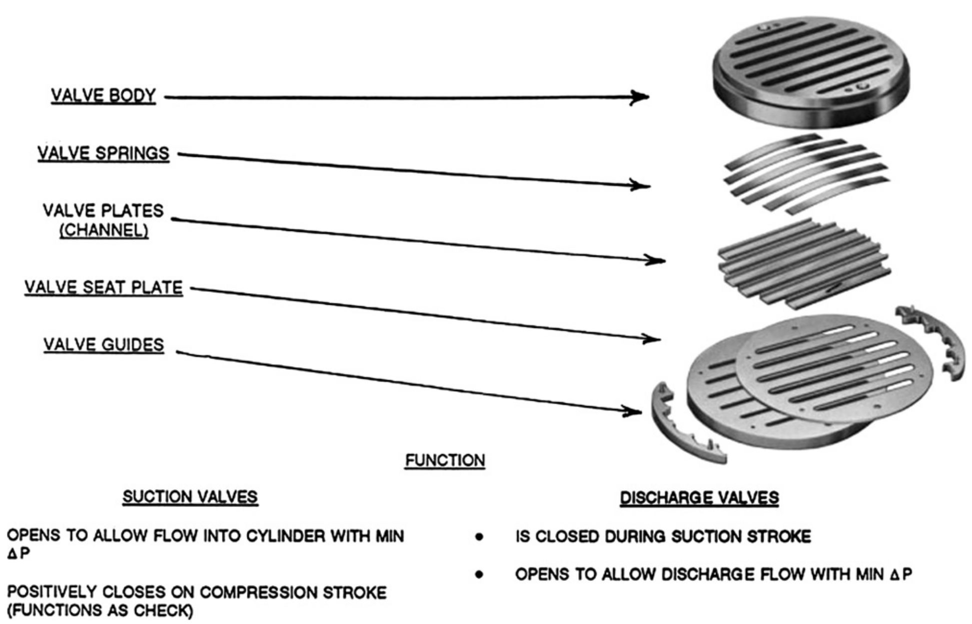


Fig 3.4.8 • Reciprocating compressor valves – Channel type, suction and discharge (Courtesy of Dresser Rand)

Reciprocating compressor cylinder valves

There are many different types of reciprocating compressor valves. Regardless of their design, all valves perform the same function, in that they allow gas to enter the cylinder, prevent recirculation flow back to the suction piping and allow gas to pass into the discharge system when the process discharge pressure at the compressor flange is exceeded. Valves are the highest maintenance item in reciprocating compressors. Their life is dependent on gas composition and condition, gas temperature and piston speed. Typical valve lives are:

- Process gas service in excess of one (1) year
- H_2 (hydrogen) gas service – 8-12 months

In hydrogen service, particular attention should be paid to cylinder discharge temperature in order to obtain maximum valve life. Cylinder discharge temperature for service with > 60% H_2 should be limited to 250°F. Recently, light weight, non metallic valves (made from PEEK) have been used successfully to increase the valve life in H_2 service to above one (1) year.

Figure 3.4.8 shows a typical channel valve assembly.

The life of the channel valves shown is controlled by the spring force of the valve springs. The channel arrangement reduces the forces on the valve seal and usually results in increased valve life.

Figure 3.4.9 depicts a ring or plate valve assembly. This type of valve is most widely used.

Regardless of the type of valve, condition monitoring of valves is important to the profitability of any operation. The following parameters should be monitored:

Type of valve	
Suction	■ Valve body temperature
	■ Compressor volume flow rate
Discharge	■ Interstage process gas temperature
	■ Compressor volume flow rate

Changes in these parameters in excess of 10% from original (baseline) values should be cause for component inspection and replacement.

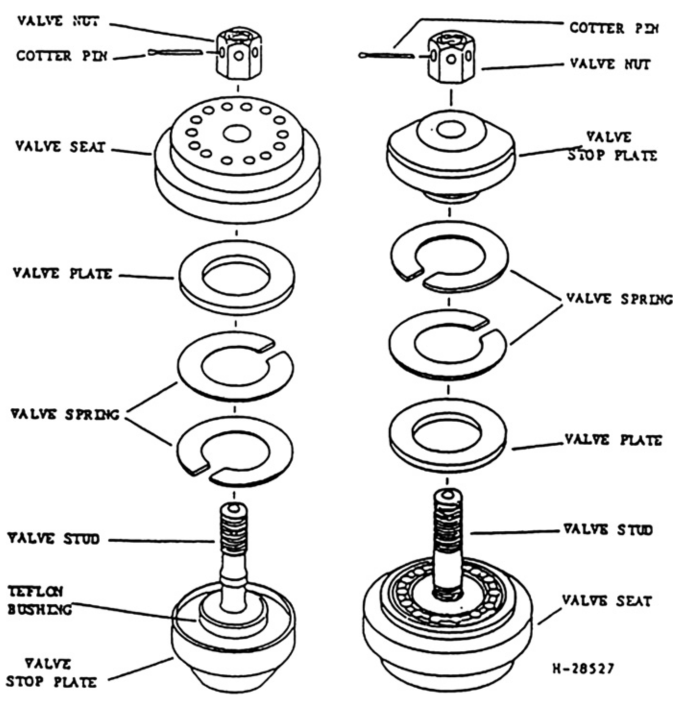


Fig 3.4.9 • HDS suction and discharge valves (Courtesy of Dresser Rand)

Piston assembly

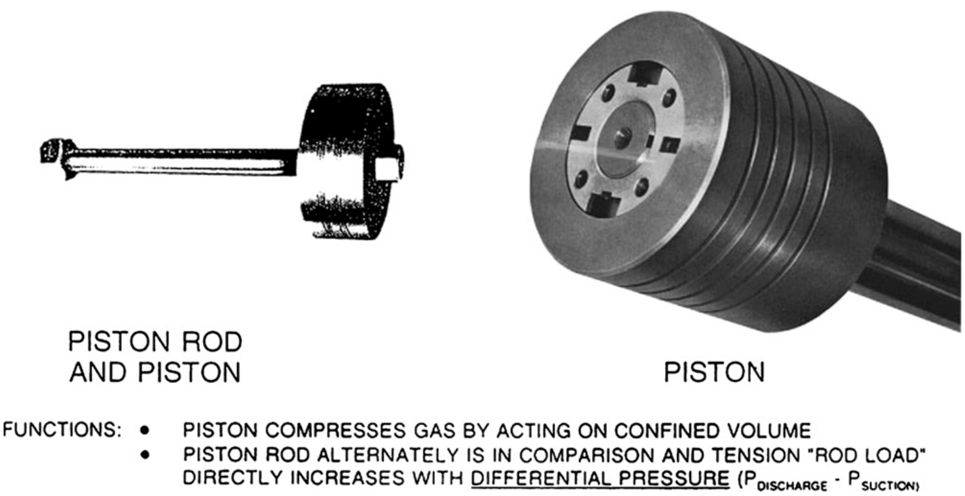
Figure 3.4.10 presents a typical piston assembly consisting of the piston rod nut, piston rod, piston and piston nut.

Piston rod materials are hardened steel and can include metal spray in packing areas to extend rod life. Piston materials can be steel, cast nodular iron or aluminum. The most common is cast iron due to its durability. Aluminum pistons are used in large cylinder applications (usually first stage) to minimize piston rod assembly weight.

Figure 3.4.11 shows piston rider bands (2) (items 111) and piston rings (3) (items 113).

The rider band and ring material is dependent on cylinder lubrication. If the cylinder is lubricated, carbon materials or

Fig 3.4.10 • Rod and piston



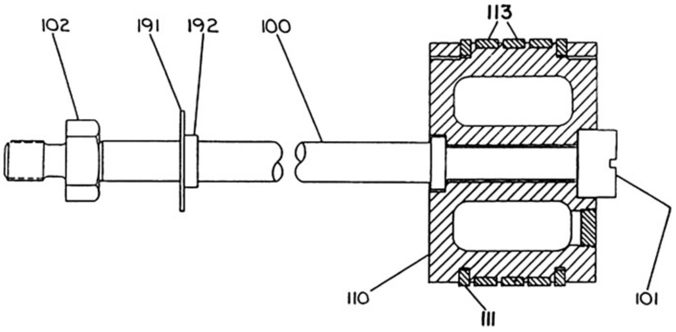


Fig 3.4.11 • HDS off-gas-piston rod and piston (Courtesy of Dresser Rand)

compounds are used. If non-lubricated service is required, PTFE materials or other Teflon derivatives are used. Note also the piston hollowed area for piston weight control. Rider band and ring life is a function of piston speed, cylinder gas temperature and cleanliness of the process gas. In many process applications, a strainer is required upstream of the compressor to prevent excessive ring wear.

Condition monitoring of rider band and piston ring wear can be accomplished by measuring and trending the vertical distance between each piston rod and a fixed point (known as rod drop). This can be accomplished either by mechanical or electrical (Bentley Nevada proximity probe) means. Of importance in piston assembly design is rod loading and rod reversal.

Rod loading is the stress (tension or compression) in the piston road and crosshead assembly caused by the $\triangle$P across

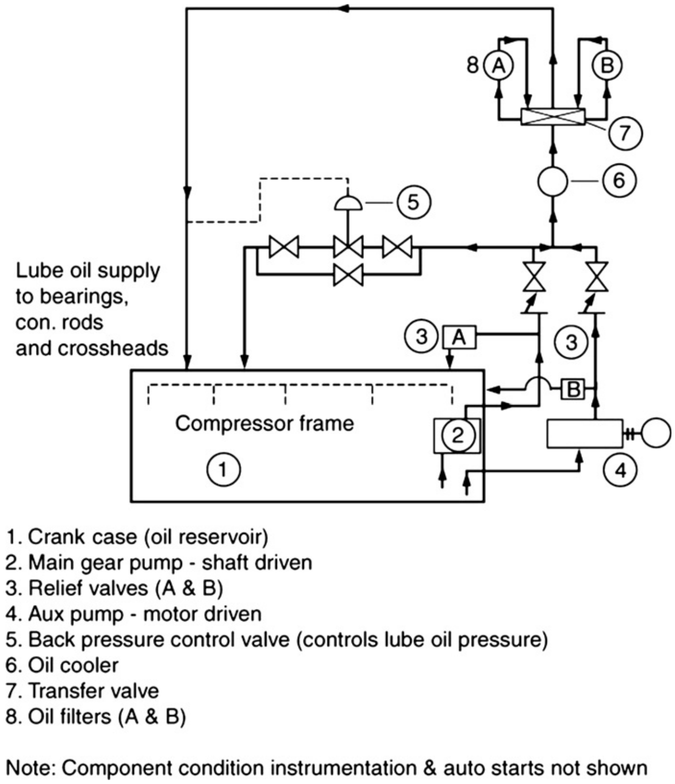


Fig 3.4.13 • Lube oil system

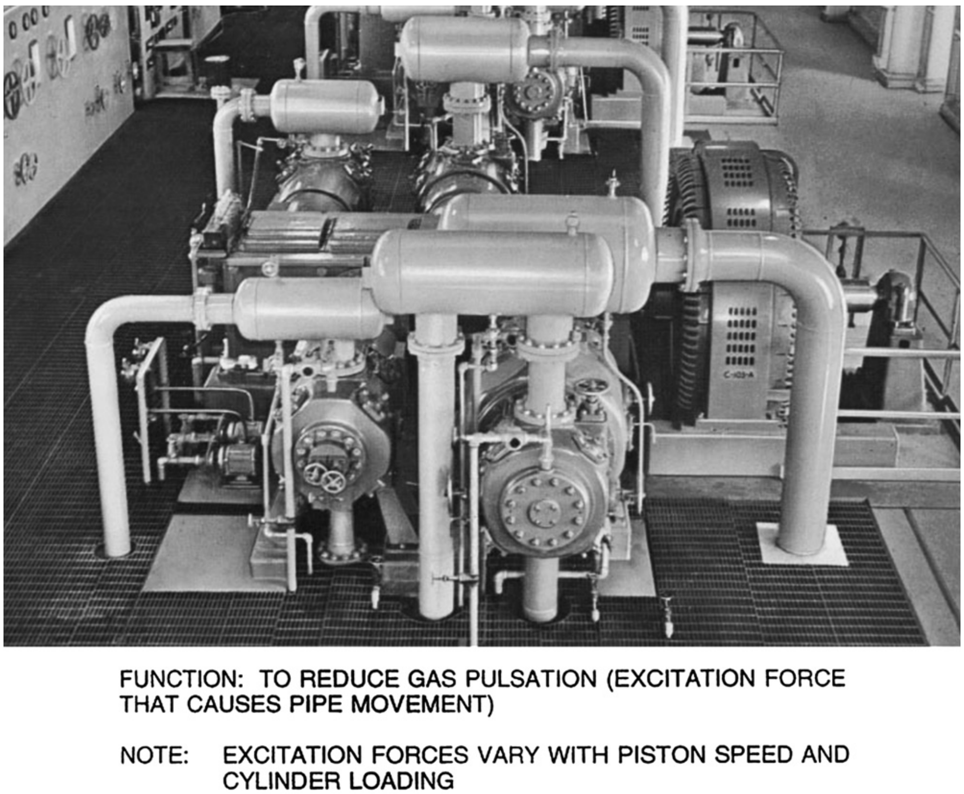


Fig 3.4.12 • Pulsation dampeners (Courtesy of Dresser Rand)

the piston. Rod load limits the maximum compression ratio that a cylinder can tolerate. This is the reason that many first stage cylinders are supplied with a suction pressure switch. Rod reversal is necessary so that the piston rod reaction forces on the crosshead pin will change allowing oil to enter the pin bushing. If the position of the pin in the bushing did not change (reverse) with each stroke, the bushing could not be sufficiently lubricated and would prematurely fail.

Pulsation dampeners

Since the action of the piston is non-continuous, pressure pulsations will be generated. Depending upon the piping arrangement, these pulsations can be magnified to destructive levels. The use of pulsation dampeners, as shown in Figure 3.4.12, can reduce pulsations to 2% or lower.

There are methods available to evaluate and simulate the effect of pulsation dampeners prior to field operation. However, the variation between predicted and actual results can be large and field modifications (installation of orifices or pipe modifications) may be necessary.

Cylinder and packing lubricators

Whenever mineral oil is compatible with the process, lubricators will be used. Lubricators can be either a positive displacement or dynamic type. It is usual to review lubrication details in the appropriate instruction book. Lubricators will increase piston ring and packing life by reducing friction.

Figure 3.4.13 presents a typical lube oil system and its function.

All instruments in the lube oil system should be continuously monitored (baseline and current conditions). Remember, component (bearing) failure will occur if any major component in the system fails to function.

Figure 3.4.14 shows a lube oil system containing a shaft driven main lube oil pump with an internal relief valve. This arrangement is a common one. Failure of the relief valve to seat can cause a low lube oil pressure trip.

Cooling system

The final topic to be covered is the cooling system. The cylinders, packing and process gas must be cooled to extend their run

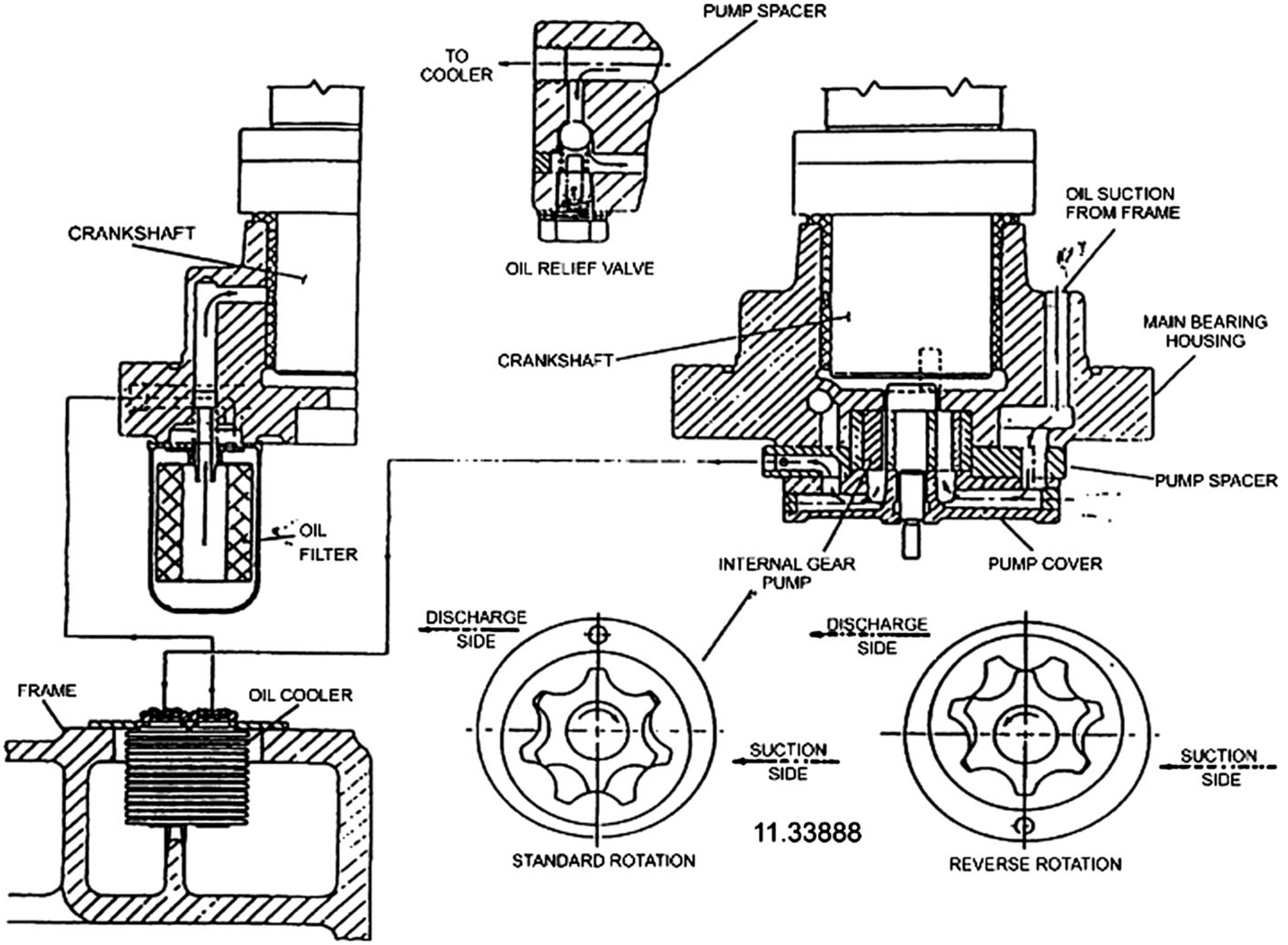


Fig 3.4.14 • HDS off-gas-lube oil system (Courtesy of Dresser Rand)

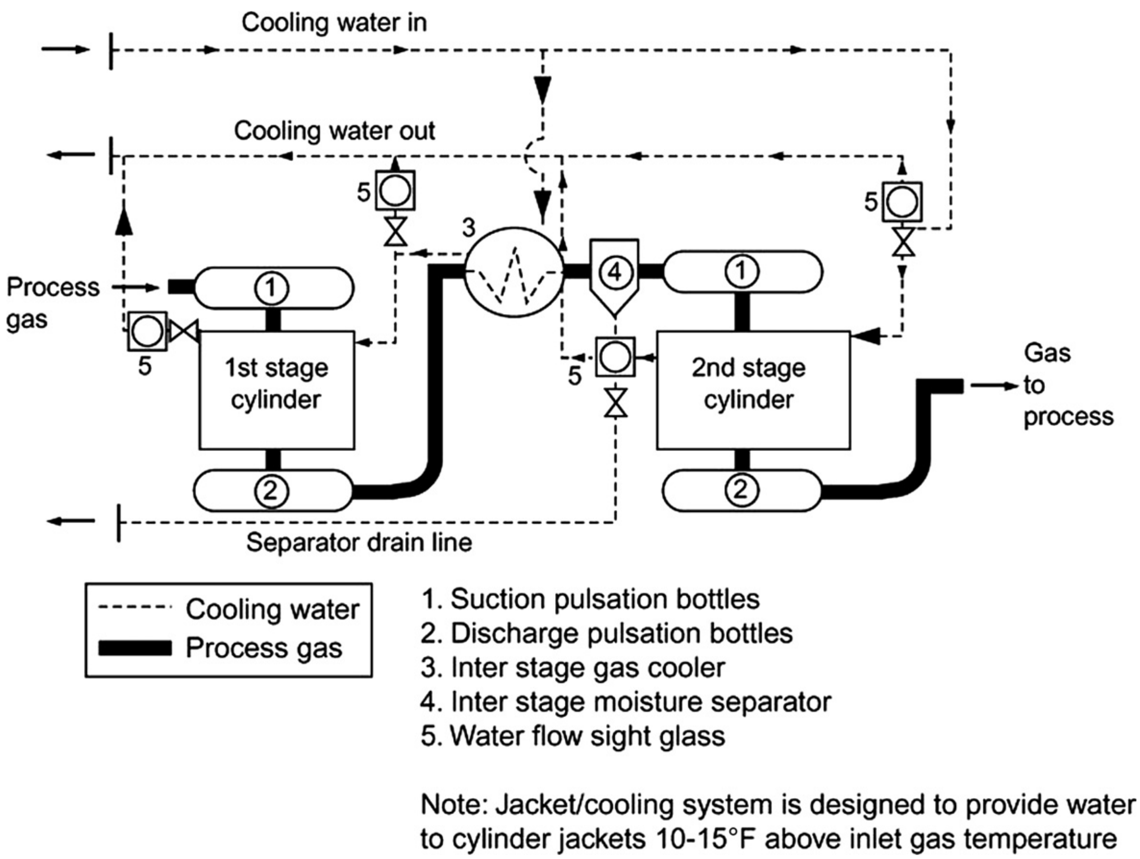


Fig 3.4.15 • Cylinder, packing and intercooler cooling water system

time, and to minimize maintenance. Figure 3.4.15 presents a typical water cooled circuit.

The temperature of the cooling water must be regulated so that moisture (condensate) will not form in the cylinder in wet gas applications. It is recommended that the tempered water system temperature in the cylinder be maintained a minimum of 10–15°F above the cylinder inlet gas temperature.

Careful monitoring of the cooling circuit is essential in determining cooler, jacket, cylinder maintenance (cleaning) requirements.

Best Practice 3.5

Restrict reciprocating compressor field pulsation limits to +/− 2% of line pressure for safe and reliable operation.

Take care during the design phase that an accurate isometric piping arrangement is used for the digital pulsation analysis.

Check field pulsation values immediately after start-up and take immediate corrective action (installation of orifices, pulsation bottle corrections, etc.) to ensure optimum levels of safety and reliability.

Lessons Learned

Failure to address pulsation problems as indicated during the design and/or field operation phase gives rise to a number of issues.

Examples of these are:

- Instrument breakage and gas release
- Piston rod breakage
- Crank case/distance piece bolt breakage
- Packing failure
- Valve failure

Benchmarks

Since the late 1980s, when working as a troubleshooting specialist, I have used this best practice to correct many field pulsation issues that had caused safety problems and had significantly reduced compressor MTBFs.

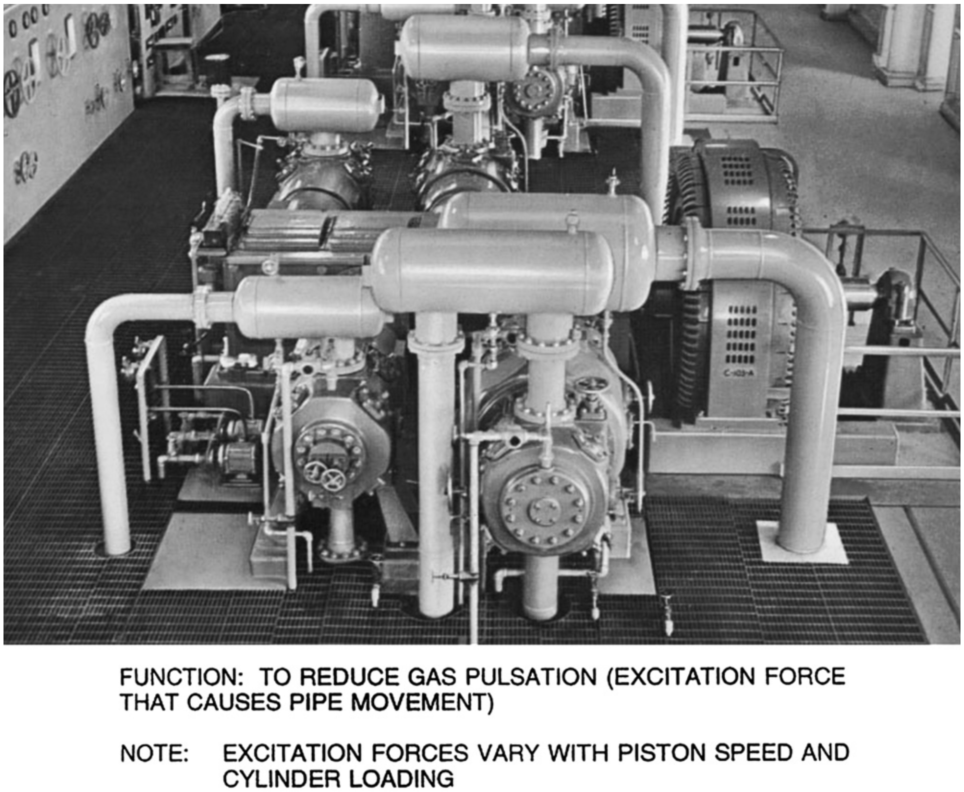


Fig 3.5.1 • Pulsation dampeners
(Courtesy of Dresser Rand)

B.P. 3.5. Supporting Material

Pulsation dampeners

Since the action of the piston is non-continuous, pressure pulsations will be generated. Depending upon the piping arrangement, these pulsations can be magnified to destructive levels. The use of pulsation dampeners, as shown in Figure 3.5.1, can reduce pulsations to 2% or lower.

There are methods available to evaluate and simulate the effect of pulsation dampeners prior to field operation. However, the variation between predicted and actual results can be large and field modifications (installation of orifices or pipe modifications) may be necessary.

Best Practice 3.6

Do not use medium or high speed reciprocating compressors (greater than 400 rpm) for plant duty

Justify low speed reciprocating compressor use based on site, company and industry experience during the early phases (pre-FEED) of the project.

Lessons Learned

The use of medium or high speed compressors in plant service has resulted in $MM in lost revenue and maintenance costs.

One plant's maintenance cost for high speed reciprocating compressors (1200 rpm), in sour gas service, was five times the cost of low speed compressors in the same plant with the same percentage of sour gas as the high speed compressors.

Benchmarks

I have used this best practice since 2000, in all projects which use lubricated reciprocating compressors in plants. This action has resulted in reliabilities for lubricated reciprocating compressors in excess of 96%.

B.P. 3.6. Supporting Material

In the last 10 years, the use of medium (400-1000 rpm) and high speed (1000-3600 rpm) lubricated reciprocating compressors has gained popularity with project personnel. Granted, the capital costs are lower, and installation can and has been skid type in many cases.

Field personnel experience is significantly different and has resulted in many hard lessons learned that now prohibit the use of this type of reciprocating compressor.

Maintenance costs, excessive pulsation and associated safety and mechanical issues have resulted in a great aversion to the use of lubricated compressors operating above 400 rpm. Typical component MTBFs for high speed (greater than 1000 rpm) are:

- Packing – less than 12 months
- Piston rings – less than 12 months
- Valves – less than 12 months
- Shutdown to repair pulsation related issues – less than 6 months.

Best Practice 3.7

Do not use lubricated screw compressors in sour gas services or when C6+ components are present in the gas analysis.

Accurately determine the gas sample in the pre-FEED phase and note if the gas is sour (contains any H_2S (hydrogen sulfide)) and/or has carbon components C6+.

If either of the two characteristics noted above are present, do not use a lubricated screw compressor.

Lube oil in these services will deteriorate (mineral or synthetic will be equally affected), hence reducing the life of bearings, and quickly plugging the lube oil filters, requiring frequent attention (often once per day!).

Lessons Learned

The use of lubricated screw compressor types in sour gas services and/or where C6+ components are present has resulted in the following consequences:

- Need for daily lube oil filter changeover
- Shutdown of compressors weekly for bearing change and complete oil system cleaning (one week duration)
- Legal suits costing $MM

Benchmarks

I have used this best practice since 2000 when multiple lubricated screw compressor field issues were experienced. All of these issues caused the end user significant costs (revenue and maintenance), and in one case resulted in legal action that lasted for over three years and did not produce positive results for the end user or vendor.

B.P. 3.7. Supporting Material

Screw compressors are the newest type of compressors. The dry screw compressor was developed in the late 1940s and did not experience wide use until the 1960s for low to medium flow plant air services. On the other hand, reciprocating compressors were developed 100 years before (circa 1850), and centrifugal compressors at the turn of the last century. In the 1980s, the concept of the dry screw compressor was modified by continuously injecting a liquid (usually lube oil), which enabled much higher compression ratios and simplified the mechanical design by eliminating the timing gears. Figure 3.7.1 and 3.7.2 illustrate the two major screw compressor designs.

Screw compressors have many inherent advantages over their competition in the low to medium flow range. Since they are positive displacement compressors, like the reciprocating type, they will draw a constant inlet volume (assuming constant speed), can meet the varying differential pressure requirements of the process and are not significantly affected by gas density changes.

However, unlike their reciprocating cousins, they can accomplish the above tasks by drawing a continuous, non-pulsating volume. As a result, pulsations are minimized, suction and discharge valves and troublesome unloaders are not required, nor is high maintenance packing. Since there is only rotary motion, all of the conventional sealing alternatives are available (including well proven dry gas mechanical seals).

In addition, the rotary motion significantly reduces the 'footprint' of the screw compressor unit compared to other positive displacement alternatives. The result is an efficient, reliable compressor type that is very competitive from an initial cost and installed cost standpoint. The advantages of screw type compressors are presented in Figure 3.7.3.

As a result of their many advantages, dry screw and flooded screw compressors have become a dominant force in low to medium flow process applications. They have also continued to grow as the preferred type of plant and instrument air compressor in this flow range.

In the 'upstream' exploration and production industry, and in gas plants, the screw compressor has become 'the type to use'. For gas gathering, the depleted fields of North America have utilized the advantages of the screw compressor to provide a highly reliable, cost-effective service. Many 'upstream' applications utilize gas turbines as prime movers for power generation, large pump and compressor drives. The screw compressor

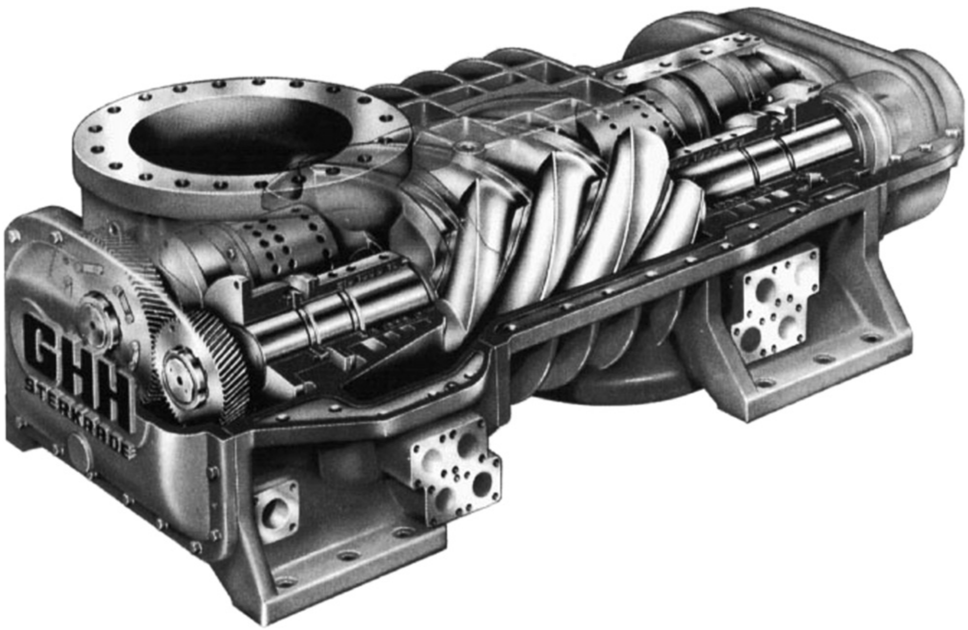


Fig 3.7.1 • Dry twin screw compressor
(Courtesy of Man/GHH)

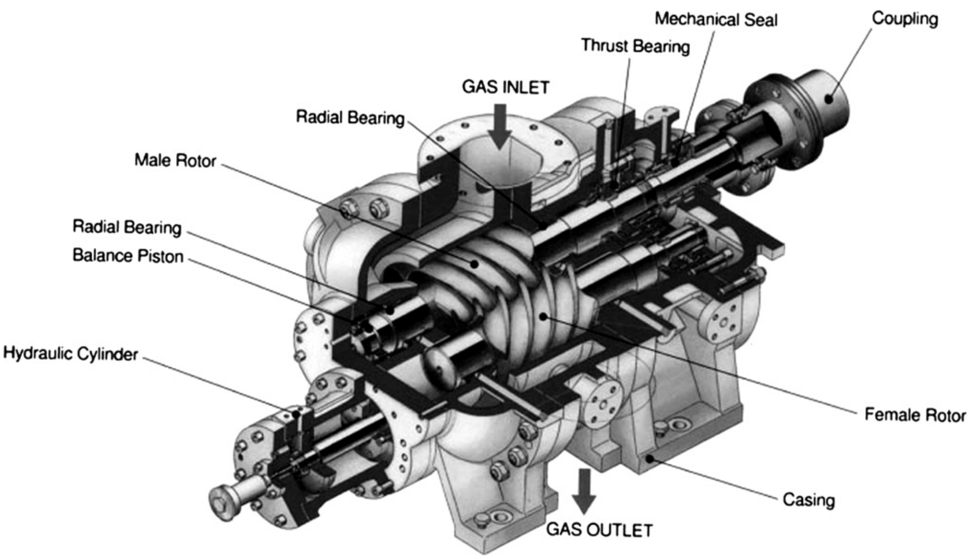


Fig 3.7.2 • Oil injected twin screw compressor (Courtesy of Kobelco-Kobe Steel Ltd.)

Screw compressors exhibit the following advantages over other positive displacement compressor types:

- Constant inlet volume flow
- Can meet variable process differential pressure requirements
- Can handle varying density and dirty gases
- Pulsation free
- Valves, unloaders and volume pockets not required
- Mechanical seals used, packing not required
- High reliability (close to centrifugal types)
- Small ‘footprint’
- Low installed cost

Fig 3.7.3 • Screw compressor advantages

is rapidly becoming the type of choice in gas turbine fuel gas booster applications, which require medium flows, pressure ratios and the ability to handle varying gas densities. As the characteristics and advantages of screw compressors become more widely appreciated, their use in all types of applications will increase.

The objective of this chapter is to familiarize the reader with the design, principle of operation, application and condition monitoring of the different types of screw compressors available. Figure 3.7.4 defines the objectives of this chapter.

Throughout this section, the reader must remember that all screw compressors are positive displacement compressors.

- Define design features and component function
- Present selection and application guidelines
- Inform regarding field condition monitoring requirements

Fig 3.7.4 • Screw compressor chapter objectives

Positive displacement – dynamic pump comparison

	Positive displacement	Dynamic
Definition	Increases pressure by operating on fixed volume in a confined space	Increases pressure by using rotary blades to increase fluid velocity
Types	Screw, gear, reciprocating	Centrifugal, axial
Characteristics	■ Constant volume ■ Variable differential head ■ Relatively insensitive to liquid properties ■ Relatively insensitive to system changes ■ Not self-limiting	■ Variable volume ■ Constant differential head ■ Sensitive to liquid properties ■ Sensitive to system changes ■ Self-limiting

Characteristic flow vs. differential head curves

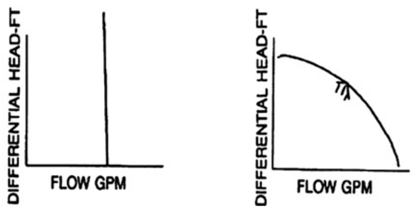


Fig 3.7.5 • Positive displacement and dynamic characteristics

As will be discussed in this section, all types of positive displacement compressors present the designer with a challenge. This challenge is to provide varying flow requirements to the process system in a safe and reliable manner. You will discover that the screw compressor industry has met the challenge in a safe, reliable and most efficient way.

Principles of operation

Figure 3.7.5 shows the basic principles of operation for a twin screw compressor. Regardless of type, dry or flooded, the principle is the same.

As the screws separate at the suction end, the volume between the male (drive) and female (idle) screw is filled with gas until the outlet screw flute passes out of the suction volute; the suction volume is a function of the mating screw volume and the speed of the compressor.

Once the screw flute passes through the section volute, the compression phase begins and continues until the screw flute enters the discharge volute. The designer determines the length of the compression phase by the machine specified requirements. The compressor ratio is a function of the volume reduction ratio and the gas characteristics (specific heat ratio K=cp/cv). Details concerning performance will be discussed later in this chapter.

Figure 3.7.7 shows the effects of operating any screw compressor on input power at greater and less than the specified compressor ratio. This situation is always the actual operating mode in the field. Since the power required is a function of volumetric efficiency (related to internal leakage through the screws), the effects of off-design operation (over or under pressurizing) are minimal (usually less than 3%).

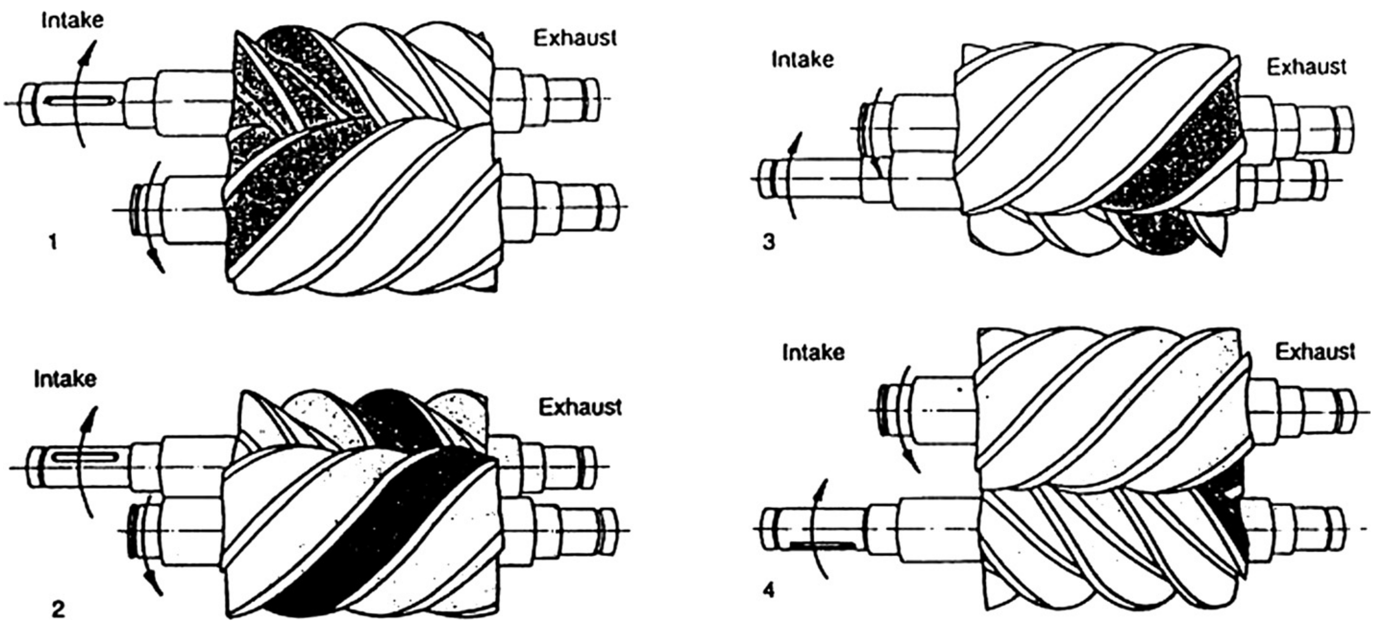


Fig 3.7.6 • Principle of operation

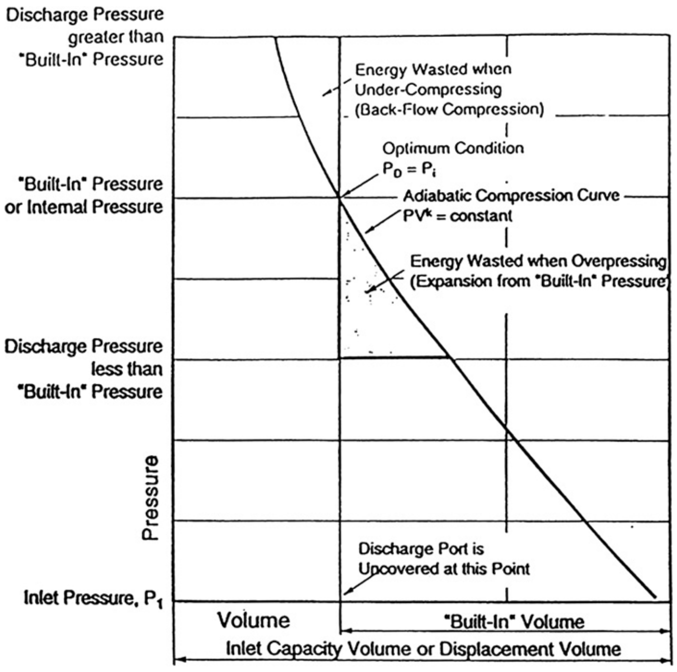


Fig 3.7.7 • Screw pv diagram

Screw compressor types

Twin screw – oil-free (non oil-injected)

Figure 3.7.8 shows a single stage, oil-free, twin screw compressor with major components noted.

The oil-free screw compressor was the first type to be developed for use as a plant and instrument air compressor. Since no lubrication is introduced in the gas stream, an external, close tolerance timing gear is used to separate the rotors. Consequently, both screws must be sealed at each end to prevent oil from entering the gas stream.

To maintain high volumetric efficiency by minimizing internal leakage (slip) losses, the backlash of the timing gears (clearance between teeth) controls the screw rotor clearance to small valves. A major reliability factor in dry screw type compressors is rotor deflection at high compression ratios.

Typically, the compression ratio for dry screw compressors is limited to around 5:1. With special screw rotor profiles, external cooling jackets and liquid injection (as great as 20% of inlet mass flow) higher compression ratios can be attained in one casing (approx. 10:1) with discharge gas temperatures being as high as 550°F. The flow range for dry screw compressors is 300–40,000 ACFM. Please refer to Figure 3.7.9 to view the various screw compressor rotor profiles and their applications.

Dry screw compressors used to be the preferred type for plant and instrument air services. However, in recent years, oil-flooded screw compressors have gained popularity in this field of

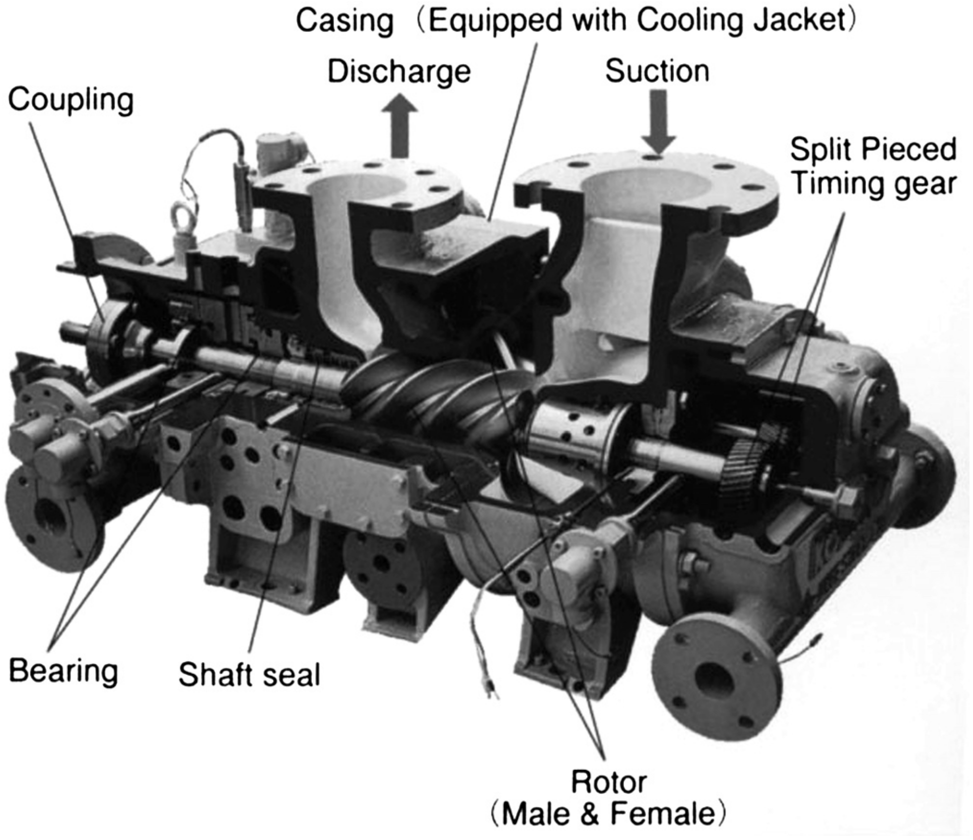


Fig 3.7.8 • Single stage twin oil free compressor (Courtesy of Kobelco – Kobe Steel Ltd)

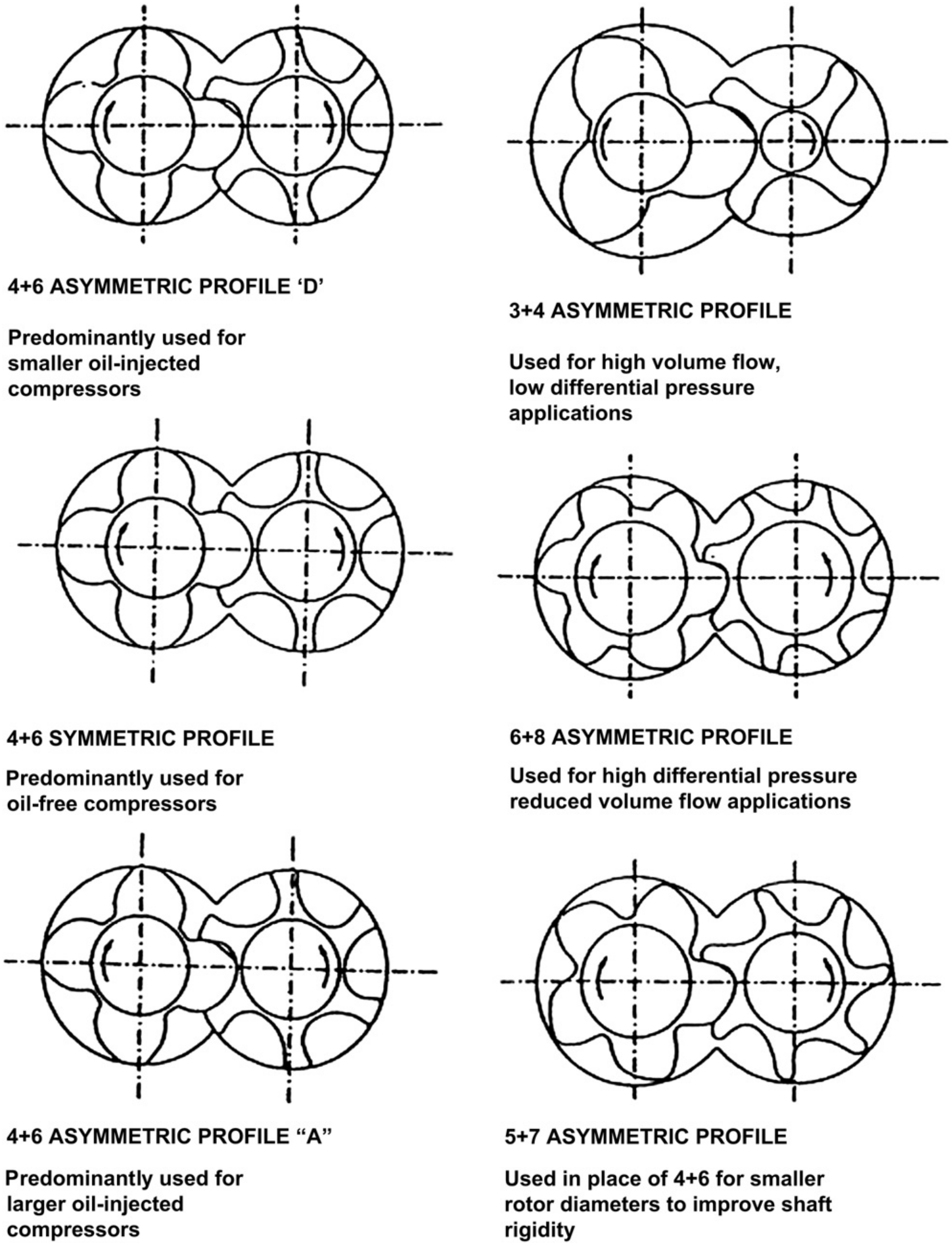


Fig 3.7.9 • Screw compressor rotor profiles

- External timing gears required
- Normal compression ratio 5:1
- Jacket cooling and/or rotor cooling and liquid injection allows operation up to 288°C (550°F) max ratio = 10:1 Δ P max = 2,585 kPa (375 PSI) with special screw profile
- Multi staging may be required to prevent high discharge temperature and rotor deflection 500–68,000 m^3/hr (300–40,000 acfm)
- Approximately $^1/_3$ speed of centrifugal compressors and 3x speed of flooded screw compressors
- Seals required at each end of screws

Fig 3.7.10 • Dry screw compressor facts

application, due to the development of high efficiency oil separation coalescers and the lower cost of a flooded-screw type compressor unit. Dry screw compressors should be considered for sour process services, since sour gas can cause oil quality deterioration, which can result in the need for frequent oil changes and possible component damage. Facts concerning dry screw compressors are noted in Figure 3.7.10.

Twin screw – oil-flooded

A twin screw, oil-flooded compressor is shown in Figure 3.7.11 with the major components noted.

The oil-flooded twin screw compressor, which was developed in the 1960s, has become the most widely used variety. The primary reason for its success is its ability to handle very high compression ratios without needing external cooling.

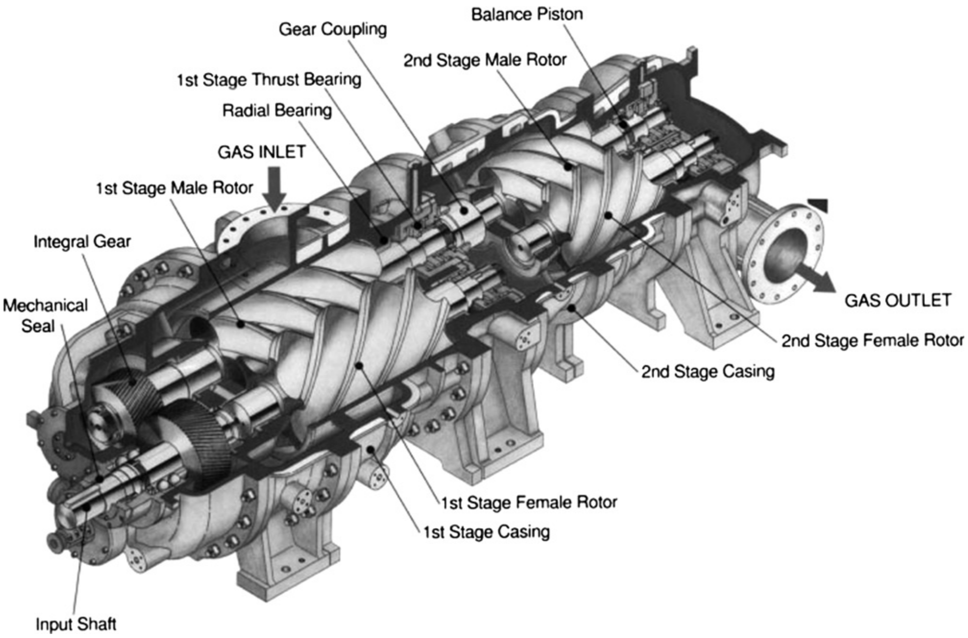


Fig 3.7.11 • Two-stage, tandem screw oil-flooded compressor. Note: integral gear used for speed increase of male rotors (Courtesy of Kobelco – Kobe Steel Ltd)

Compression ratios as high as 25:1 are possible, and ΔPs of 5,500 kPa (800 PSI) can be attained with special screw profiles (6+8). Flow range for this type is 170–14,500 m^3/hr (100–8,500 ACFM). Oil injection enables operation at high compression ratios, but limits discharge temperature to 100°C (210°F). Use of synthetic oils allows discharges temperature to reach 120°C (250°F). The injection of oil also eliminates the necessity for timing gears, and reduces the number of shaft end seals required to one.

Another advantage of oil-flooded screw compressors is the efficiency of their capacity control system. The use of oil internal to the compressor allows the use of a slide valve in the compressor section of the compressor. By varying the stroke of the valve, the volume ratio can be varied. The result is that flow can be adjusted between 10–100%.

Finally, flooded-screw compressors have proven that they can tolerate dirty, difficult to handle gases. The continuous oil injection serves as an 'anti-foulant liquid' and prevents fouling build-up.

It can be seen, therefore, that the screw compressor has many advantages over the other types and can usually be purchased for a lower cost. The facts concerning flooded screw compressors are detailed in Figure 3.7.12.

- External timing gears not required
- Compression ratios as high as 25:1
- Discharge temperature limited to 100°C (210°F) (mineral oil)/ 120°C (250°F) (synthetic oil)
- 170–14,500 m^3/hr (100–8,500 acfm)
- $^1/_3$ speed of dry screw compressors
- Only one shaft end seal required
- Rotor cooling not required
- Compatibility of oil with gas must be confirmed

Fig 3.7.12 • Flooded screw facts

However, the compatibility of the injected oil with the process gas must be examined before purchasing a flooded type. There are two primary concerns:

- Process system deterioration
- Oil degradation

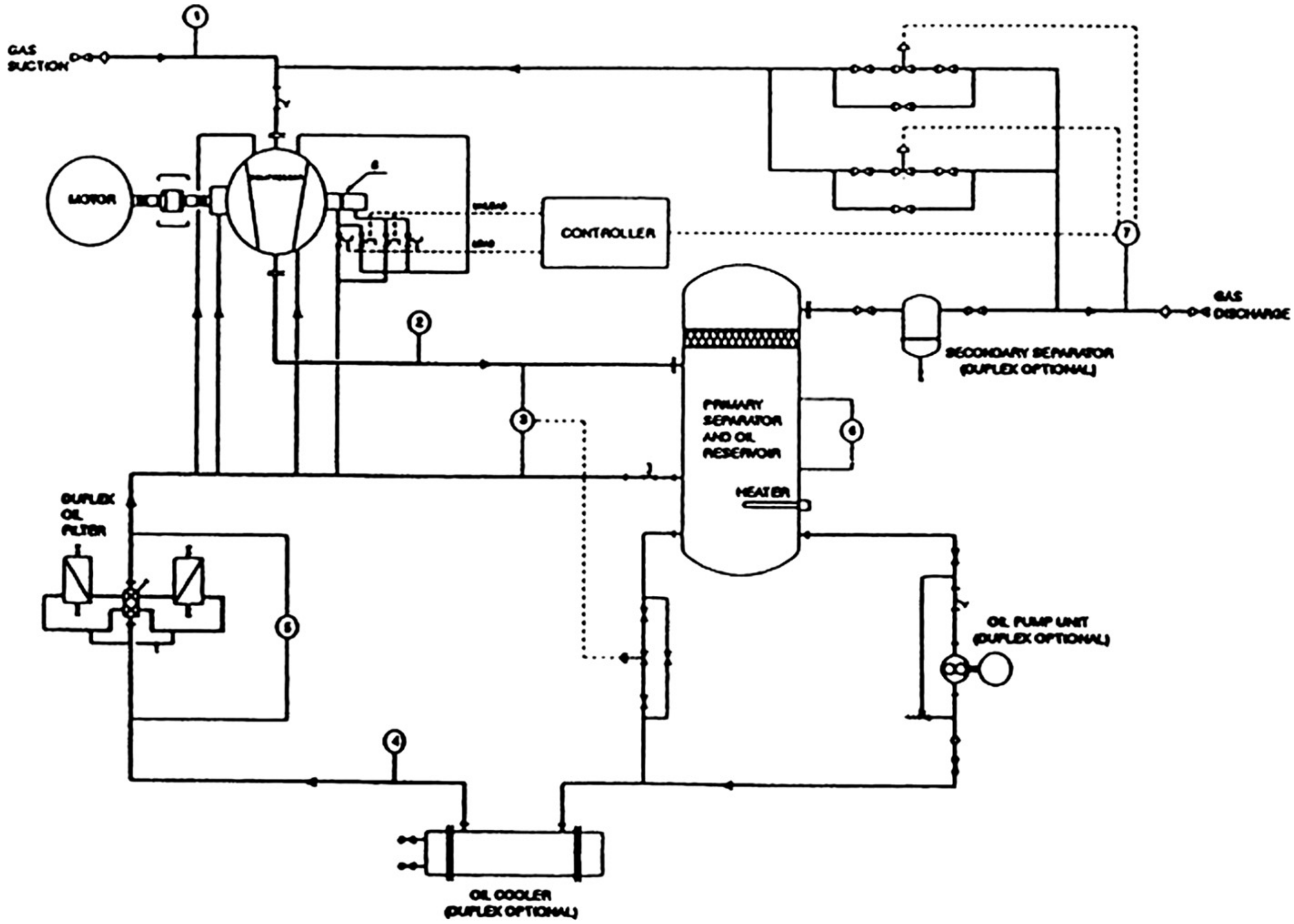
Prior to purchase of a flooded screw compressor, the entire downstream process system must be examined to confirm that small quantities of oil will not affect coolers, reactors, etc. The efficiency of the oil separation systems used is high (99.9%). However, upsets can cause oil to be transferred downstream. A typical oil separation system used for a flooded screw compressor is shown in Figure 3.7.13.

The vendors who are quoting for a screw compressor should be consulted regarding modification to the standard system that can be made to meet requirements. Available options are:

- Separate lube and seal oil injection systems
- Increased separator vessel retention time
- Self cleaning separator vessels upstream of the compressor

However, this activity must take place early in the project (phase 1), since the cost of the unit will increase.

The other concern with flooded screw compressors is oil degradation caused by interaction of the process gas with the injected oil. Please refer again to Figure 3.7.13. The retention time of the oil in the separator is typically 1–1$^1/_2$ minutes. Therefore, any contaminant in the process gas may not have sufficient time for removal. There have been some bad experiences, particularly in gas recovery applications where a sour, dirty gas (containing asphaltenes) has required frequent oil and oil filter change out. Our recommendation is to discuss these facts in detail with the vendor prior to the order (phase 1 of project) and require references detailing situations where the equipment is operating. It has been our experience that this



1. Suction instrumentation
2. Discharge instrumentation
3. Oil/gas differential pressure instrumentation
4. Oil temperature/pressure instrumentation
5. Oil filter differential pressure instrumentation
6. Oil level instrumentation
7. Skid discharge instrumentation
8. Capacity control slide valve

Fig 3.7.13 • Oil separation system (Courtesy of Man/GHH)

action, unfortunately, is taken after installation of the compressor. These facts are presented in Figure 3.7.14.

In phase 1 of project:
- Confirm compatibility of oil with process
- Confirm oil will not be degraded by process gas
- Visit installations to confirm vendor claims

Fig 3.7.14 • Flooded screw reliability

Performance relationships

All screw compressors are positive displacement compressors. Performance of screw compressors is calculated using the adiabatic process. An adiabatic compression process assumes that no heat is lost in the compression process. The path of compression is defined by:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\left(\frac{K-1}{K}\right)}$$

Figure 3.7.15 defines the performance relationships used for screw compressors.

Screw compressor performance relationships
- CFM = (RPM) (DM^3) (L/DM) (C_0) where:

DM = male rotor diameter – inches
L/DM = length to diameter ratio of male rotor
C_0 = rotor profile constant
4 + 6 profile = 15.853
3 + 4 profile = 18.082
4 + 6 ‘A’ profile = 17.17
4 + 6 ‘D’ profile = 16.325
3 + 4 ‘A’ profile = 19.372
6 + 8 ‘D’ profile = 10.364

Fig 3.7.15 • Performance relationships

- $P_2 = P_1 \frac{(T_2)}{(T_1)}^{(k/k-1)}$
- $V_2 = V_1 = \frac{(T_2)}{(T_1)}^{(1/k)}$

where : $P =$ PSIA

$T = {}^\circ R$

${}^\circ R = {}^\circ F + 460^\circ$

$V =$ volume – cubic ft / min

- $\text{Adiabatic H.P.} = \dfrac{P_1\ Q_1\ (P2/P1)}{229 \frac{(K-1)}{K}}$
- $\text{BHP} = \dfrac{\text{Adiabatic H.P.}}{\text{Adiabatic efficiency (TOTAL)}}$

Where: Adiabatic efficiency total = (adiabatic efficiency) (mechanical efficiency)

- $T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)}$
- $\text{Adiabatic efficiency} = \dfrac{\Delta\ \text{T adiabatic}}{\Delta\ \text{T actual}}$

Fig 3.7.16 • Performance relationships

The relationships in Figure 3.7.16 can be used to either estimate screw compressor performance for selection purposes, or to determine field performance. When determining screw compressor field performance, a trend of delivered flow is most useful since all screw compressors are positive displacement compressors and will deliver a constant volume flow. Reduction of flow is an indication of an increase of slip (internal leakage). Care must be used in trending flow however. Rotor speed, suction throttle valve, slide valve and bypass valve (if supplied) position must be constant.

Trending adiabatic efficiency is another useful indicator of performances. Again, care must be taken to ensure oil injection (flooded screws), jacket water temperatures and rotor cooling flows (dry screws), in addition to valve positions, are constant. These facts are presented in Figure 3.7.17.

Screw compressor field performance check guidelines

- Trend volume flow
  Note: ensure speed and control devices are at constant values.
- Trend adiabatic efficiency
  Note: ensure oil injection rates, jacket cooling temperature and rotor cooling flows and valve positions are at constant values.

Fig 3.7.17 • Field performance checks

Mechanical components

It is useful to always remember that any type of rotating machine contains five major components. It is important to know their function and monitor their condition. Screw compressors are no exceptions. The five major component systems are:

- Rotor
- Journal bearing
- Thrust bearing
- Seals
- Auxiliary systems

Please refer to Figure 3.7.18 which shows a typical dry screw compressor, and Figure 3.7.19 which shows a typical flooded screw compressor. Both figures are top (plan) views, which will be used to describe the function of the major mechanical components.

Rotor

As previously shown, there are various screw profiles available. Their use depends on the process condition and type of screw compressor (dry or flooded). The most common profile is the asymmetric 4 + 6 (4 lobe male rotor and 6 lobe female rotor). The major mechanical concern is rotor deflection. Therefore, the L/D (rotor supported length divided by its diameter) and number of lobes will vary directly with capacity and pressure ratios. High flow rates may use only 3 male lobes and high pressure ratio applications may employ 6 male lobes.

We recommend that the contractor (or end user) requires the vendor to produce references to show their rotor experience during the bidding phase. These references should be contacted to confirm that reliable field operation for the proposed rotor configuration has indeed been experienced.

Rotor speeds are low relative to centrifugal compressors. Consequently, screw compressor rotors normally are rigid (they operate below their first critical speed). Dry screw compressor tip speeds are 160m/sec (350 ft/sec) and flooded screw speeds are 30–50 m/sec (100–175 ft/sec).

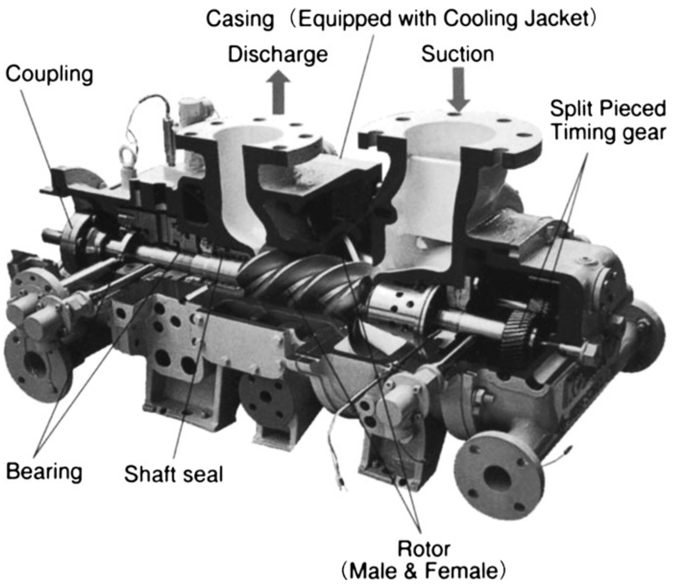


Fig 3.7.18 • Dry screw compressor (Courtesy of Kobelco – Kobe Steel Ltd)

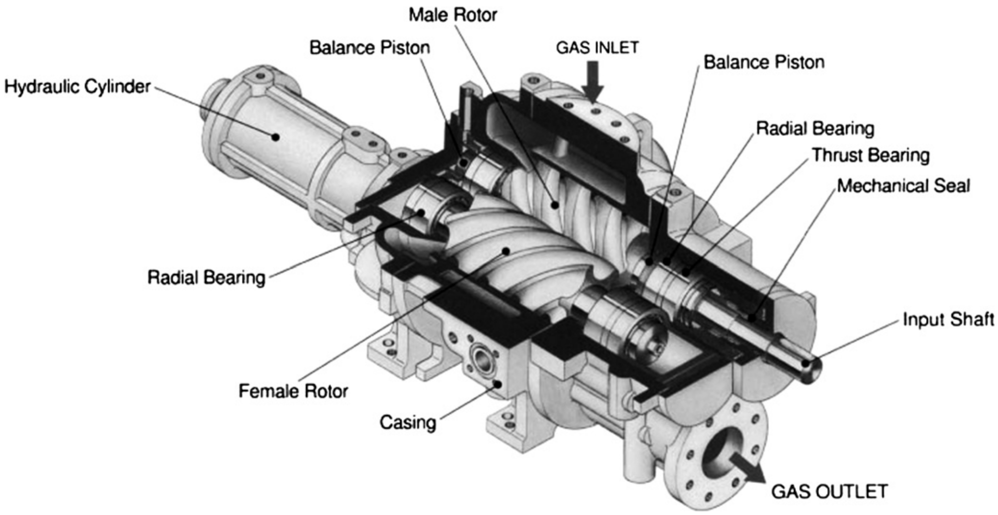


Fig 3.7.19 • Flooded screw compressor (Courtesy of Kobelco – Kobe Steel Ltd)

Journal bearings

The usual journal bearing selected are roller type anti-friction bearings. Larger compressors, above 220 kW (300 BHP) use sleeve or tilt pad bearings.

It is important to confirm proper bearing size and selection during the bidding process. Anti-friction bearings should be checked to confirm that DN number (diameter of bearing bore in mm multiplied by shaft speed) is within acceptable limits and the L-10 life is a minimum of 25,000 hours. Sleeve bearing loads (based on projected area) should be less than 1,725 $\frac{kN}{m^2}$ (250 PSI).

Thrust bearings

Small screw compressors use angular contact anti-friction bearings. As stated above, DN and L-10 life should be confirmed to be acceptable during the bid phase.

Larger screw compressors above 220 kW (300 BHP) may use plain, tapered land or even tilt pad thrust bearings for larger size, above 750 kW (1000 BHP). Regardless of type, bearing loads should be less than 1,725 $\frac{kN}{m^2}$(250 PSI). Larger screw compressors use balancing devices to control the thrust load (balance pistons).

Timing gears

As previously mentioned, dry screw compressors require external timing gears to ensure proper rotor clearances. An increase in rotor clearance of 0.01 mm (0.0004") can reduce efficiency by 1%. The timing gears are precision-ground gears that require continuous lubrication. Small compressors, less than 220 kW (300 BHP), can use self-contained ring oil lubrication. Larger compressors will require a pressurized lubrication system.

Flooded screw compressors do not require external timing gears, but do require significant amounts of oil, which must be injected into the screws (usually in the bottom of the casing). As an example, a typical oil injection rate is 27 liters per minute (7 GPM) per 170 m^3/hr (100 acfm) flow rate for air compressors.

Sealing devices

Dry screw compressors require that sealing devices be installed at each end of each rotor. This is necessary to prevent timing gear oil from entering the process stream, and process gas from entering the lube oil system. Flooded screw compressors only require a sealing device on the drive screw coupling end.

Regardless of type of screw compressor, there are many sealing alternatives available. Some of the common types are:

- Labyrinth
- Restrictive carbon ring type
- Mechanical (liquid) seals
- Dry gas seals

Auxiliary systems

Small screw compressors can be supplied with self-contained lubrication (ring oil) and sealing systems. Depending on operating conditions, a jacket cooling system (dry screws) may be required.

Larger screw compressors will employ a pressurized lubrication system and perhaps a liquid or gas seal buffer system.

All flooded screw compressors will require an oil separation system, which is usually combined with the lubrication system. A typical P&ID for such a system is shown in Figure 3.7.20.

Experience has shown that the reliability of the bearing and seal component is a direct function of auxiliary system component selection and design. The auxiliary system(s) are the only major source of potential cost reduction for screw compressor vendors. Since screw compressors are relatively new, their specifications have not reached the sophistication of those for reciprocating and centrifugal compressors. We recommend that all auxiliary systems be thoroughly reviewed in the bidding phase, with references required during the co-ordination meeting for large compressors. Special care should be given in the pre-order phase to gas/oil separator retention time, and vendor experience with this item in similar applications.

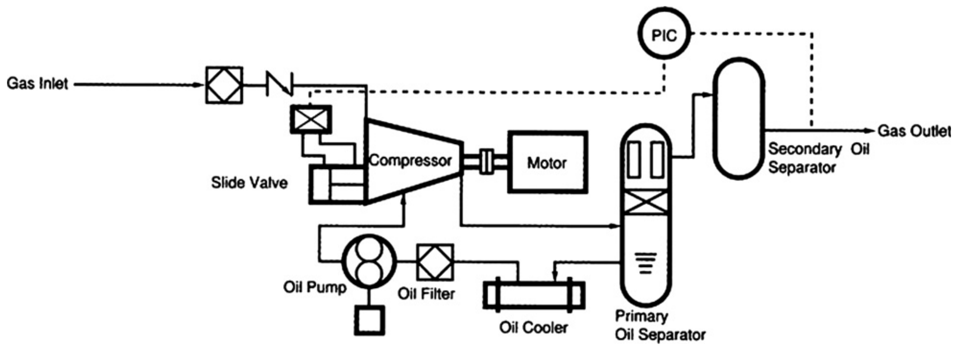


Fig 3.7.20 • Typical oil flooded screw oil separation system (Courtesy of Kobelco – Kobe Steel Ltd)

Capacity control

Since screw compressors are positive displacement compressors, their inlet volume flow is constant. If the speed of the rotor remains constant, like reciprocating compressors, various methods are used to allow capacity to be varied. The four control methods used are presented in Figure 3.7.21.

Screw compressor capacity control methods:

- Variable speed
- Suction throttling
- Slide valve
- Bypass

Fig 3.7.21 • Capacity control methods

Variable speed

As shown in the previous section, the capacity of the compressor is a function of rotor profile and rotor speed. Variable speed is the preferred method of capacity control for oil-free compressors. It is not the preferred method for oil-flooded compressors, since the levels of injection oil would have to be varied with speed, and the slide valve method is available.

Suction throttling

Suction throttling enables the mass flow of the screw compressor to be controlled by changing the inlet gas density. Care must be used to ensure that the maximum compressor ratio is not exceeded, since this could cause rotor deflection and/or excessive discharge gas temperature.

Slide valve

Oil injected screw compressors are usually fitted with a slide valve that allows the suction volume to be varied between 10 and 100%. A slide valve inside the screw case housing is shown in Figure 3.7.22.

This stepless method of capacity control maintains high efficiency and is the main advantage of oil-injected screw compressors. The slide valve method has recently been employed in oil-free compressors for a limited number of applications.

The operation of a slide valve is shown in Figure 3.7.23. The slide valve is an axial, moveable segment of the casing cylinder wall. As it is moved from the suction end, towards the discharge end, flow is bypassed to the suction. Oil is injected through ports in the valve, so cooling the recycled flow. This method of control is the most efficient after the variable speed method.

Bypass control

This method utilizes an external control valve to bypass excess gas back to the suction. The recycled gas must be cooled. The control of the bypass valve can be either pressure or flow. Bypass control is the most inefficient method of capacity control.

Selection guidelines

As previously mentioned, screw compressors have a flow range of 170–68,000 m^3/hr (100–40,000 acfm) and can produce compression ratios as high as 25:1. In order to optimize compressor efficiency, accurate process data must be made available to the quoting vendors. The required input data is the same as for other types of compressors, and is shown in Figure 3.7.24.

It is most important to accurately define gas contaminants (asphaltenes, etc.) and the percent per unit volume. Levels of

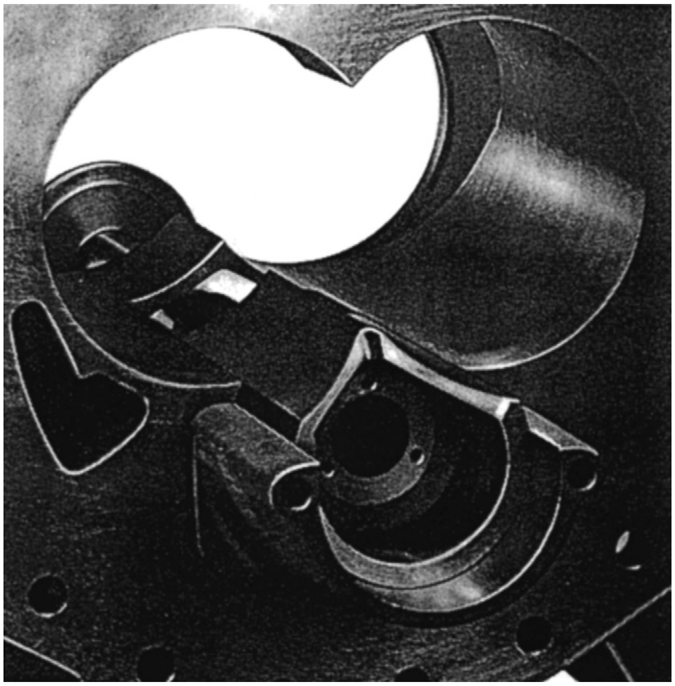


Fig 3.7.22 • Slide valve control

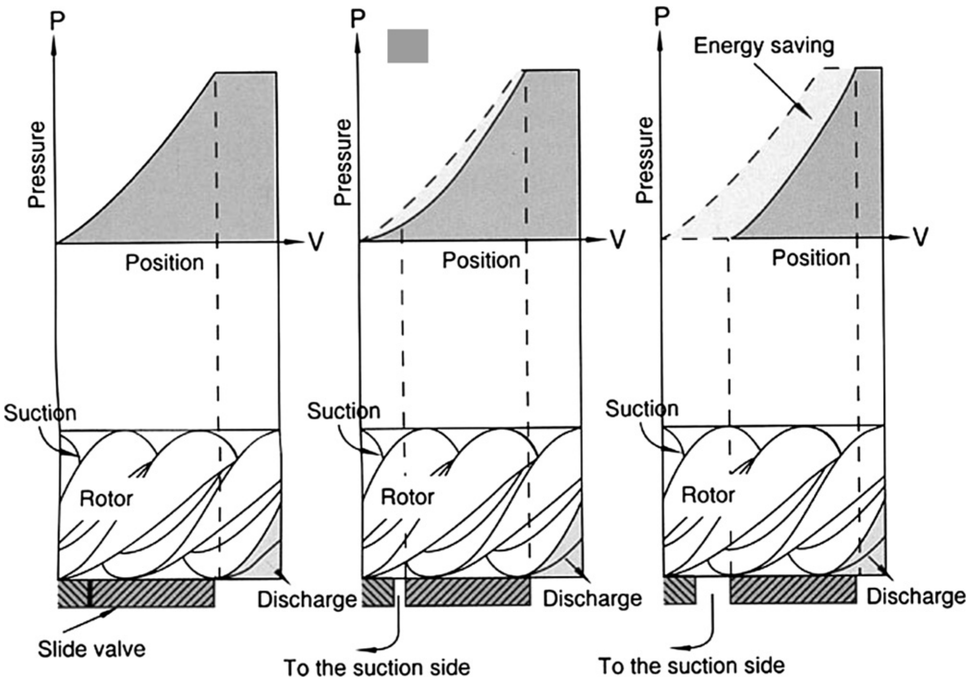


Fig 3.7.23 • Slide valve operation
(Courtesy of Kobelco e Kobe Steel Ltd)

- P_1, P_2
- T_1
- Gas analysis (for each case)
- Gas contaminents
- Flow rate
- Off design flow requirements

Fig 3.7.24 • Required process data

sour gas (H_2S) must be accurately defined, since they will determine material types and whether a flooded screw can be used.

Dry or wet screw?

As discussed in this chapter, the technical and cost advantages of flooded screw compressors have made them the compressor of choice. However, special modifications may be required for them to be used in sour gas service. The low retention time of oil in the separation vessel (60–90 seconds) does not allow sour gas components to be vented from the reservoir oil. This can lead to oil contamination, screw component damage and the need for frequent oil changes. Reservoir capacity can be increased, but at an extra cost. The use of dry screw compressors should be considered in sour gas service, and this option should be evaluated against properly designed oil flooded compressors. Vendor experience lists should be required and checked.

Condition monitoring

Condition monitoring requirements for screw compressors should follow the principle of component condition monitoring. The following major components should be monitored:

- Rotor
- Journal bearing
- Thrust bearing
- Seals

The suggested minimal condition monitoring requirements are presented in Figure 3.7.25.

Screw compressor minimum condition monitoring requirements:
- Performance (rotor)
- Flow trend
- Efficiency check
- Manual vibration readings (accelerometers/velocity)
- Thrust bearing temperature measurement (RTDs)

Fig 3.7.25 • Minimum condition monitoring required

Best Practice 3.8

Centrifugal horizontal split casing compressor nozzles should always be located on the bottom of the casing for lowest mean time to repair (MTTR).

Be sure to justify either horizontal split with bottom nozzles or radial split casing with top nozzles to minimize time to repair (MTTR) during the pre-feed phase of the project:

The time to disassemble a horizontal split case with bottom nozzles is 1 – 2 days maintenance time.

The time to remove the inner casing from a radial split compressor is 2 – 3 days maintenance time.

The time to disassemble a horizontal split case with top nozzles considering piping removal and re-assembly without exceeding external piping forces and moment limits is 7 days or more maintenance time.

Lessons Learned

In order to minimize construction costs on motor drive, back pressure steam turbines or gas turbine drivers, the use of a horizontal split centrifugal compressor with top nozzles will result in internal component removal times that are 2 to 3 times longer.

Benchmarks

Since the 1990s, I and Forsthoffer Associates, Inc. have used the following best practice for selecting centrifugal compressor casing types. It saves 4 – 5 days maintenance time, and therefore increases daily revenue for this time, potentially in excess of 10MM additional revenue.

- If the driver is a condensing steam turbine use bottom nozzles since the height of the compressor platform will be set by the condenser size and condensate pump NPSH requirements.
- For all other types of drivers that are planned to be installed at ground level, use barrel type compressors with top nozzles – the capital cost compared to horizontal compressors is approximately 10% increase.

B.P. 3.8. Supporting Material

In this section details concerning the compressor case and the stationary internals will be covered.

We will first define the functions of the casing as follows:

- Flow guidance
- Positive sealing
- Full rotor support

The different types of turbo-compressor cases will be defined:

- Single stage
- Multi-stage
- Horizontal split
- Radial
- Integral gear

Nozzle orientation will also be covered. Emphasis will be on minimal disturbance to the process piping during compressor disassembly. Casing fabrication types will be discussed highlighting cast and fabricated casings. Casing stresses and deflections due to internal forces and external loads will be covered.

Casing functions

The major functions of any compressor case, regardless of type are:

- To provide support of all bearing seals and stationary internals
- To provide positive containment of the process fluid
- To provide efficient guidance of the process fluid

Often, the support function of the casing is not mentioned when stating the major casing functions. Excessive casing forces caused by the process piping or foundation can cause bearing, seal and rotor failures. Positive containment of the process fluid is accomplished by the design and manufacture of leak-tight joints. All casings are hydrostatically tested at 1.5 times their maximum working pressure prior to shipment. Gases containing more than 50% H_2 require a gas test at rated pressure, in addition to the hydrostatic test.

Casing joints are either horizontal or vertical (radial). Horizontal joints are usually metal to metal, as opposed to vertical joints, which always incorporate an 'O' ring, although some manufacturers do incorporate an 'O' ring in their horizontal design. In my opinion, metal to metal horizontal joints are preferred to ensure that a leak will not develop if the 'O' ring is damaged.

An effective design feature or modification for horizontal joints is to 'relieve' the joint. That is, to minimize the joint contact area, and therefore increase the joint contact stress by machining the horizontal faces in certain areas of the casing flange. The machined areas are usually 0.032"–0.062" (0.8–1.5mm) deep. A plan view of a typical relieved casing is shown in Figure 3.8.1. The relief can be in either the top or bottom half of the casing joint.

Barrel or radially split casings are used for gases with high H_2 content, or for pressures above approximately 600 PSI. The advantages of the barrel type compressor are:

- Minimum casing joint area
- Removal of internals with removal of process piping

The disadvantages are:

- Special tools are required for disassembly
- Additional maintenance space is required for removal of internals

When a two-case application uses barrel type compressors, it is advisable to design the driver with double shaft extensions for ease of maintenance.

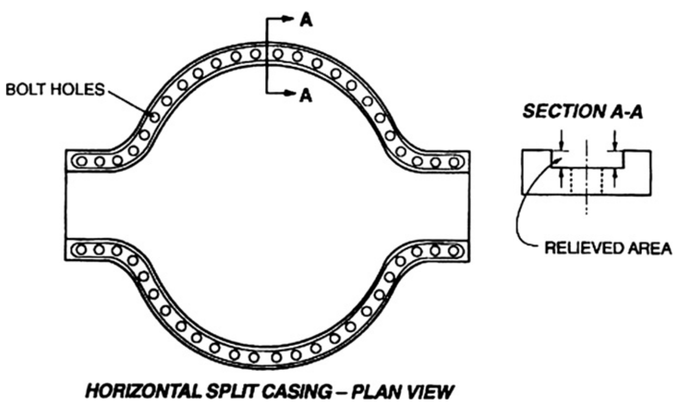


Fig 3.8.1 • Relieved horizontal joint – horizontal split casing – plan view

Casing types

The available types of turbo-compressor casings are:

- Single stage radial split
- Single state integral gear
- Multistage radial split
- Multistage horizontal split
- Multistage integral gear

Figure 3.8.2 shows an example of a single stage, radially split casing.

This type of casing is typically used for high pressure applications, and for gases with high hydrogen content. Its design is similar to that of a single stage overhung pump. The design shown is 'front pullout', that is, the impeller must be removed for seal and/or shaft removal. Other designs allow back pullout. This type offers the advantage of removing the internals without having to remove the process piping.

A single stage casing for an integral gear centrifugal compressor is shown in Figure 3.8.3 (Sundyne Compressor).

This type of compressor is used in low flow, high head applications. The integral gearbox (not shown) enables this type of compressor to operate at speeds above 30,000 rpm. The casing is cast and is also a 'back pullout' design, which allows removal of internals without affecting the process piping.

An example of a multi-stage radially split or barrel compressor is shown in Figure 3.8.4.

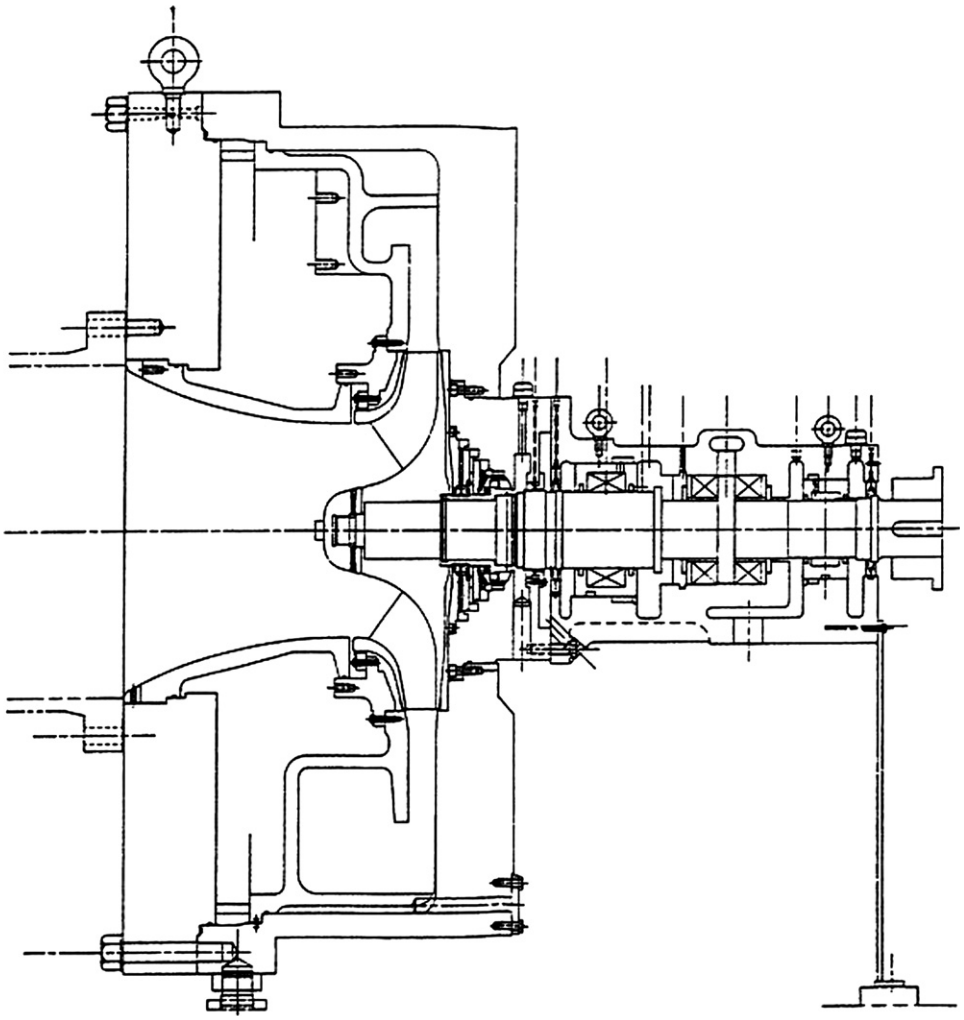


Fig 3.8.2 • Single stage overhung turbo-compressor (Courtesy of A-C Compressor Corp.)

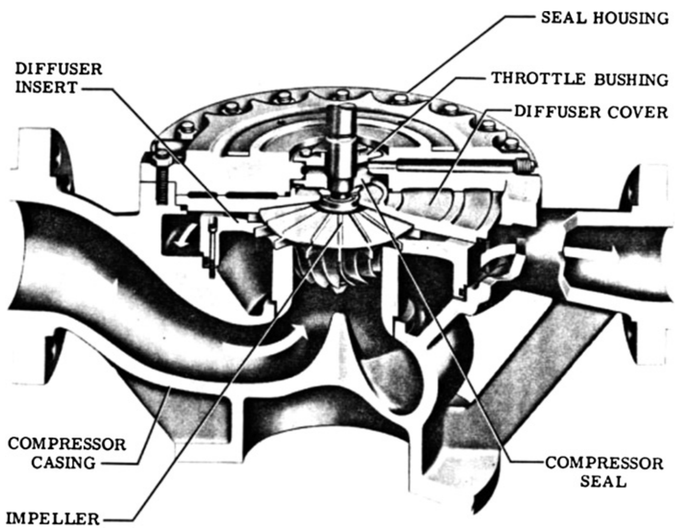


Fig 3.8.3 • Single stage high-speed compressor (gear box and motor removed) (Courtesy of Sundstrand Corp.)

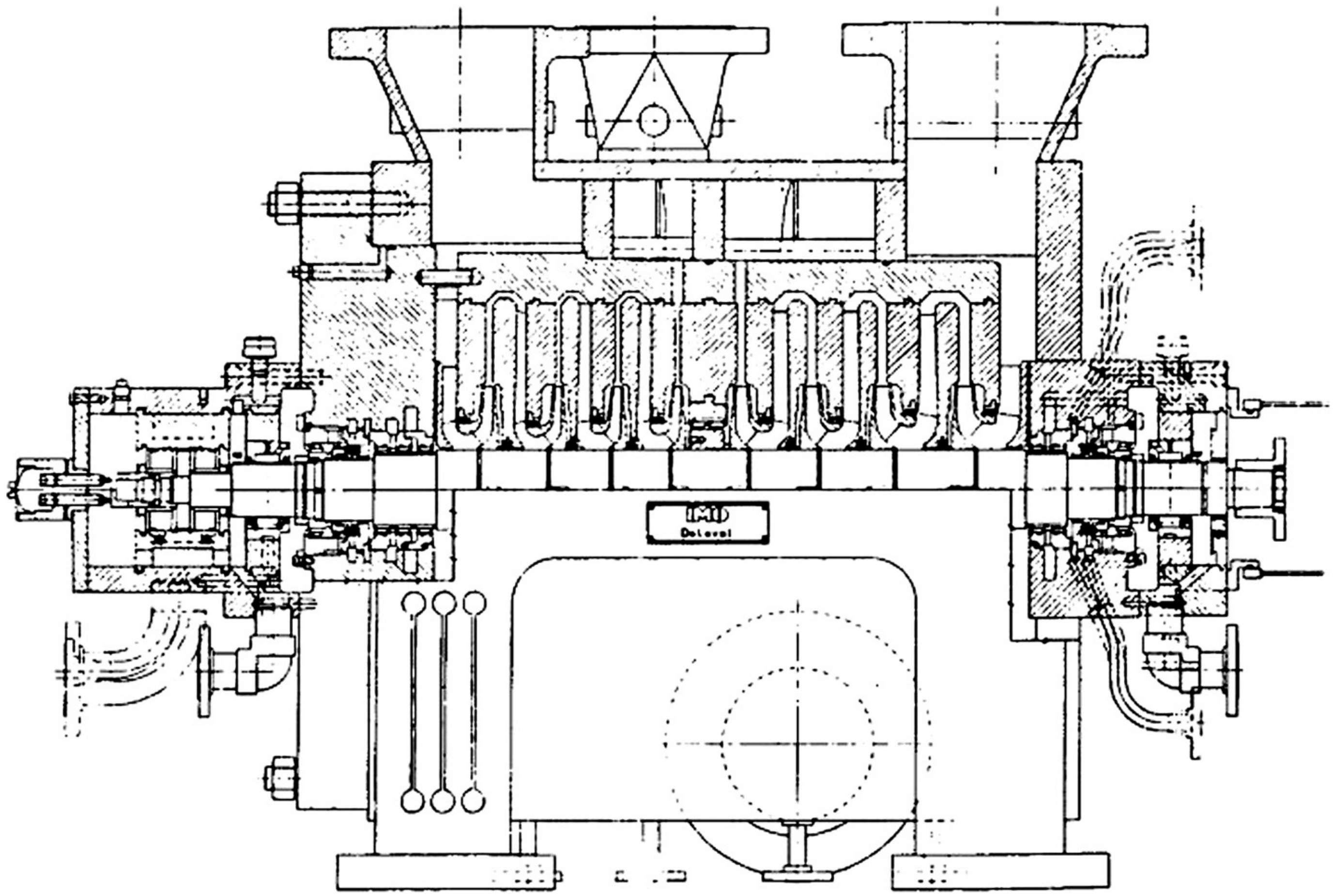


Fig 3.8.4 • Multistage barrel compressor (Courtesy of IMO Industries, Inc.)

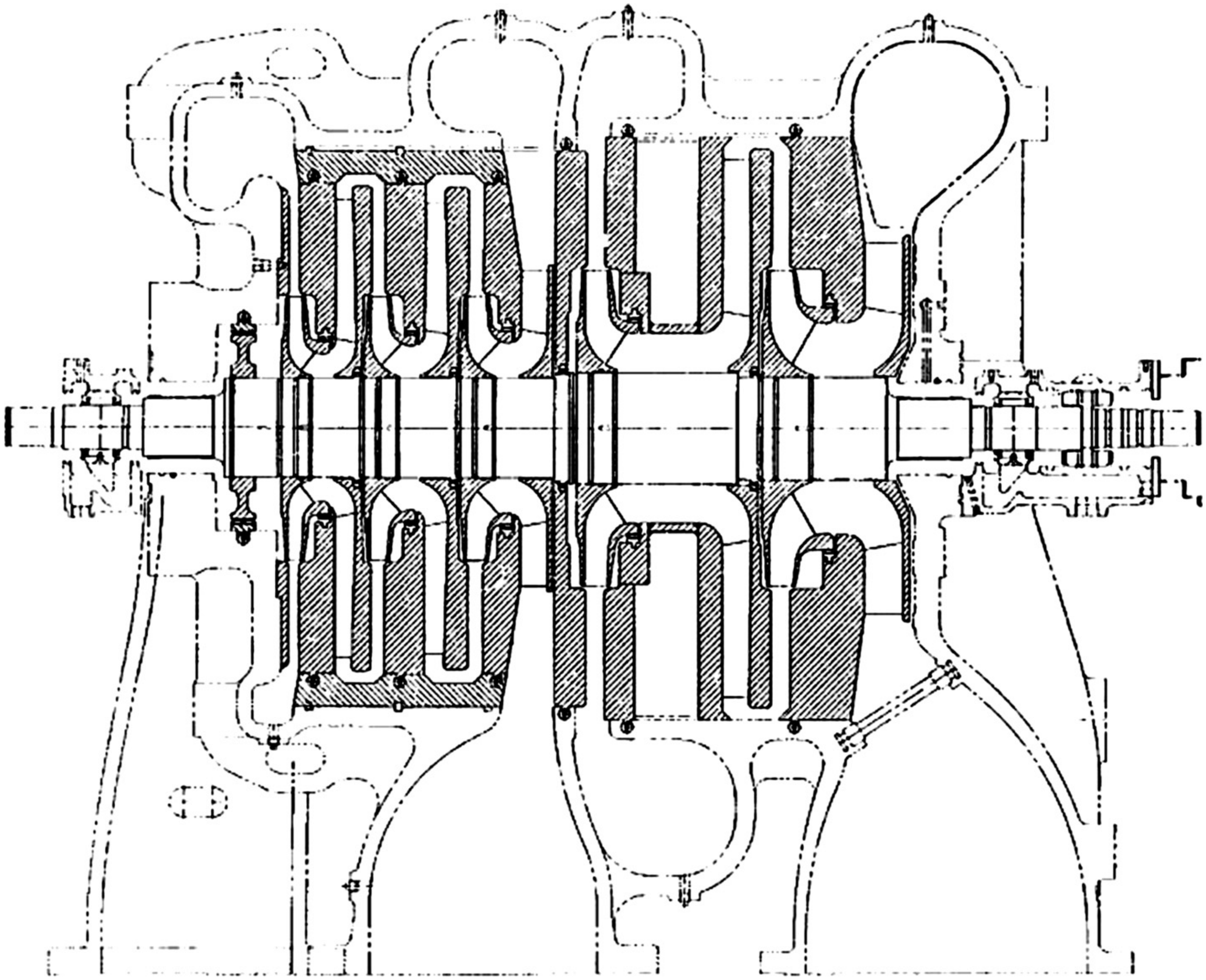


Fig 3.8.5 • Multistage intercooled turbo-compressor (Courtesy of IMO Industries, Inc.)

Figure 3.8.5 depicts a multistage, horizontally split, centrifugal compressor casing. Most multistage compressors are horizontally split and should be used whenever process pressures and/or gas compositions permit. The advantage of a horizontally split casing should be ease of removal of internal parts. The example shown meets this requirement, since all process piping is located on the lower half of the casing. When process piping is connected to the top half of the compressor casing, the maintenance advantage is lost.

Readers are advised to become involved in the initial process plant design, to ensure that bottom connected horizontal compressor cases are installed if at all possible. A cost analysis should be performed to determine the additional cost for bottom connections. Note: compressor vendors require five (5) pipe diameters straight runs to process connections. When performing a cost analysis, be sure to include the cost penalty for the additional removal and assembly time for top mounted piping. Also be sure to include the revenue lost per day of additional downtime if the compressor is unspared. Typical additional times for piping removal and assembly are 2–3 days or longer. If the cost is prohibitive, another option would be a barrel compressor with top connections.

Best Practice 3.9

During the pre-bid project phase (before priced vendor bids are received), screen for proven impeller or blade row flow, head coefficient and tip speed experience on similar gas compositions.

Centrifugal and axial compressors are custom designed, but should incorporate proven impellers or blades and gas path parts.

The success of the Factory Acceptance Test (FAT), field start-up and process life safe and reliable compressor reliability is directly related to impeller, blade and gas path component integrity.

As a result, vendor requirements should be noted in the invitation to bid (ITB), to provide the following parameters for each stage (impeller or blade row). This will ensure optimum FAT results and field safety/reliability:

- Flow coefficient experience
- Head coefficient experience
- Tip speed experience
- Impeller inlet (eye) Mach number experience
- Above parameters for similar gas compositions

Lessons Learned

The failure to review for impeller/blade experience prior to vendor acceptance can result in extended FAT time, delayed field start-up and continual safety and reliability field issues.

Accepting vendor-proposed impellers and blades without a review of their experience and of the relevant design details has led to many unexpected surprises during factory acceptance performance testing and field start-ups.

Once impellers or blade rows are designed and operating, field changes are difficult, time consuming, will produce revenue loss and are difficult to confirm, since field instrumentation does not have the same accuracy as FAT instrumentation.

Benchmarks

I have used this best practice since the mid-1970s to achieve problem free FATs, smooth start-ups and optimum centrifugal and axial compressor safety and reliability (of over 99.7%).

B.P. 3.9. Supporting Material

The review of each impeller stage or blade row for vendor installed 2 year operating experience during the pre-bid stage of the project requires each quoting vendor to provide the following parameters for each quoted impeller or blade row:

- Flow coefficient map of all experienced flow coefficients
- Head coefficient map of all experienced head coefficients
- Impeller tip speed, map of all similar flow and head coefficient impellers

Best Practice 3.10

Limit closed centrifugal compressor head per impeller values for optimum impeller performance and reliability in the project specification.

Limiting centrifugal compressor impeller head per stage will ensure the optimum flow range for each impeller and the overall compressor performance flow range. In addition, it will ensure that all impeller stresses are within safe, proven limits that will result in failure-free impeller life.

Limit heavy gas applications (molecular weights above 40) to 24 kJ/kg or 8000 FT-lbf per impeller.

Limit other services below 40 molecular weight to 3000 meters or 30 kJ/kg or 10,000 lbm per impeller.

In applications where the use of these head limits forces the compressor into 2 cases FT-lbf/lbm, be prepared to justify this requirement based on the loss of daily revenue. Reducing the head per impeller in the field will either require reduced operating speeds and proportional lower revenue or a new compressor train design.

Do not accept impeller heads above these values even if vendor experience is proven.

Lessons Learned

Failure to meet predicted performance curve flow limits beyond the design point and field impeller failures have occurred when the above head limits were exceeded in the compressor design.

Centrifugal compressor head vs. flow performance curves fell off sharply in head and flow beyond the rated point, but had to be accepted since the centrifugal curve shape is not guaranteed. This reduction in flow means a lower production rate for the life of the compressor, which amounts to a significant loss of revenue.

Closed impeller failures in the impeller cover, hub side plate and scallop failures (between the impeller vanes at the outside diameter on the cover or hub) were experienced when the head per impeller limits were exceeded.

Benchmarks

I have used this best practice since the mid-1970s and it has resulted in:

- Meeting or exceeding predicted compressor predicted curve shape beyond the rated point
- Fault-free closed impeller performance (zero compressor impeller failures)

B.P. 3.10. Supporting Material

The use of these head per impeller limits, prior to the priced proposal submission, allows the vendors to re-select and definitely meet these requirements. Do not be persuaded by experience charts, etc. Remember most importantly (see BP 1.1) that the objective of the machinery vendor is to design the machinery safely and properly for the warranty period, while offering the lowest possible price. The objective of the operating plant is to produce maximum product revenue for the life of the process unit (20 plus years). Today (2010), unspared centrifugal compressor downtime can exceed $5MM/day.

Best Practice 3.11

Require centrifugal compressor head rise to be a minimum of 5% in order to prevent control and protection system issues in the field.

Centrifugal compressor head rise is defined as the head at surge condition divided by the head at rated point.

The lower is the head rise, the more rapid a change in flow for a change in the head required by the process.

Review each proposed impeller head rise during the pre-bid phase, and require the vendor to re-select any impellers that have a head rise that is less than 5%.

Note that for heavy gas applications, greater than 40 molecular weight, this may be difficult and in those cases, acceptance of the highest impeller head rise available from the vendor will have to be considered, based on stated vendor experience and possible discussion with end users.

Lessons Learned

I have been involved with many surge problems related to flat (low head rise) performance curves. Flat compressor characteristic curves require rapid surge systems reaction times to prevent surge from occurring.

Benchmarks

This best practice has been used since 2000, and has resulted in optimum compressor safety and trouble free operation (reliability above 99.7%) and no continuing surge system problems.

B.P. 3.11. Supporting Material

The factors involved

The parameters necessary to define a given fluid are presented in Figure 3.11.1. Note that only two parameters are necessary to define a fluid in the liquid state since it is incompressible. On the other hand, three times that number are required to define that fluid in its vapor state, since the vapor is compressible.

Liquid (incompressible)	Gas (compressible)
Specific Gravity (S.G.)	Molecular Weight (M.W.)
Viscosity (υ)	Specific Heat Ratio (K)
	Compressibility (Z)
	Pressure (P–kPa or PSIA)
	Temperature (T–°K or °R)

Fig 3.11.1 • What factors define a given fluid

Figure 3.11.2 shows the relationships that are used to determine the head (energy) required to increase the pressure of a fluid in its liquid and vapor state. Note how much the density of the fluid influences the amount of energy required to meet a certain process requirement. When one considers that the additional amount of head produced as a centrifugal compressor's flow rate decreases from rated point to surge point is on the order of only 10%, it can be seen that a small change in gas density can result in a significant flow reduction and possibly compressor surge.

The effect on turbo-compressor pressure ratio

The pressure ratio produced by a dynamic compressor is affected by gas density. Figure 3.11.3 shows that, for a given compressor flow and speed, the head produced by a dynamic compressor is essentially constant. Therefore, any change in M.W., T, K or Z will change the pressure ratio produced. This information is presented in tabular form for changes in molecular weight and inlet gas temperature.

The effect on the compressor head

It is commonly thought that dynamic compressor head produced is always constant for a given flow rate and speed. Figure 3.11.4 presents this fact for the same compressor operating on different gases (O_2 and N_2).

This statement is not true for a fluid in the vapor state since head in a dynamic compressor is produced by blade velocity and gas velocity. Gas velocity will change will change with gas

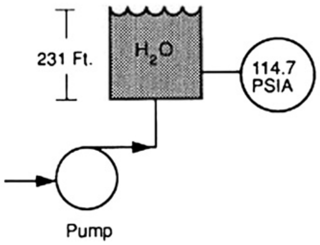
Fluid head

LIQUID

$$\underset{(\text{Ft.})}{\text{HEAD}} = \frac{2.311 \times \Delta P}{\text{S.G.}}$$

Water

ΔP = 100 PSI
HEAD = 231 Ft.
P_1 = 14.7 PSIA , T_1 = 100°F



GAS

$$\underset{(\text{Ft.})}{\text{HEAD}} = \left[\frac{1545}{\text{M.W.}}\right](T_1)\left(\frac{K}{K-1}\right)(Z)\left[\frac{P_2}{P_1}^{\left(\frac{K-1}{K}\right)} - 1\right]$$

Nitrogen

ΔP = 100 PSI
HEAD = 86,359 Ft.
P_1 = 14.7 PSIA , T_1 = 100°F

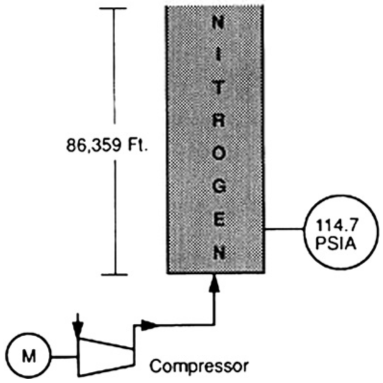


Fig 3.11.2 • Fluid head

THE EFFECT OF A GAS COMPOSITION AND TEMPERATURE CHANGE ON THE TURBO-COMPRESSOR PRESSURE RATIO

HEAD$_{\text{ISENTROPIC}}$ IS RELATED TO PRESSURE RATIO BY:

$$HD_{ISEN} = \left(\frac{1545}{M.W}\right)(T_1)\left(\frac{K}{K-1}\right)(Z)\left[\frac{P_2}{P_1}^{\frac{K-1}{K}} - 1\right]$$

ASSUMING HD$_{\text{ISEN}}$ IS CONSTANT FOR A GIVEN FLOW,

$$\frac{P_2}{P_1} = \left(1 + \frac{(HD_{ISEN})\ (M.W)}{(1545)\ (T_1)\ \left(\frac{K}{K-1}\right)\ (Z)}\right)^{\frac{K}{K-1}}$$

THEREFORE THE FOLLOWING TABLE CAN BE DEVELOPED:

EFFECT OF GAS AND T CHANGES ON PRESS. RATIO		
MOLECULAR WGT.	INLET TEMP.	PRESSURE RATIO
INCREASES	CONSTANT	INCREASES
DECREASES	CONSTANT	DECREASES
CONSTANT	INCREASES	DECREASES
CONSTANT	DECREASES	INCREASES

Fig 3.11.3 • The effect of a gas composition and temperature change on the turbo-compressor pressure ratio

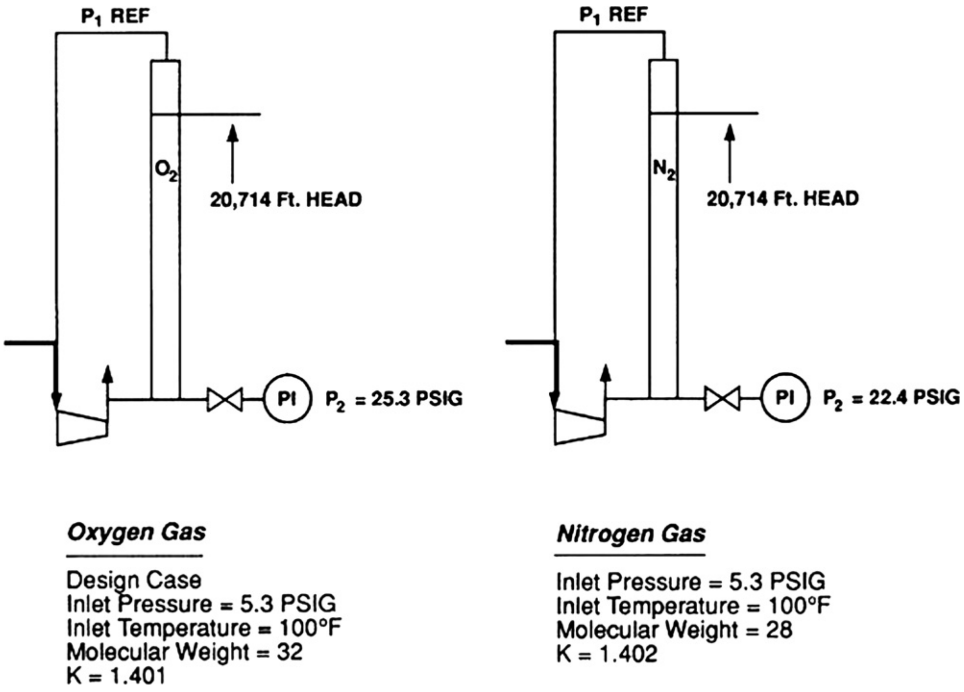


Fig 3.11.4 • The effect of gas composition change on head

density since a gas is compressible. These facts are presented in Figure 3.11.5.

The assumption that compressor head remains constant for a given flow with gas composition and temperature changes is not true because:

- Head is generated by impeller tip speed and exit velocity relative to the blade
- Gas composition and temperature changes affect the compression ratio
- Volume flow rate changes with pressure, temperature and compressibility
- Since the impeller exit area is fixed, a change in exit volume rate will produce a change in velocity

Note: For changes on the order of 20%, it is common practice to assume head is constant for a given flow and speed

Fig 3.11.5 • The effect of a gas composition and temperature change on turbo-compressor head

Please refer to Figure 3.11.6 which shows the relationship between gas velocity (V_{rel}) blade tip speed (U) and tangential gas velocity in a centrifugal compressor.

Since the head produced by any dynamic impeller is proportional to blade tip speed and gas tangential velocity, reduced gas velocity through the impeller (V_{rel}) will increase the head produced as shown in Figure 3.11.6. This is the result of increased gas tangential velocity for a given impeller diameter and speed. As shown in Figure 3.11.5, gas velocity (V_{rel}) will vary with gas density.

Figure 3.11.7 presents the effect of gas density changes on impeller produced head, surge point and choke point. It can be seen that curve shape is influenced by gas density changes. Therefore, a low density gas will always have a greater flow range than a high density gas.

The effect on system resistance

Figure 3.11.8 presents the effect of gas density change on the system resistance curve. A slight change in the friction drop in pipes, fittings and vessels results from a change in gas density.

The effect on turbo-compressor flow rate

The effect of gas density changes on actual mass and standard flow rates is shown in Figure 3.11.9. Note that gas density changes will change the operating point of each compressor

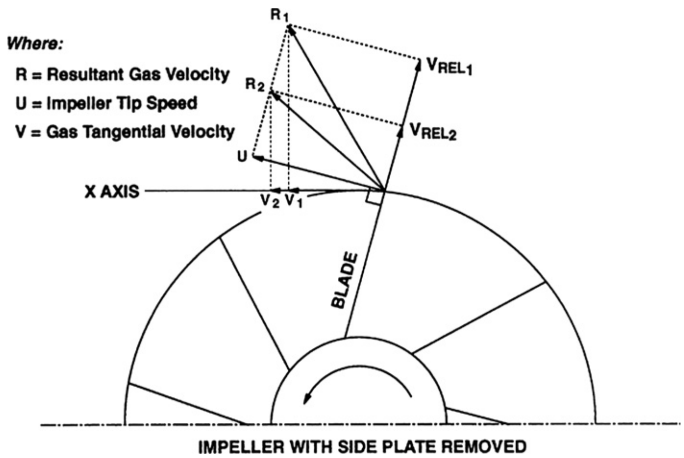


Fig 3.11.6 • Head produced α (U)(V_T)

CONDITION	VEL_{REL} EXIT	HEAD	SURGE POINT	CHOKE POINT
M.W. INCREASES	DECREASES	INCREASES	+ FLOW	- FLOW
M.W. DECREASES	INCREASES	DECREASES	- FLOW	+ FLOW
T_1 INCREASES	INCREASES	DECREASES	- FLOW	+ FLOW
T_1 DECREASES	DECREASES	INCREASES	+ FLOW	- FLOW

ABOVE CHANGES ASSUME A CONSTANT INLET ACTUAL VOLUME FLOW RATE

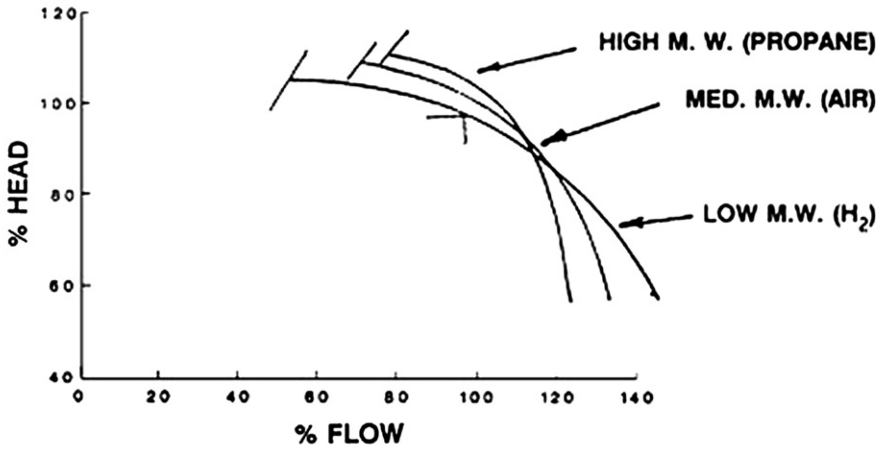


Fig 3.11.7 • Turbo-compressor impeller head change and curve shape summary

stage in a multistage compressor. Depending on the impeller selection, this change could have an adverse effect on the operation of a dynamic compressor, causing surge and corresponding high vibration, temperature, flow changes, etc.

The effect on power

As shown in Figure 3.11.10, dynamic compressor required power increases directly with gas density up to the choke flow or stonewall region of the performance curve. In the choke flow region, the head produced by the compressor approaches zero since the gas velocity is equal to its sonic velocity.

Figure 3.11.11 shows the effect on compressor section performance that results from a change in the molecular weight of the gas. As previously discussed, molecular weight changes can result in compressor stage mismatching, which can cause significant mechanical damage to the compressor train.

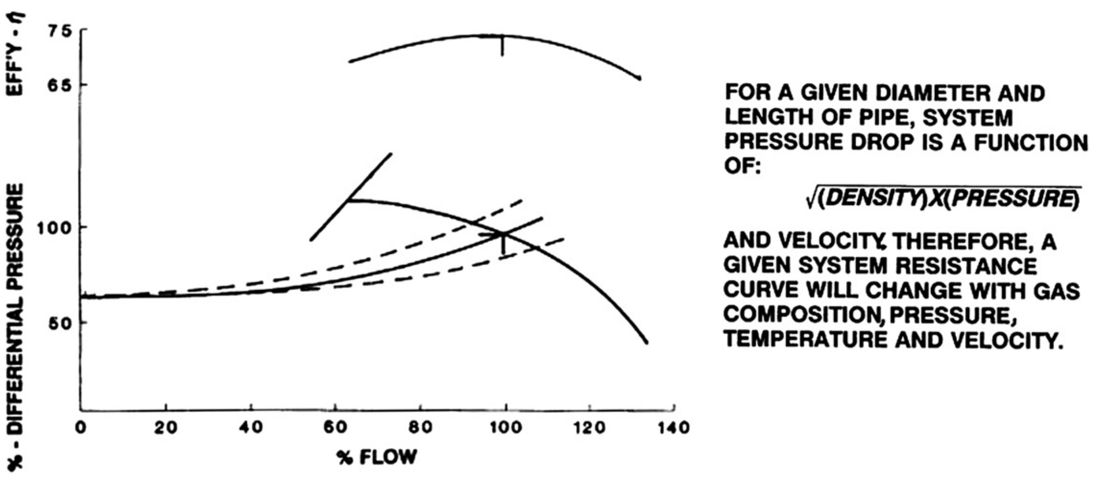


Fig 3.11.8 • The effect of process changes on the system resistance curve

- The actual volume flow rate will vary as a result of the operating point change which is the intersection of the turbo-compressor curve (pressure vs. flow) and the system resistance
- The mass flow rate (lbs/min) will be the product of the new actual volume flow rate (ft^3/min) and the gas density (lb/ft^3) at the new gas conditions (M.W., P, T, Z)
- The standard volume flow rate (scfm) will be the product of the new actual volume flow rate (ft^3/min) at its pressure temperature corrected for standard conditions (14.7 PSIA and 60°F)

Fig 3.11.9 • The effect of gas composition on turbo-compressor flow rate

$$\text{Power (kW)} = \frac{\text{Head}\left(\frac{\text{m-kgf}}{\text{kgm}}\right) \times \text{Mass flow}\left(\frac{\text{kg}}{\text{hr}}\right)}{3{,}600\left(\frac{\text{m-kgf}}{\text{min.-kW}}\right) \times \eta(\%)} + \text{Mech. losses (kW)}$$

$$\text{Power}\left[(\text{BHP}) = \frac{\text{Head}\left(\frac{\text{ft-lb}}{\text{lb}}\right) \times \text{Mass flow}\left(\frac{\text{lb}}{\text{min}}\right)}{33{,}000\left(\frac{\text{ft-lb}}{\text{Min-H.P.}}\right) \times \eta(\%)} + \text{Mech. losses (BHP)}\right]$$

Fig 3.11.10 • The effect on power

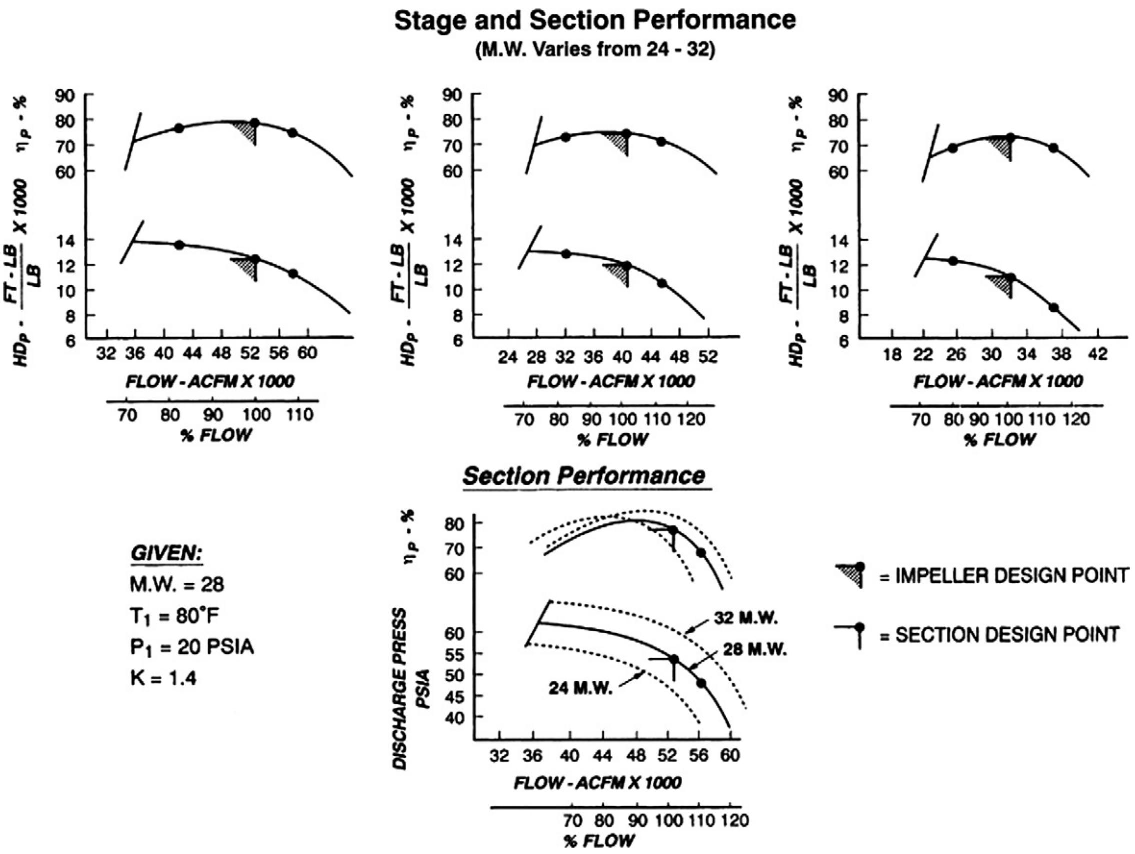


Fig 3.11.11 • Stage and section performance

Best Practice 3.12

Select individual impellers or blade rows to operate as close to best efficiency point (BEP) as possible.

This will ensure:

- Maximum overall compressor curve flow range
- Trouble free heavy gas impeller operation for gas molecular weights above 40

Ensure that each proposed impeller, when operating at the rated condition, is a maximum of +/− 10% from its best efficiency point, and +/− 5% from its best efficiency point for gases above 40 molecular weight.

Agree to review and not retain the individual vendor individual performance curves since they are proprietary.

Many compressor vendors can presently (2010) manufacture each impeller to operate at its best efficiency point when operating at the specified compressor rated operating point.

Lessons Learned

Root Cause Failure Analysis (RCFAs) of impeller failures confirm the cause to be operation in the overload region (more than 20% greater than rated flow) of the individual impeller curves.

Benchmarks

Since the mid-1980s I have implemented the best practice of review in the pre-bid phase of individual proposed performance curves for each impeller, and have required vendor re-selection of impellers if the rated operating point for each impeller was greater than +/− 10% from the impeller best efficiency point, or +/− 5% away if the gas molecular weight was greater than 40. This best practice has resulted in trouble free Factory Acceptance Tests (FATs) and compressor field reliabilities in excess of 99.7%.

B.P. 3.12. Supporting Material

See the supporting material for B.P: 3.11.

Best Practice 3.13

Gas density changes of more than +/− 20% affect centrifugal compressor performance curve (head vs. actual flow) shape. Require the vendors to re-issue the performance curve in these instances.

Impeller head is produced by the velocity of the impeller tip and the tangential velocity of the gas through the impeller (Euler's turbo equation).

Unlike pump impeller head, where the liquid is incompressible, compressor head is affected by the gas density. This is the reason that a centrifugal pump can be tested on water and, provided the field liquid is not viscous, will yield exactly the same performance head vs. flow curve in the field. By Euler's turbo equation, the tangential velocity of the liquid through a pump impeller is the same regardless of the liquid density, since the liquid is incompressible and the impeller geometry (flow area) is unchanged.

However, if the gas density in a centrifugal impeller or axial blade changes by more than 20%, the tangential velocity of the gas will significantly change and the individual impeller performance curve shape (head vs. flow) will change.

Lessons Learned

Field performance checks, which did not consider gas density change, originally confirmed that a compressor required maintenance, only to find that the internal condition of the compressor was in good condition when inspected. After this, it was discovered that the operating gas density conditions had changed by more than 20%, and a performance recheck using the vendor revised performance curve for the actual gas density showed the compressor to be in good condition.

There are many cases where field performance checks, carried out where a compressor was thought to be in need of maintenance, have showed that it was the performance curve that was in error, because the gas density had changed by more than 20%.

Only the vendor can re-issue a new overall performance curve because only the vendor is in possession of the individual impeller performance curves.

Benchmarks

This best practice has been used successfully since the mid-1990s, for field testing prior to turnarounds. In some cases, field performance where gas density has changed by 20% or more was in fact acceptable, when compared to the re-issued curves for the correct gas density.

In these cases, time-consuming compressor dis-assembly was averted, thus saving turnaround time and allowing additional production revenue (3 − 7 days depending on the type of compressor case and nozzle orientation − top nozzle orientation on horizontal split compressor cases being the longest time).

B.P. 3.13. Supporting Material

The compressor stage

To begin our discussion, let us observe a typical compressor stage shown in Figure 3.13.1.

A compressor stage is defined as one (1) impeller, the stationary inlet and discharge passages known as the inlet guide vanes, the diffuser and the seals, namely the eye labyrinth seal and the shaft labyrinth seal. Each compressor stage at a given flow and impeller speed will produce a certain amount of head (energy) and have a specific stage efficiency. It can be observed that any dynamic curve (turbo-compressor or pump) has the characteristic of producing increased energy only at a lower fluid flow assuming the inlet speed and the inlet gas angle are constant.

Before we continue, a few important facts and relationships need to be presented. These relationships are: the definition of a vector, tip speed, flow as a function of velocity and flow related to conditions and the concept of actual flow. In addition, there is the important concept of an equivalent orifice. Refer to Figure 3.13.2.

Given any impeller configuration, specific areas can be reduced to equivalent orifices: the eye or inlet area, the discharge area between any two vanes, the eye seal and the hub or shaft seal.

This concept makes it much easier to understand that, for a given area, gas flow will change directly proportional to the differential pressure and the compressor stage. It can be seen in Figure 3.13.2 that there is an optimum design velocity for the inlet of the impeller and the discharge of the impeller. These velocities are controlled by selection of a proper inlet eye area and discharge area based on the impeller flow requirement. Again, the concept of an equivalent orifice is helpful in understanding that the gas velocity is dependent on the geometry of the specific impeller. Also, the process system can be reduced to a simple orifice. In any process, the suction discharge sides of the process can be conceived as orifices placed either at the inlet or the discharge of the compressor flanges for a given flow condition.

Table 3.13.1 presents the definitions of the facts and relationships necessary for the discussion that follows. Please note

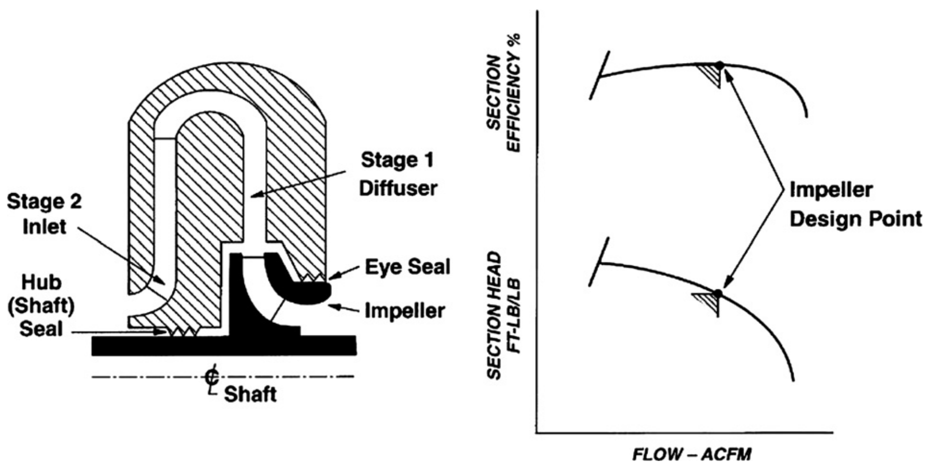


Fig 3.13.1 • The compressor stage and characteristic curve

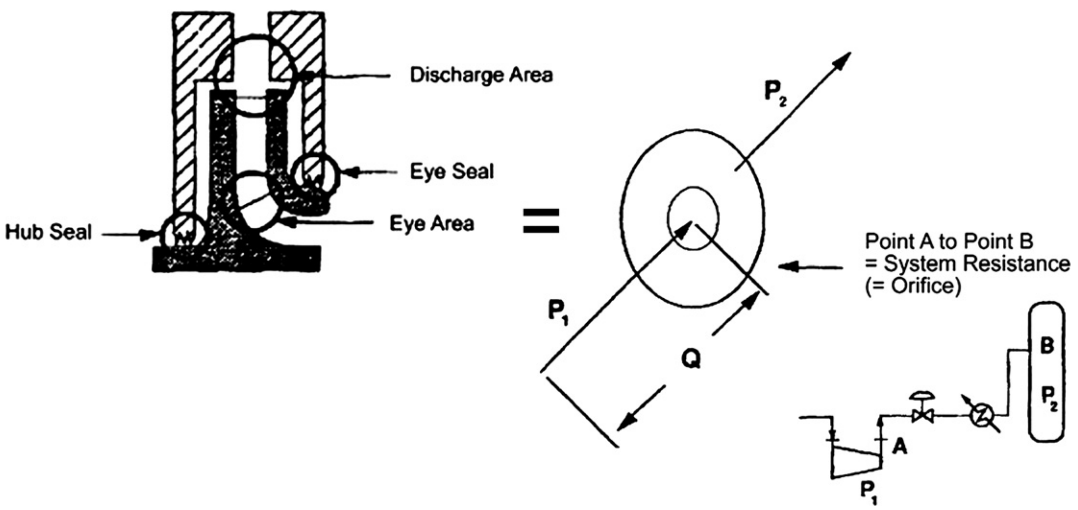


Fig 3.13.2 • Reduce it to an equivalent orifice

Table 3.13.1 Facts and relationships

■ A vector describes magnitude and direction →	
Tip speed $V = \dfrac{DN}{19,108}\left(\dfrac{(D)(N)}{229}\right)$ US units	
Flow related to velocity Q = AV ['Q' = (A)(V)(60)]	
■ Flow related to conditions (compressible flow)	$Q_F - Q_I \times \dfrac{P_I}{P_f} \times \dfrac{T_f}{T_I} \times \dfrac{Z_f}{Z_I}$
Where U = Tip velocity (m/sec or ft/sec)	f = Final condition
D = Diameter (mm or in^2)	I = Initial condition
N = Speed (rpm)	P = Pressure (kPa or PSIA)
Q = Flow rate (m^3/hr or ft^3/min)	T = Temperature (°K or °R) °K = °C + 273
A = Area (m^2 or ft^2)	°R = °F + 460
V = Velocity (m/sec or ft/sec)	Z = Compressibility

that the relationships presented are in British units. Metric units are not presented in this section, but can be easily derived referring to appropriate conversion tables.

Impeller with side plate removed

To begin our discussion, assume that we are operating at the impeller design point (as shown in Figure 3.13.3) and that we have removed the side plate of the impeller, and are examining the flow between any two vanes. Typical impellers are shown in Figure 3.13.3, and the schematic of an impeller suitable for our purposes, showing its upper half, with the side plate removed is shown in Figure 3.13.4.

In Figure 3.13.4 we can see that only two velocities need to be considered to properly describe the generation of head (energy). At the tips of the vanes there are two velocities that are present: the blade tip velocity, identified as U, and the velocity relative to the blade, identified as V_{REL}.

The blade tip velocity is the function of the diameter of the blade and the blade rotational speed. The velocity relative to the blade (V_{REL}) is a function of the area between the blades, the flow rate at that location and the angle of the blade at the discharge of the impeller. Summing these two velocities, the resultant or absolute velocity defines the magnitude and the direction of the gas as it exits the blade. For this discussion, we assume that the velocity relative to the blade exactly follows the blade angle; that is, the slip is equal to zero. This assumption can safely be used since it will not impact the final conclusion of our discussion.

Impeller discharge velocities

If we now resolve the absolute velocity noted in Figure 3.13.4 (R) into its x and y components, the x axis projection of the component is the tangential velocity of the gas at the impeller discharge (refer to Figure 3.13.5). Euler's energy equation states 'The energy created by any turbo machine is proportional to the product of the tip speed and the tangential velocity'.

Let us now assume that the head required by the process changes such that the flow V_{REL} through the impeller reduces. Referring to Figure 3.13.6 let us again examine the discharge velocity to see what happens at this reduced flow condition.

Assuming that the rotor speed is constant, it can be seen that the value of the tip speed does not change, since it is a function of impeller diameter and shaft speed.

However, the velocity relative to the blades (V_{REL}) will be reduced, as a result of a lower volume flow passing through a fixed area, resulting in a low velocity relative to the blade at the discharge. If we again sum the velocity vectors to obtain the absolute velocity R (refer to Figure 3.13.7), we can see that the angle of the gas exiting the blade is significantly reduced and the x projection of the tangential velocity will be greater than the previous value (refer to Figure 3.13.8).

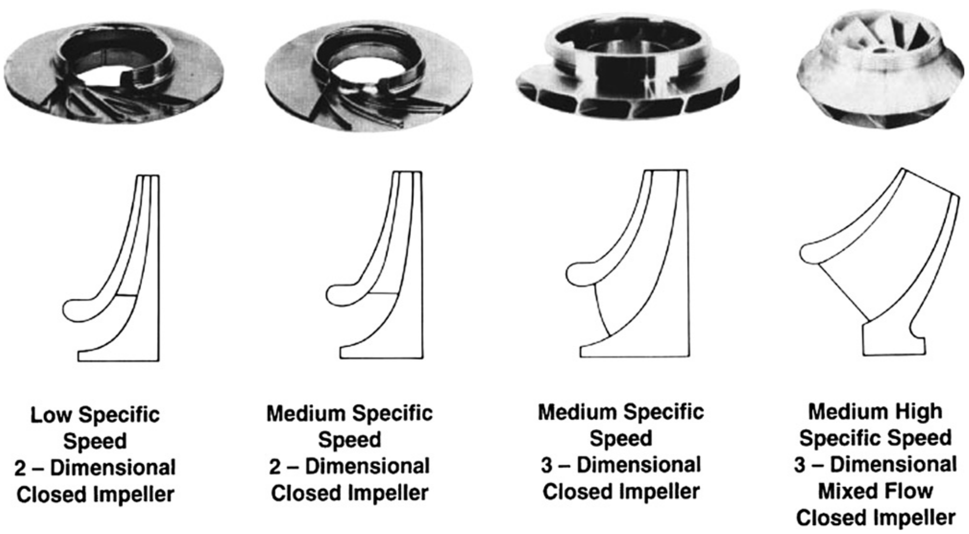


Fig 3.13.3 • Typical impellers (Courtesy of IMO Industries, Inc.)

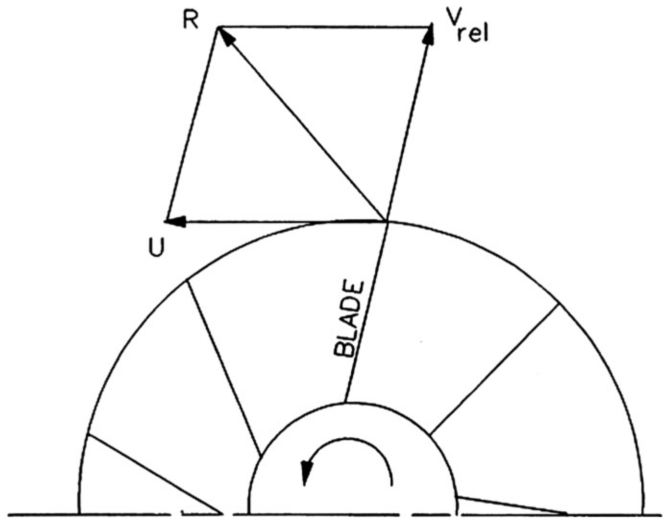


Fig 3.13.4 • Impeller with side plate removed

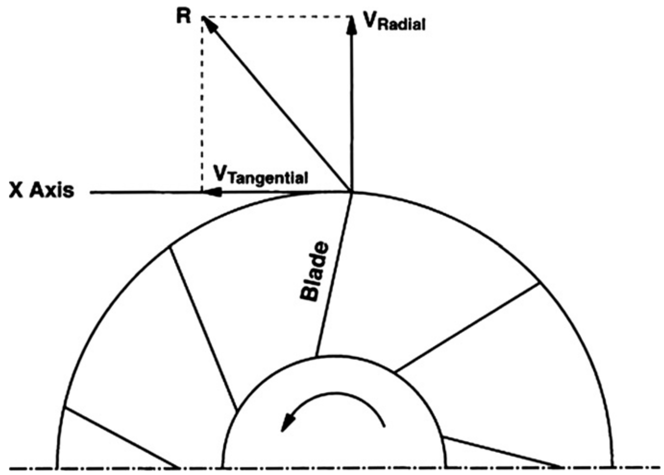


Fig 3.13.5 • Impeller with side plate removed

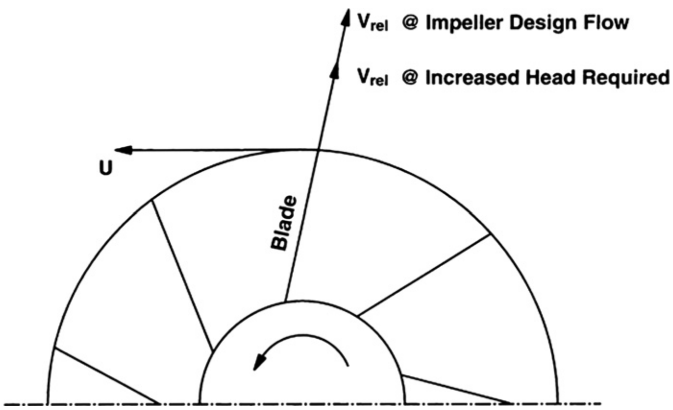


Fig 3.13.6 • Impeller with side plate removed

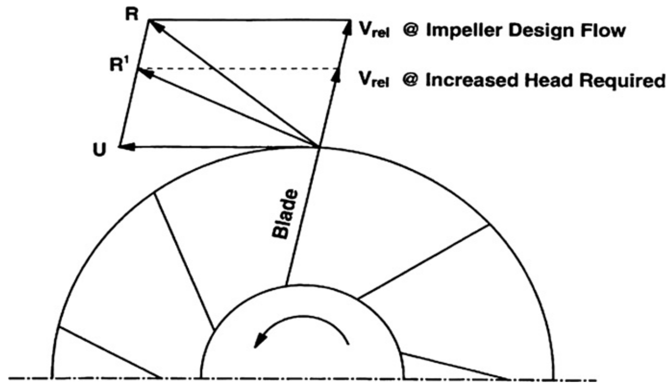


Fig 3.13.7 • Impeller with side plate removed

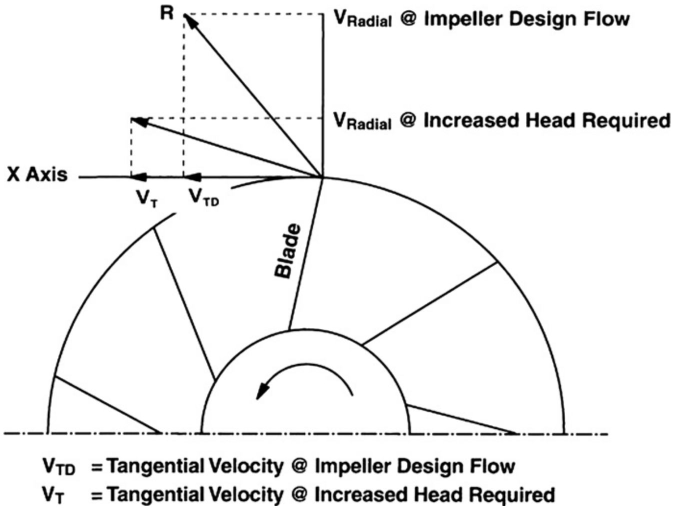


Fig 3.13.8 • Impeller with side plate removed

Since the head (energy) produced by the blade is proportional to the tip speed (unchanged) and the tangential velocity (increased), we can see that the reduction of flow through the blade has resulted in increased head or energy imparted to the fluid. Practically, this makes sense, since the slower the gas proceeds through the vane, the more time it has to pick up energy imparted by the blades, and as a result will increase the energy produced within the impeller. Therefore it can be seen for all dynamic blades and impellers which increase the energy of the fluid by the action of the vane on the fluid can increase fluid energy only at a lower flow rate, assuming the speed of the impeller and the inlet angle of the fluid to the blade remain unchanged.

Blading types

Backward lean

The previous discussion focused on the characteristic of a backward leaning vane. Most turbo machinery vanes are of this type, since they produce a greater head rise from impeller design point to the low flow operating point. The low flow limit of operation for centrifugal compressors is known as surge.

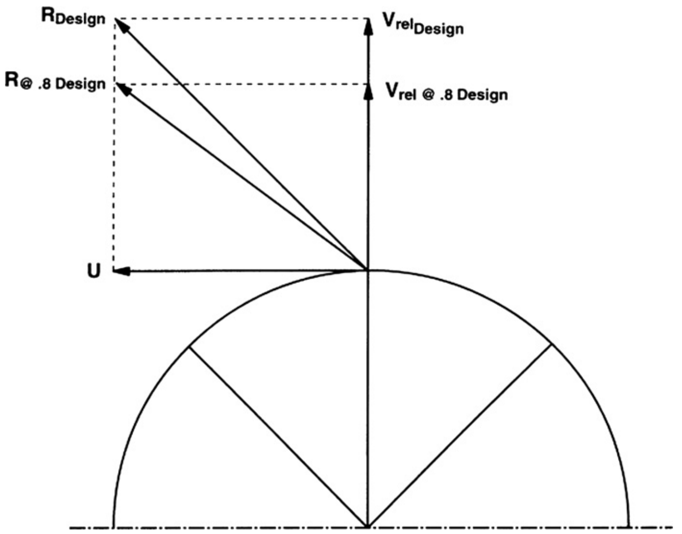


Fig 3.13.9 • Radial blading

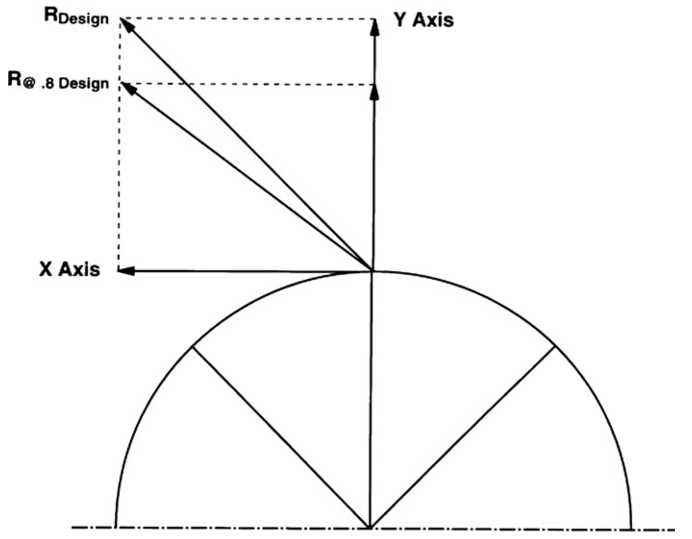


Fig 3.13.10 • Impeller with side plate removed

Head rise is defined as the head produced by the impeller at the low flow operating point divided by the head produced by the impeller at the impeller design point. Today, the industry prefers backward leaning impellers with an external or exit blade angle of approximately 40 to 50 degrees. This blade angle will produce head rises in the range of 5–15% depending on the gas density.

Radial

Radial vanes are used in some older design, open type, first stage impellers, and in some modern impellers that operate at a very low flow. Let us now examine the effect of a radial blade on the performance curve. If we were to design an impeller with radial blades let us examine again what would happen when we changed flows from a rated point to a lower flow. At the rated point the blade tip speed and velocity relative to the blade will be as shown. Refer to Figure 3.13.9.

Notice that the velocity relative to the blade is completely radial, assuming zero slip, and consequently the absolute velocity is the sum of the two vectors. Again we project the tangential velocity on the x axis projection from the absolute velocity and note its value as shown in Figure 3.13.10.

At a lower flow, tip speed will remain constant (assuming constant shaft speed) and the relative velocity will decrease as in the case of the backward leaning blade. However, note that since the relative velocity follows the radial blade path, the magnitude of the tangential velocity remains constant regardless of the value of relative velocity. This is shown in Figure 3.13.11.

Since the energy generated by the blade is the product of tip speed (unchanged) and a tangential velocity (unchanged) the design head (energy) produced in a radial impeller will remain essentially constant. Therefore, the curve shape will be significantly flatter and will possess much less of a head rise than that of a non radial vane. In reality though, the effects of friction will in fact produce a curve shape that will increase from high flows to low flows but the effects will produce much less of an energy increase.

This value is typically approximately 3% head rise or less. This is an important fact to remember, since the operating point of any dynamic machine will be the intersection of the head required and the machinery curve head produced. A characteristic curve with a low head rise will have greater sensitivity to process changes than a curve with higher head rise.

In summary, it should be noted that the previous discussion can be equally applied to pump impellers, since pumps also operate on a fluid (liquid). One very important thing to remember from this discussion, however, is that regardless of the type of liquid used in pumps, velocity relative to the blade will never change since the fluid is incompressible. In the case of a turbo-compressor, however, this will not be true, since the gas is compressible and the velocity relative to the blades at the discharge will change as a result of pressure and temperature of that gas at the exit. Therefore the statement that head (energy) produced by a compressor impeller will remain constant at a given speed is not totally true.

Having previously discussed the performance characteristics of a single compressor stage, we will now examine the effects

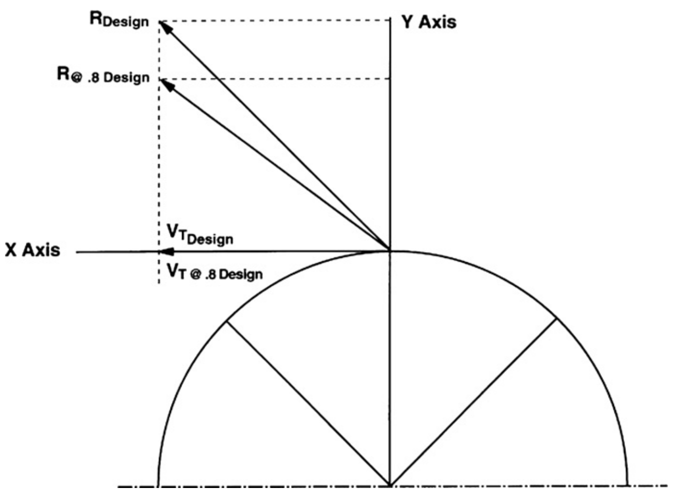


Fig 3.13.11 • Impeller with side plate removed

which multistage compressor configurations have on the overall compressor performance curve.

The stage curve is defined as the curve representing the performance of one stage which consists of the impeller, the stage seals, the diffuser, the cross over and return passage. Each compressor blade row or impeller stage has a specific performance curve that is plotted as actual volume flow vs. head (energy) and actual volume flow vs. efficiency.

The amount of impeller stage head is limited by both mechanical and aerodynamic factors. Mechanical factors include impeller stresses and blade natural frequencies. Aerodynamic factors are tip speed, optimum efficiency, Mach number and flow range considerations.

A compressor section is defined as the number of stages between turbo-compressor casing nozzles. A section can contain one or more stages. Its performance is defined by a section curve. The number of stages per section is limited by discharge temperature, process gas characteristics, casing and configuration, and rotor stiffness considerations.

We will examine stage and section performance for an ideal designed compressor case, a case with fouled impellers, and a case with varying molecular weight.

Any multistage compressor is designed on the basis that succeeding impellers will compress a lower volume flow resulting from the compressibility of the specific gas handled. Remembering that each application is designed for only one operating point, the compressor designer matches each successive impeller as closely as possible to achieve operation at the individual impeller design point, such that each individual impeller will operate at its best efficiency point. It is important to understand that in many applications, operating at the compressor section rated point does not mean operating at each individual compressor impeller best efficiency (design point). Many multistage compressors of older design were built using specific impeller designs. For any given application, the succeeding impeller may not be at its optimum efficiency point (design point) but may be significantly far from that operating point. A good rule of thumb in selecting a multistage compressor is to ensure that when operating at the compressor section rated point, each impeller operates at approximately $+/-$ 10% of its design point. Today (2010) some compressor manufacturers are using computer (CAD/CAM) design to manufacture each impeller individually, so that each operates at its design point, giving a minimum of mismatch between impellers. Therefore, it can be seen that the overall performance section curve is the composite of the operation of each individual impeller performance curve.

If the performance of one or more of the impeller stages in a compressor section were to deteriorate, for instance as a result of fouling or increased labyrinth seal clearances, the overall performance curve will be affected. The amount of this effect will depend on the performance deterioration of the individual impeller stages. We have shown a case where the first and second impellers of a three stage compressor section become fouled. The resulting section operating curve can deteriorate significantly in the terms of head (energy), flow range and efficiency.

Another case to consider is the change in gas molecular weight. Remembering that once an impeller is designed, its energy production at a given flow point is essentially fixed, we can see that pressure rise will change as gas composition changes. It must be remembered that each gas composition requires a different amount of head (energy) to increase its pressure level to a given amount. An impeller or blade is designed for only one gas composition and therefore only a specific amount of energy is designed in a blade or impeller for a given application at a specific flow rate. In a multistage compressor, the effect of gas composition change can have a significant effect on the overall compressor curve. If, for instance, a gas of higher molecular weight were to be handled, the pressure produced in the first stage of a multistage compressor section would increase. Since the gas is compressible, this would result in a reduced volume to the second stage. This impeller was initially designed for a higher volume but now will handle a lower volume. Also, a dynamic blade or impeller will produce higher energy at a reduced flow, therefore compounding the effect of the molecular weight increase with increased blade energy capability resulting in a further reduction in flow rate to the succeeding stage. This fact will result in a shifting of the section curve from the design point towards the surge point and could result in surge of a compressor with no significant change in system resistance.

The stage curve

Most of the dynamic compressors used today are multistage because the head required by most processes are in excess of the head produced by one stage of a dynamic compressor. Typical values for one stage of dynamic compression are:

- Centrifugal closed impeller 3,050 $\frac{\text{m-kg}_f}{\text{kg}_M}$ or 10,000 $\frac{\text{ft-lb}_f}{\text{lb}_M}$
- Centrifugal closed impeller 4,575–7,625 $\frac{\text{m-kg}_f}{\text{kg}_M}$ or 15,000–25,000 $\frac{\text{ft-lb}_f}{\text{lb}_M}$
- Axial blade row 915 $\frac{\text{m-kg}_f}{\text{kg}_M}$ or 3,000 $\frac{\text{ft-lb}_f}{\text{lb}_M}$

Figure 3.13.12 shows the components associated with one stage of a centrifugal closed impeller. For each impeller or blade stage, there exists a specific head vs. flow performance curve. In order to achieve maximum efficiency and flow range in a multistage dynamic compressor, each stage should be operating at its maximum efficiency point when the compressor is at its rated (guaranteed) point. This is not true for many older designs, where only a limited number of compressor impeller or blade designs were available to select from. The amount of head produced by one stage is limited by certain mechanical and aerodynamic factors. These factors are presented in Figure 3.13.13.

As an example, open impellers can produce as much as 250% greater head than a closed impeller because the impeller stresses at higher speeds are significantly reduced by omitting the impeller cover or side plate.

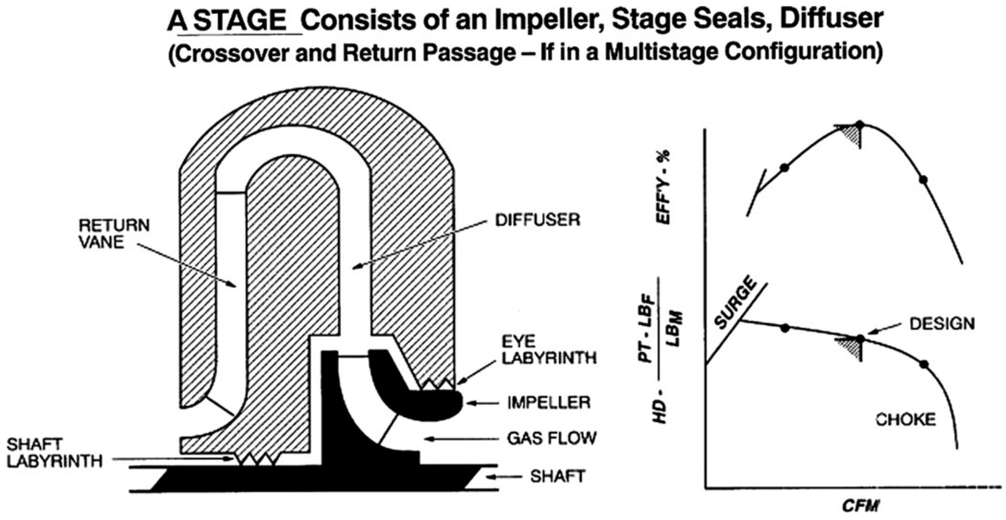


Fig 3.13.12 • A stage consists of an impeller, stage seals, diffuser (crossover and return passage – if in a multistage configuration)

Compressor impeller stage head is limited by:

Mechanical factors	Aerodynamic factors
■ Impeller stresses	■ Tip speed
■ Blade natural frequencies	■ Optimum efficiency
	■ Mach number
	■ Flow range considerations

Fig 3.13.13 • Factors limiting compressor impeller stage head

The overall curve

Frequently, there is much confusion over the terms 'compressor stage' and 'compressor section'. Process engineers and operators usually use the term 'stage' to describe what properly is termed a compressor section. This is probably because process flow diagrams only show a stage and not the individual impellers in the usual block diagram format. Figure 3.13.14 defines a compressor section, and shows a typical section performance curve.

In order to properly define compressor performance, a performance curve is required for each section. Each sectional performance curve is developed from the individual impeller stage curves. An important fact to remember concerning compressors is that the design of each succeeding stage is based on the predicted preceding stage performance. If the preceding stage performance is not as predicted, the next stage will be affected. The greater the number of stages, the greater this effect will be. It is commonly called mis-matching. Any change in gas density (pressure, molecular weight, temperature, compressibility) will affect the flow into the succeeding stage and thus affect the performance. The most effective way to minimize the effect of mis-matching is to require that each compressor stage operate as closely as possible to its design or best efficiency point.

The number of stages per section for dynamic compressors are also limited by performance and mechanical factors (refer to Figure 3.13.15).

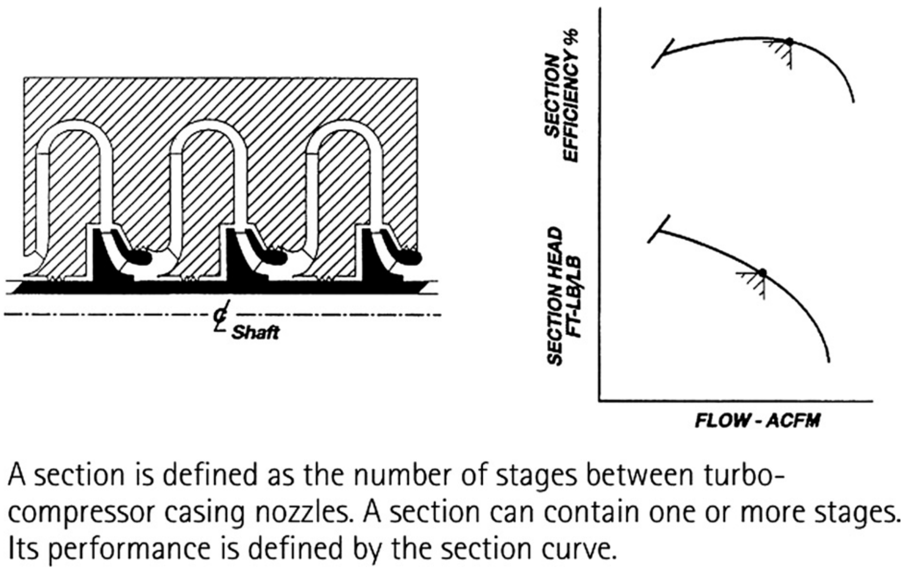


Fig 3.13.14 • Compressor section - definition/ performance curve

- Discharge temperature
- Process gas characteristics
- Casing configuration
- Rotor stiffness

Fig 3.13.15 • The number of stages per section – limiting factors

Some processes can cause accumulation of solid materials (fouling) within the compressor impellers or stationary passages. Such a phenomenon is usually temperature related, and can influence the number of intercooled sections for a given application. Also, a large number of stages on a single rotor can reduce the rotor stiffness and thus reduce the natural frequency (critical speed) of the compressor.

Determining section performance

As previously discussed, the performance curve for any section is derived from the individual stage curves of each impeller in that section. Figure 3.13.16 shows an example of a three stage nitrogen compressor section.

Note how the inlet volume flow to each successive section is reduced and how it depends on the impeller head produced by the proceeding stage. Figure 3.13.17 shows the effect of fouling the first and second stage impellers in the same compressor.

Note how this affects the operating points on the third stage: ● = original operating point, ○ = new operating point in fouled condition. The same effect would occur if the interstage labyrinth clearance increased from erosion or vibration on the first and second stages. Any reduction of head produced in a preceding stage will increase the volume flow in a succeeding stage and reduce the overall head produced by a compressor section.

Figure 3.13.18 shows the effect of changing the molecular weight in the same compressor. Note that for molecular weight changes of less than 20%, the head produced by a dynamic compressor stage does not change significantly. The greater the density of a gas, the greater the discharge pressure and the closer the surge point to the design point.

This is because once any dynamic compressor stage is designed, the head (energy) produced for a given flow and speed is fixed. The greater the density of the gas (proportional to molecular weight), the higher the pressure produced and the lower the volume flow. Flow is inversely proportional to pressure (Boyle's Law). Since surge is caused by low flow, a dynamic compressor handling a denser gas will surge sooner.

In the case of reduced gas density, the opposite effect will occur. The discharge pressure will be reduced and the surge point will move to the left, farther from the design point.

This concludes the chapter concerning individual stage and overall performance. One final comment: individual stage performance curves are vendor proprietary information and are not reproduced. The only opportunities that an end user has to review these curves are:

- Prior to an order, or
- During a design audit

Readers are encouraged to review this information during the bid phase of a contract, and to determine if all individual impeller operating points when operating at the guaranteed point (rated) are as close as possible to the individual impeller best efficiency points.

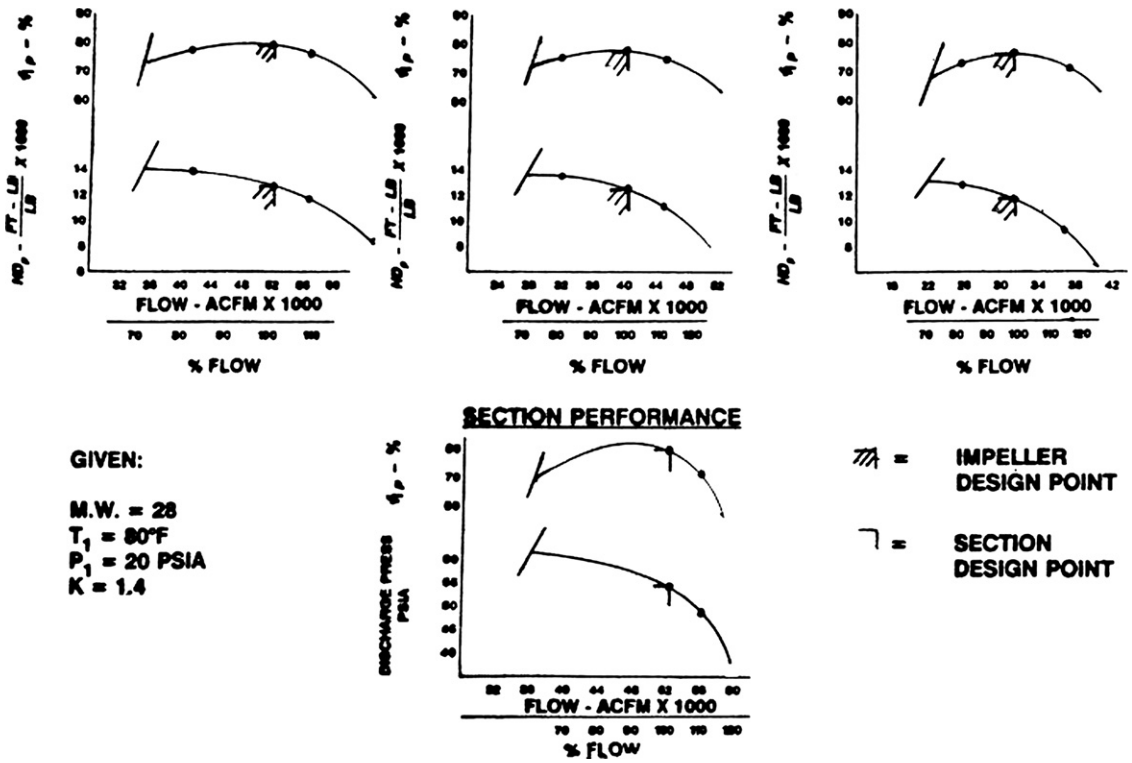


Fig 3.13.16 • Stage and section performance

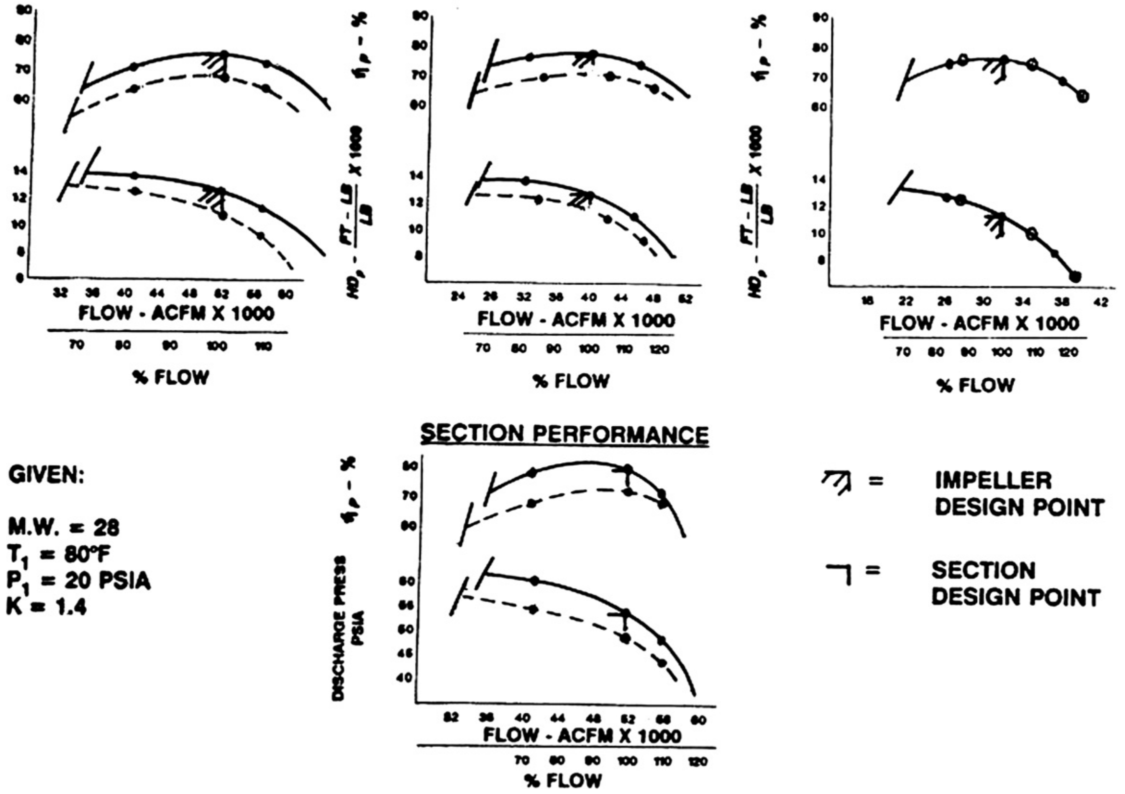


Fig 3.13.17 • Stage and section performance (1st and 2nd impellers fouled)

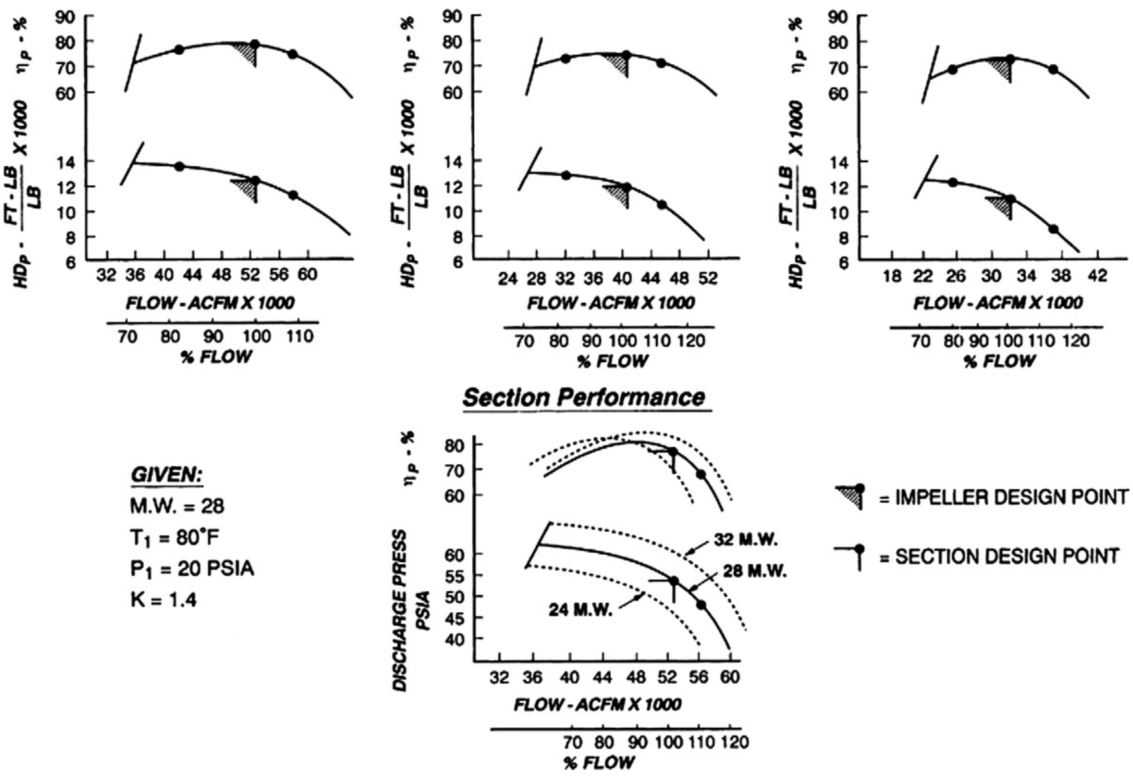


Fig 3.13.18 • Stage and section performance (M.W. varies from 24–32)

Best Practice 3.14

Positively define compressor fouling (debris accumulation on impellers and gas passages) by performance calculations and phase angle changes (heavy spot of unbalance).

Performance deterioration (efficiency and head reduction) can be caused by:

- Gas density changes (see Best Practice 3.13)
- Internal stage seal wear
- Assembly error
- Fouling

Fouling will have the greatest effect on compressor efficiency, reducing its value by 10% or more. Fouling can also be determined by monitoring phase angle change on the rotor since fouling on the impeller will break off irregularly causing unbalance and change of the heavy spot (phase angle change).

If fouling is the cause of the performance deterioration, it can be usually corrected without compressor disassembly, thus saving downtime and revenue losses.

Lessons Learned

Failure to monitor performance and phase angle of vibration has caused many compressor disassemblies when the fouling could have been removed by cleaning the compressor internals without disassembly.

Performance deterioration usually dictates that the compressor must be disassembled. This may in fact be the case, but, if rotor phase angle is not monitored, a fouled compressor may be disassembled unnecessarily, causing significant amounts of downtime (3 –7 days) and corresponding loss of revenue.

Benchmarks

The best practice of using on line and off line washing in upstream, downstream chemical and refinery compressor applications has been used since the 1980s to prevent compressor disassembly during a turnaround.

B.P. 3.14. Supporting Material

The material contained in this section will aid in preventing and correcting fouling. It is based on our experience since the 1980s, which has prevented many unnecessary compressor disassemblies.

Thus far, we have investigated the cause of the compressor curve, that is, the method in which energy is produced in any blade row or impeller. Energy is produced by the change in two velocities, namely the difference between inlet and discharge velocity relative to the blade and the change between inlet blade tip speed and discharge blade tip speed. In this section we will explore what will happen to the energy production of a blade row or impeller if that blade row or impeller were to change in area. Therefore, we will explain the mechanism of impeller fouling, with examples of the effect of fouling at the operating point. We will also discuss measures to prevent and correct fouling.

The mechanism of fouling

As mentioned earlier, one can reduce any blade row or impeller to a series of equivalent orifices. Flow is a function of area and velocity.

Whenever any blade row or impeller is designed, the designer sets inlet and discharge blade areas such that optimum velocities relative to the blade will be achieved at each location. By a series of tests and experience, designers have defined optimum relative velocity rather well. Therefore the resulting inlet and discharge areas will produce optimum velocities and corresponding optimum impeller efficiencies. If, however, the areas were to change, and flow passages were to become rough and non-continuous, an impeller performance change would result. Figure 3.14.1 shows the effect of fouling on a closed centrifugal impeller.

Impeller fouling is defined as the accumulation of debris in the impeller or blade passage, which reduces the flow area and roughens the surface finish. The distribution of the foulant on

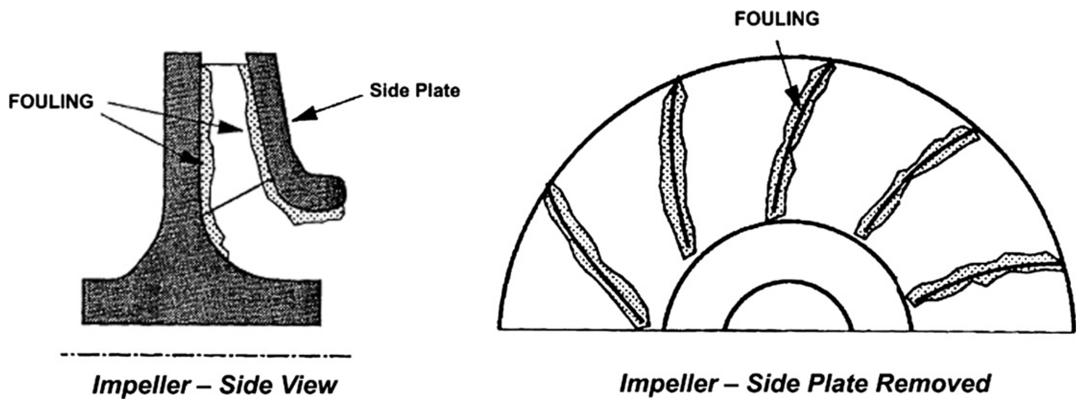


Fig 3.14.1 • Fouling – the effect on the operating point. Left: impeller – side view. Right: impeller – side plate removed

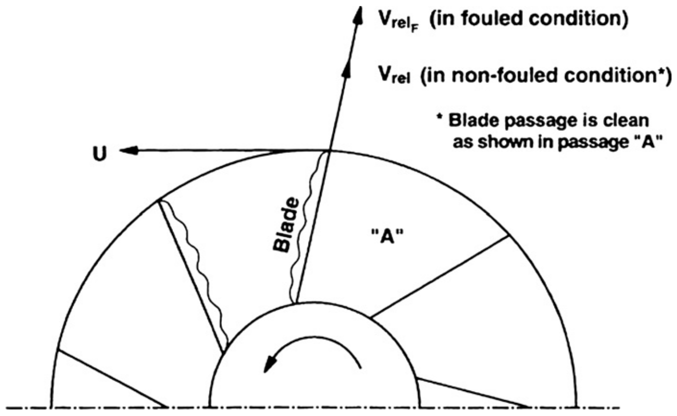


Fig 3.14.2 • Impeller with side plate removed

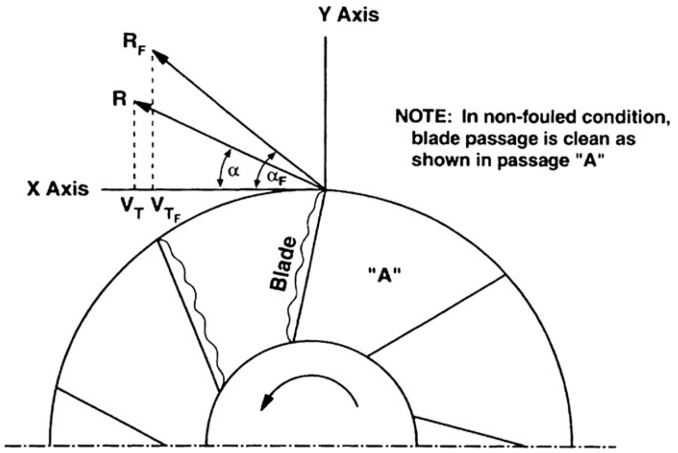


Fig 3.14.4 • Impeller with side plate removed

the impeller or blade row is non-uniform and usually changes with time. Flow patterns within the impeller or blade cause unequal distribution. In addition, the forces exerted on the foulant cause it to chip off with time as it becomes dry and brittle. This results in a change in rotor balance and a change in performance (head and efficiency).

The effect of fouling on the operating point

If we refer back to the previous example of a backward leaning centrifugal compressor impeller, the effect of fouling can be understood. Figure 3.14.2 shows the effect of fouling on the relative velocity.

Since the area of the flow passage is reduced when the impeller is fouled, V_{REL} will increase, the flow angle, $\propto$, will increase and therefore result in an absolute velocity (increased R) as shown in Figure 3.14.3.

The increase in $\propto$ and R due to fouling will reduce the tangential velocity of the gas as shown in Figure 3.14.4.

Since the head (energy) produced by the impeller is the product of the impeller tip speed “U”, which does not change in the fouled condition, and the tangential velocity which is reduced, the head produced will be reduced in the fouled condition. In addition, the non-uniform distribution of the foulant will reduce the efficiency of the impeller stage.

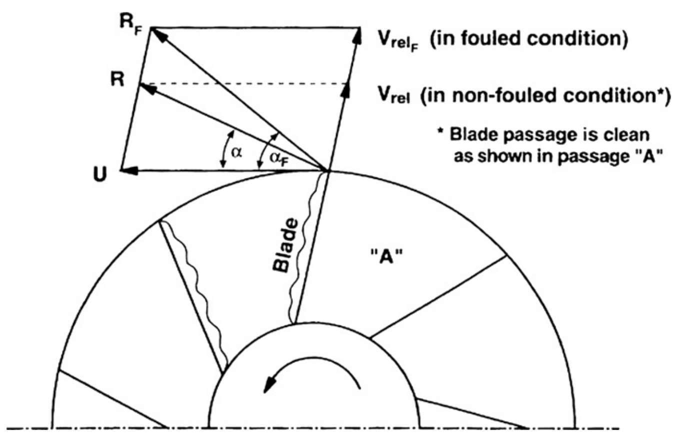


Fig 3.14.3 • Impeller with side plate removed

Figure 3.14.5 shows the effect of fouling on the impeller stage curve. Impeller fouling is the accumulation of material in the impeller passages that reduces flow area and roughens surface finish. It reduces impeller head capacity and efficiency.

Note that the surge margin actually increases slightly in the fouled condition. This is because the cause of surge is low gas velocity. Since the area of the flow passage is reduced, the gas velocity increases thus increasing the surge margin. The surge margin is defined as the flow at surge divided by the impeller design flow. However, the stage head produced by the impeller at any flow rate is reduced. Therefore, for the same process system head required, the impeller flow rate will be reduced thus forcing the operating point closer to the surge line.

The causes of fouling

The causes of fouling are described in Figure 3.14.6.

Fouling is difficult to predict. In air services, it is a function of the environment which can change with time. Fouling of an air compressor or the air compressor section of a gas turbine can suddenly occur. Causes could be:

1. A new industrial plant in the area
2. A new furnace stack in the area
3. Change in prevailing wind direction
4. Land erosion caused by de-forestation

In process applications, fouling can also occur unannounced. One example that comes to mind is a reformer recycle compressor upgrade in which I participated. A new, 30% larger, recycle compressor was installed, and immediately after start-up the entire compressor was fouled with ammonium chloride. The previous recycle compressor had not fouled in 10 years of operation! Figure 3.14.7 shows the removed rotor after only one week of operation.

Figure 3.14.8 shows a close-up of the fouled impeller looking into the discharge passage. What was the root cause? After an exhaustive troubleshooting period, it was discovered that we forgot to consider the suction side of the process system in our upgrade of the reformer unit. It seems that a 30% increase in suction line gas velocity and an undersized, poorly

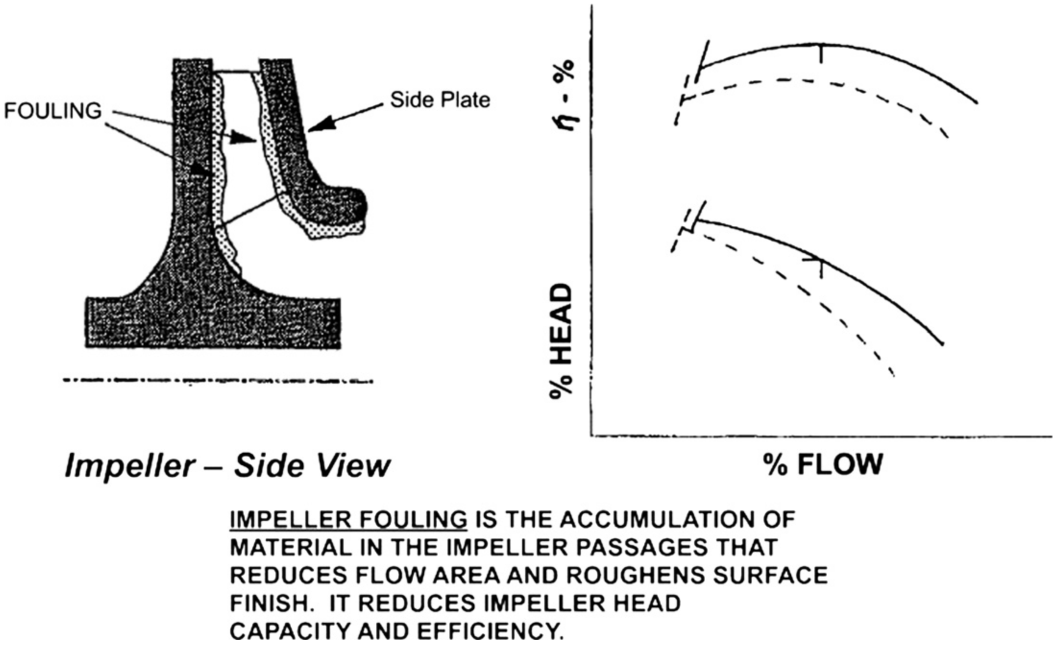


Fig 3.14.5 • Impeller fouling

- Airborne debris – air service
- Suction separator carry-over
- Process gas characteristics
- Temperature sensitive
- Pressure sensitive

Fig 3.14.6 • The causes of fouling

designed suction drum demister pad resulted in an entrainment velocity sufficient in power to carry over ammonium chloride into the compressor. Redesign of the demister pad solved the problem.

In other processes, a slight change in operating conditions can cause polymers to be formed that will adhere firmly to the impeller passages. Polymerization can occur in ethylene plant cracked gas compressors, wet gas compressors and linear low density polyethylene recycle compressors. In these services, the temperature of the process gas within the compressor must be kept below the critical temperature that will initiate the formation of polymers. This temperature varies with the process, but generally should be below approximately 100°C (212°F) to avoid problems. Process gas temperature change within any compressor is a function of compressor efficiency.

Therefore, compressor stage efficiency deterioration due to excessive stage seal clearances, impeller erosion, axial misalignment, etc. can initiate fouling in these processes. Most centrifugal, and all axial compressors, have more than one stage. Fouling can occur in all stages, only the first stage or in any combination. The overall effect of fouling on compressor performance will be the accumulative effect of deterioration in each stage. Figure 3.14.9 shows an example of fouling in a three stage centrifugal compressor.

Fig 3.14.7 • Removed rotor after a week of operation

Fig 3.14.8 • Close-up of the fouled impeller

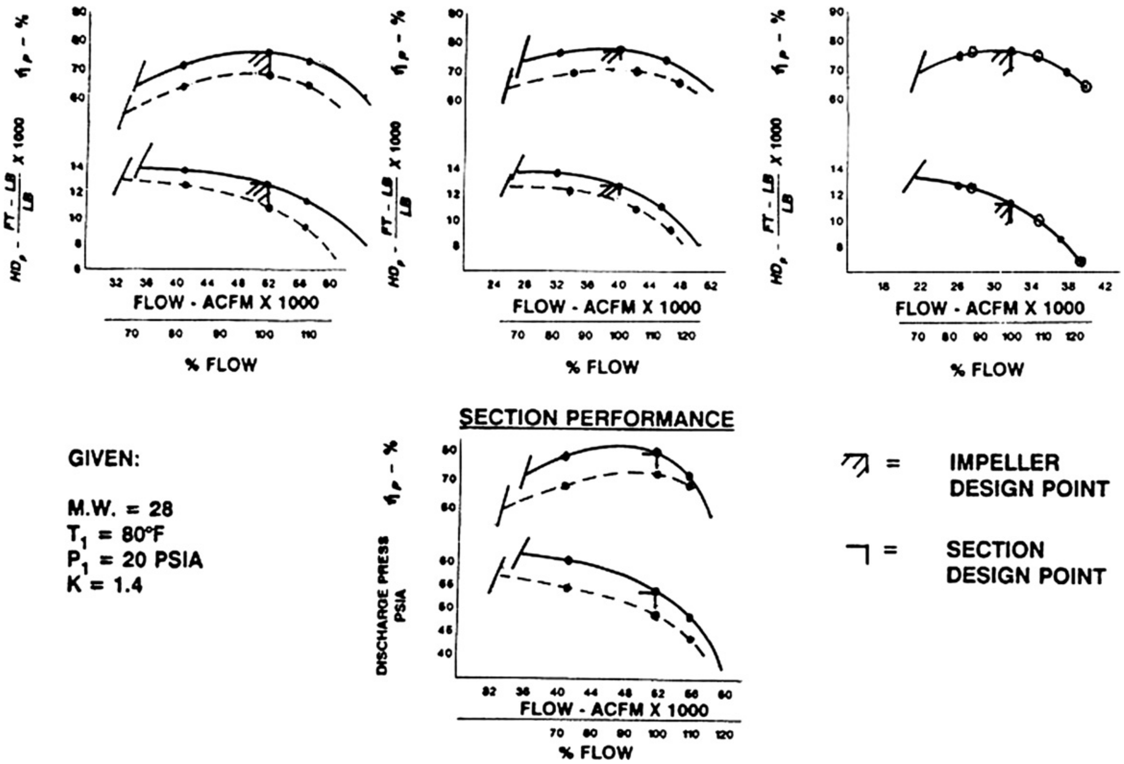


Fig 3.14.9 • Stage and section performance (1st and 2nd impellers fouled)

Therefore, periodic condition monitoring of compressor performance can provide advance warning to correct compressor internal deficiencies at the first opportunity.

Detecting fouling by condition monitoring

As previously discussed, the mechanism of fouling involves non-uniform accumulation and removal of debris from the impeller or blade row, which results in a performance design change in the mass accumulation of the foulant on the impeller. This action will change the rotor balance and the phase angle (the heavy spot) of the unbalance. The suggested condition monitoring practice to detect and confirm fouling is presented in Figure 3.14.10.

- Trend radial vibration and phase angle – a random change in both parameters indicates fouling
- Trend compressor head and efficiency. Plot results on compressor test curve – an increasing deficiency margin confirms fouling

Fig 3.14.10 • Condition monitoring guidelines to determine fouling

Preventing and correcting fouling

Thus far we have seen how fouling can occur and how it can cause surge. The detrimental effects of surge have been discussed in a previous section. How then can we prevent fouling if possible? And how can we correct fouling, if and when it occurs? Can we correct fouling on-line? These are important questions that must be answered. Figure 3.14.11 presents solutions for preventing and correcting fouling.

- Preventive measures
- Process control
- Proper inlet air filter selection and maintenance
- Condition monitoring (gas pressure, temperature and vibration)
- On-line solvent injection
- Impeller or blade coatings
- Corrective measures
- Slow roll or stationary wash
- On-line random washing

Fig 3.14.11 • Preventing and correcting fouling

Preventing fouling

Listed below are proven methods for preventing fouling.

1. Process control – Accurate control of process conditions can prevent fouling in applications where polymers can be formed. Control of temperature is usually the most important. The following applications can be affected by excessive process temperature:

- Ethylene cracked gas
- Linear low density polyethylene

- High density propylene
- Fluid catalytic cracker off gas (wet gas)
- Thermal catalytic cracker off gas (wet gas)
- Coker gas

The temperature below which fouling can be prevented varies with each process, compressor and application. Monitoring of process conditions is necessary to establish a threshold temperature in each case. In some cases, fouling cannot be prevented with the existing compressor. It may be necessary to modify the aerodynamic design and/or add additional cooling. Note that water injection is used quite regularly in ethylene unit cracked gas compressors to reduce gas temperature. However, an industry best practice is to keep the amount of injection water below 3% of the inlet mass flow.

2. Proper inlet filter selection and maintenance – In air compressor and gas turbine applications, filter selection is an important factor in preventing fouling. Site conditions must be accurately monitored prior to installation. It is advisable to contact filter vendors to survey site conditions. Filters must be able to prevent debris and excessive moisture from entering the air compressor. In platform applications and installation close to water, special designs must be used to prevent excessive chloride (salt) build-up. It is likely that multi-stage air filtration should be considered. Fouled air compressor sections of gas turbines can reduce output power by 3–5%.

Another important consideration is filter maintainability. For crucial (un-spared) services, filters with on-line cleaning capability should be considered. In recent years pulse type ("Huff and Puff") filters that use a reverse air pulse to clean filter cartridges when filter pressure drop becomes excessive have become popular.

Regardless of the type of air filtration used, or not used, remember environmental conditions can change as previously discussed in this module.

3. Condition monitoring of compressor parameters – Regardless of the service, condition monitoring of molecular weight, pressure and temperature will provide valuable information regarding process condition changes that can cause or increase fouling.

4. On-line solvent injection – Certain process applications have had success with on-line solvent injection. The objective of this measure is to *continuously* inject a small amount of solvent to reduce the friction coefficient of the blade and impeller surface and thus prevent the fouling from accumulating on the surface. The object of solvent injection is often misunderstood. The idea is to prevent foulant accumulation, not clean the impeller or blade on-line. Non-continuous solvent injection will allow the impeller or blade surface to dry and thus promote fouling. Even if a unit injection is continuous, selection and location of injection spray nozzles is a critical factor in success. Insertion of spray nozzle(s) in the inlet pipe of a compressor section is usually not as effective as injection directly into each stage of the compressor. Knowledge of solvent vapor pressure and internal compressor temperatures is necessary to determine if section or stage solvent injection is required.

Type of spray nozzle to be used can best be determined by discussions with process designers or other users having similar process applications.

There is no question about it, solvent injection is expensive! Most solvents used are naphtha based. Many users have required flow meters at each injection point to determine the proper amount of injection. Typically, the total amount of solvent injection in one compressor case should be 1–2% of mass flow, but the total amount should never exceed 3% of weight flow. Excessive solvent injection will erode leading edge blade tips at the blade to cover junction on closed impellers and can cause impeller failure if not detected in time.

5. Impeller or blade coatings – In the last few years, some users have investigated and have used impeller and blade row coatings to reduce friction in lieu of continuous solvent injection. Based on my experience, this has met with mixed results. There has been success with coating impellers with Teflon–based material in ethylene cracked gas service. It is recommended that the process designer and compressor vendor be contacted for additional information in this regard.

Corrective measures

If fouling cannot be immediately prevented, various off-line and on-line cleaning procedures are available. Off-line cleaning methods are definitely preferred to on-line cleaning methods, since they do not expose the compressor to vibration excursions and have proven to be more effective if a compressor is severely fouled. Failure to immediately remove all foulant from the rotor can cause catastrophic failure, since a large amount of unbalance can be instantaneously introduced into the rotor system. General guidelines for an off-line or "crank" washing procedure are presented in Figure 3.14.12.

1. Determine proper solvent by solubility test of foulant material.
2. Confirm solvent does not harm compressor internals (aluminum or Teflon-based labyrinth).
3. Fill and vent case to be sure it is completely filled with solvent. Note: use warm solvent, not to exceed 120°C (250°F) if possible. Be sure to confirm solvent is acceptable for use at this temperature.
4. Use turning gear or small belt drive assembly to turn case at 40–60 rpm during wash.
5. Measure conductivity of solvent solution prior to start of wash.
6. Change wash fluid frequently and continue wash until conductivity of wash liquid is equal to conductivity of initial value measured in step 5 above.

Fig 3.14.12 • Off-line cleaning guidelines

If an on-line wash is to be used, it should be employed periodically – usually at least once a week – and started when the rotor is in new (clean) condition. As stated previously, the injected amount of wash fluid should not exceed 3% of mass flow. Process and compressor designers should be consulted regarding the exact procedure to be used. It is advisable that a radial vibration trip system be employed if on-line washing procedure is to be implemented. This action will prevent catastrophic compressor failure in the event of an incomplete on-line wash.

We have included some examples of fouling along with preventive measures and cleaning methods in Table 3.14.1.

Table 3.14.1 Examples of fouling		
Service	**Preventive action**	**Cleaning method**
Air	Good inlet filtration	Water wash catalyst or nut shell
Cracked gas	Process gas temperature control/continuous injection	Solvent wash
FCC wet gas		
TCC wet gas		
Coker gas	Continuous injection	Solvent wash
Polyethylene	Control process conditions and possible impeller coating	Stationary impeller Heating (off-line)
Reformer recycle	Prevent carry-over	Steam/water wash

Best Practice 3.15

Restrict the use of low solidity diffusers (LSD) to positively eliminate gas disturbances which can result in reduced centrifugal compressor safety and reliability.

Low solidity diffusers are used to direct the gas out of the impeller in a radial direction and not in the log spiral, as vaneless diffusers do. The result is reduced friction in a LSD, producing increased head and efficiency, but less flow range when compared to a vaneless diffuser.

If the fixed angle of the LSD vanes do not match the impeller exit gas angle, gas disturbances can occur (stall cells) at frequencies less than approximately 40% of operating speed, which can excite the rotor to produce large external forces and shaft vibration.

The forces exerted on the rotor by LSD stall increase with the molecular weight and density of the gas, and can be large if the molecular weight of the gas is greater than 40.

Lessons Learned

The use of low solidity diffusers (LSD) has resulted, in some cases, in severe production reductions and compressor vibration trips, due to gas disturbances (stall) that result from the mismatching of the LSD vane angle and the impeller exit gas angle.

Benchmarks

This best practice has been used since the late 1990s, when I was involved with a LSD type propylene compressor (molecular weight = 44) that could not operate beyond 103% flow from the rated point without tripping the compressor in the field.

During the Factory Acceptance Test (FAT), which was performed on nitrogen gas, it was discovered that there was a sub-synchronous vibration frequency component greater than 25% of the overall vibration at a frequency of 40% of operating speed. The recommendation was to reject the compressor, in accordance with project and API 617 specifications, modify the LSD vane angles and retest. The project team decided to accept the compressor based on start-up schedule considerations.

The compressor had to operate at flows less than 103% of rated flow for 3 years until the problem was corrected. During this period, many compressor trips occurred that resulted in losses of revenue totaling millions of dollars.

B.P. 3.15. Supporting Material

Low solidity diffusers (LSD) have been used since the late 1980s to increase the head produced per stage, and potentially reduce the number of stages in a multistage centrifugal compressor, and so reduce the quoted capital cost.

The name ‘low solidity diffuser’ comes from the fact that the vanes do not extend through the entire length of the diffuser, but are located approximately in the middle of the diffuser passage for purposes of avoiding pulsating impeller/LSD vane forces on the impeller – which would otherwise result in impeller cover breakage.

Surge facts

Surge can be a very dangerous phenomenon if it is allowed to continue. The damage caused by surge in any type of dynamic compressor can render it inoperable for periods of time in excess of two (2) months. Considering that most dynamic compressors

- Surge is a high speed phenomenon. Flow reversals can occur in less than 150 milliseconds
- Reversal rate is 30 to 120 cycles/sec
- Pressure rapidly fluctuates
- Noise generated
- Temperature increase (can be rapid)
- Mechanical damage can occur
- Unit may trip
- Intensity varies with the application

Fig 3.15.1 • Surge facts

are not spared, this can cause a significant loss of revenue. Typical daily product revenues for a world class process unit can exceed US$1,000,000 per day! Figure 3.15.1 shows some of the most common effects of surge.

Of all the effects listed, by far the most damaging is the rapid temperature increase, since this can cause internal rubs of the compressor at operating speed, resulting in impeller breakage, diaphragm breakage, extreme labyrinth seal wear and possible case breakage.

Figure 3.15.2 shows the results of an actual surge test performed on a 1,865 kW (2500 HP) solar compressor. It shows the results obtained by a strip chart recorder during a surge cycle. Δp_o is the flow change and Δp_c is the pressure ratio change. Notice how, prior to the surge event, the pressure ratio required (Δp_c) increases while the compressor flow rate (ΔP_o) is decreasing. Observe that once a surge cycle is detected, the output of the controller works to open the anti-surge valve immediately. This fact is important to remember when trying to justify a field surge test to plant operations.

A compressor will never be damaged by a surge test if it is conducted properly. As shown in Figure 3.15.2, strip chart recorders are used and the surge system runs automatically, but the surge control line has been moved to the left of the actual surge location. The surge control line defines the set point of the surge controller and hence the point at which the surge control valve will open.

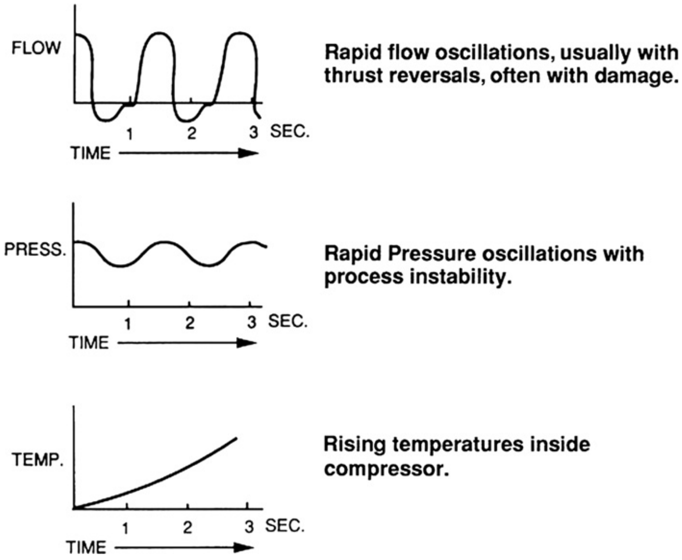


Fig 3.15.3 • The surge phenomena

Figure 3.15.3 presents the characteristics of flow, pressure and temperature during surge. The degree of parameter change is directly proportional to the density of the gas: the higher the gas density, the greater the effect of the surge.

The limits of the curve

The limits of any type of dynamic compressor curve are the low flow limit (surge) and the high flow limit (choke flow). Surge, as we shall see, is caused by low flow turbulence while choke flow is caused by high velocity friction. Figure 3.15.4 shows a typical compressor curve and the side view of a compressor stage (top half view only). It can be seen that the limits of any dynamic compressor curve are a consequence of gas velocity in the compressor stage.

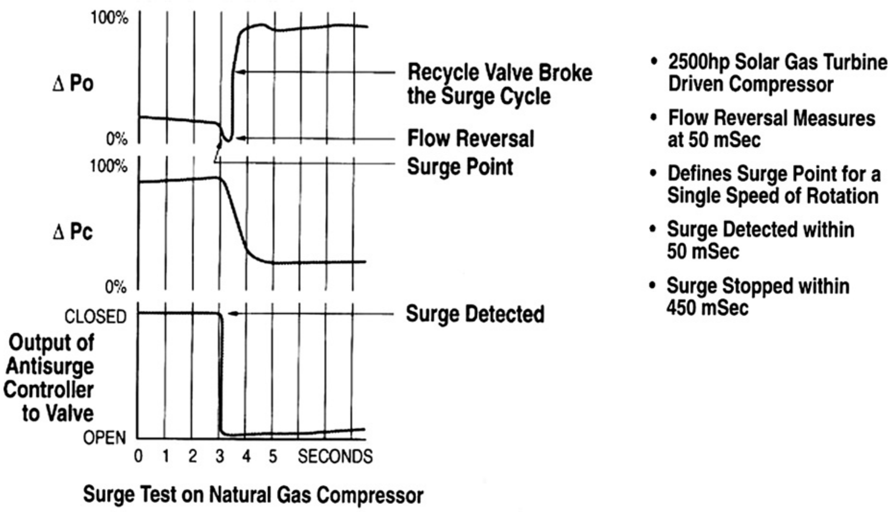


Fig 3.15.2 • A surge test (Courtesy of Compressor Controls Corp.)

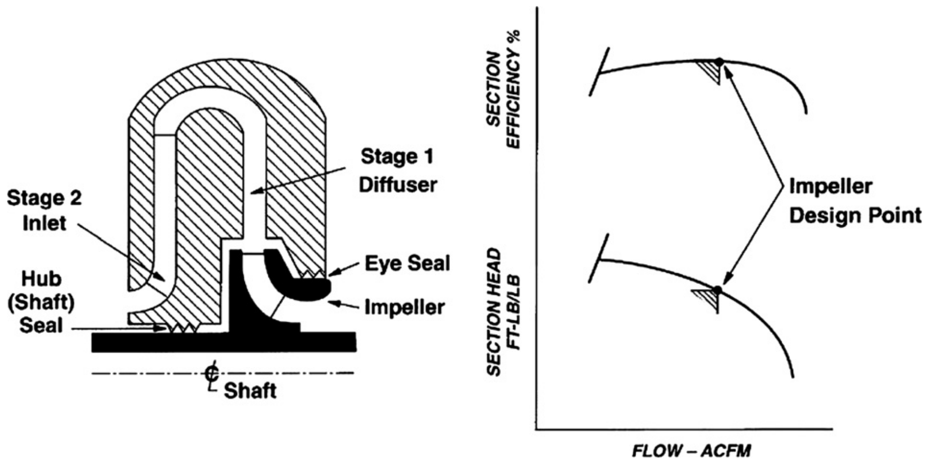


Fig 3.15.4 • The compressor stage and characteristic curve

What causes surge

Surge is a process system phenomenon, which results from flow separation, caused by low gas velocity, anywhere in a compressor stage (inlet guide vane, impeller suction, impeller mid section, impeller discharge or diffuser). Centrifugal pumps experience the same flow separation phenomenon at low flows, which sometimes causes the liquid to vaporize, thus resulting in recirculation, which can cause cavitation.

As the process system requires more head in any type of dynamic compressor, the flow is reduced to a point that causes flow separation. This event is commonly known as stall. An example of diffuser stall is shown in Figure 3.15.5. This diagram depicts a view of a simple impeller with the side plate removed and a vaneless diffuser. As the gas leaves the impeller via vector 'R', the resultant of the gas velocity through the impeller and the impeller tip speed, it passes through the diffuser in a path that approximates a logarithmic spiral. Since the relative gas velocity is the 'y' component of 'R', and will decrease with decreasing flow rate, the gas angle ∝ off the blades will decrease. As a result, the path of the gas off the blade will become longer with decreasing flow rate. At surge flow, the velocity off the wheel will be so low that the path of the gas will not leave the diffuser. This will cause a reduction in the head produced by the impeller. At this moment, the compressor is said to be in a state of flow separation, flow instability, or stall.

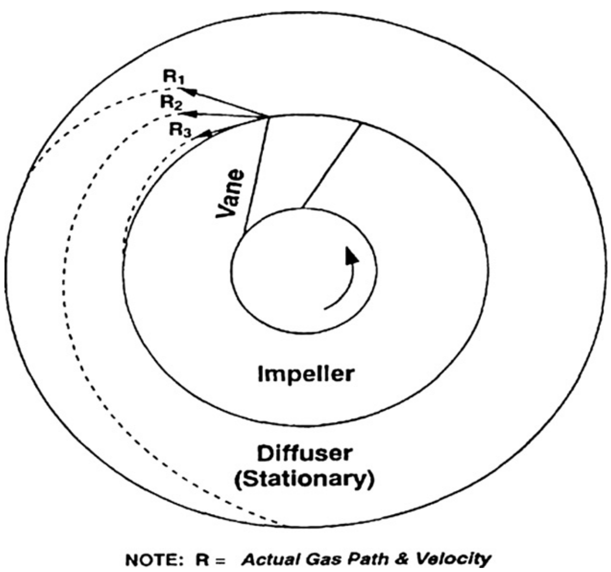


Fig 3.15.5 • Diffuser stall

Stall can be initiated by flow separation at any point within a compressor stage (inlet, mid-section, discharge, etc.). Regardless of the location of the flow separation within the compressor stage, a reduction in head produced by the impeller will occur. The dynamic compressor performance curve actually decreases to the left of the surge line, since the flow separation (stall) increases losses in the stage and reduces head produced. Please refer to Figure 3.15.6.

Draw a decreasing head curve to the left of the surge line. On this curve, place an operating point. This point would be the initial point in the surge cycle. Figure 3.15.7 shows a close-up of the process system that is given in Figure 3.15.6.

Once flow separation occurs, the head produced by any dynamic compressor decreases and the process gas present from the compressor discharge flange to the check valve flows backwards through the compressor. This backflow causes the volume shown in Figure 3.15.7 to be evacuated, resulting in a low discharge pressure. Since the head (energy) required by the process system is a function of discharge pressure, the head required by the process will decrease, allowing the compressor to operate in the high flow region of the performance curve. The reversal of dynamic flow caused by flow separation (stall) in the compressor stage and the recovery of flow resulting from reduced discharge pressure is defined as a surge cycle. This surge cycle will continue until either the head produced by the compressor is increased or the head required by the process is reduced. The quickest way to eliminate surge is to rapidly reduce the discharge pressure by opening a blow-off or recycle valve in the discharge process system.

Please refer back to the 'Surge Facts' section of this chapter to review the damaging effects of surge.

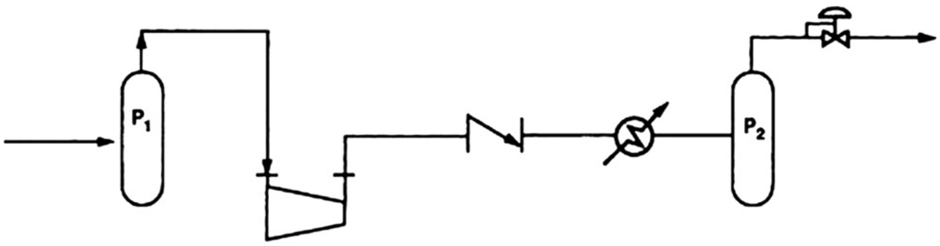
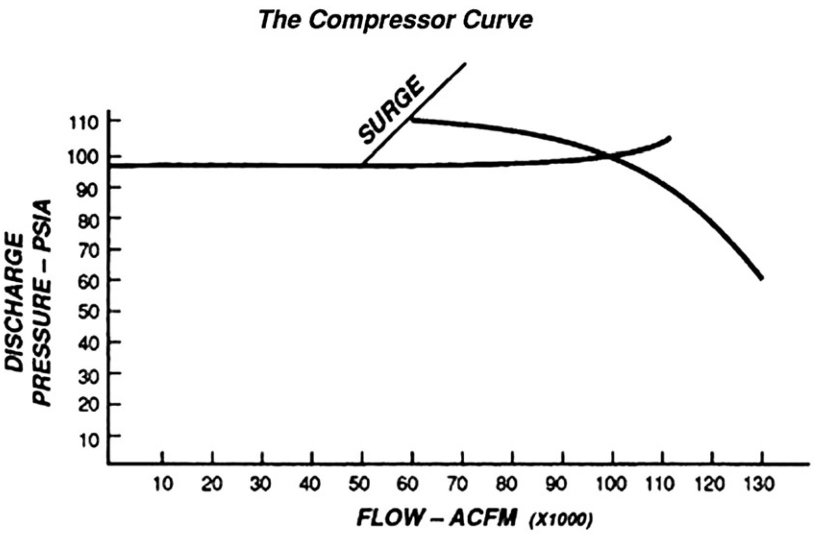
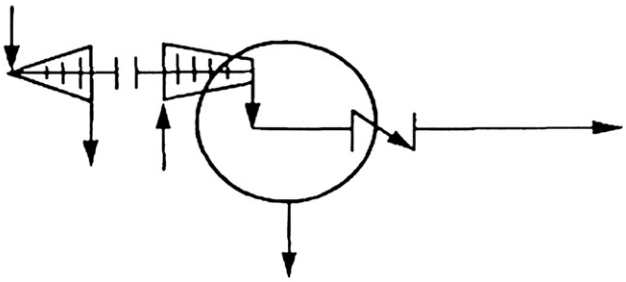


Fig 3.15.6 • Surge – the effect on the system



60,000 ft^3/min = 1ft^3/millisecond



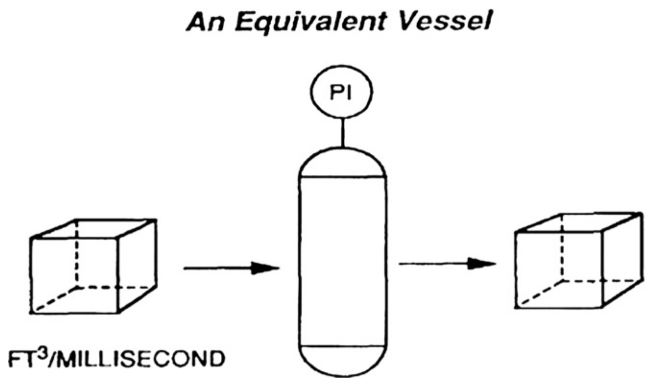


Fig 3.15.7 • Flow in and out of a vessel

Best Practice 3.16

Use integral geared centrifugal compressors only for spared compressor applications.

We do not recommend that un-spared integral geared compressors are used for un-spared process duty because:

- Integral gear centrifugal compressors were originally designed for plant and instrument air services, which would use spared compressors.
- They are maintenance intensive since they use multiple bearings, seals, gear meshes and operate at high speeds (above 50,000 RPM for the last stage).
- Because the gas is cooled after each impeller stage, their performance characteristics and reliability are dependent on intercooler condition.
- Since they are typically supplied with only an overall surge protection system and not individual impeller stage protection, they are prone to surging if intercoolers do not attain design heat removal requirements.

Lessons Learned

Un-spared integral geared compressors have lower reliability than between bearing centrifugal compressors (95%) and greater MTTR due to their many components.

I have been involved with many integral geared compressor failures since the 1990s and have experienced the additional maintenance time required to inspect and install bearings, seals and confirm acceptable gear mesh contact. Some clients have instituted a best practice not to use un-spared integral geared compressors for process applications including process air applications.

Benchmarks

This best practice has been used since the 1990s and has successfully convinced project teams not to use integral geared compressors for un-spared process applications. Between bearing compressors were justified on the basis of optimum reliability (99.7%+ compared to 95%) and less mean time to repair (MTTR).

B.P. 3.16. Supporting Material

A multistage, integral gear compressor casing is shown in Figure 3.16.1.

Typical applications are plant and instrument air and inert gas compressors. The casing consists of a cast or fabricated gear case and bolted on individual stage cast casings. Some designs allow rotor removal without having to disconnect the process piping.

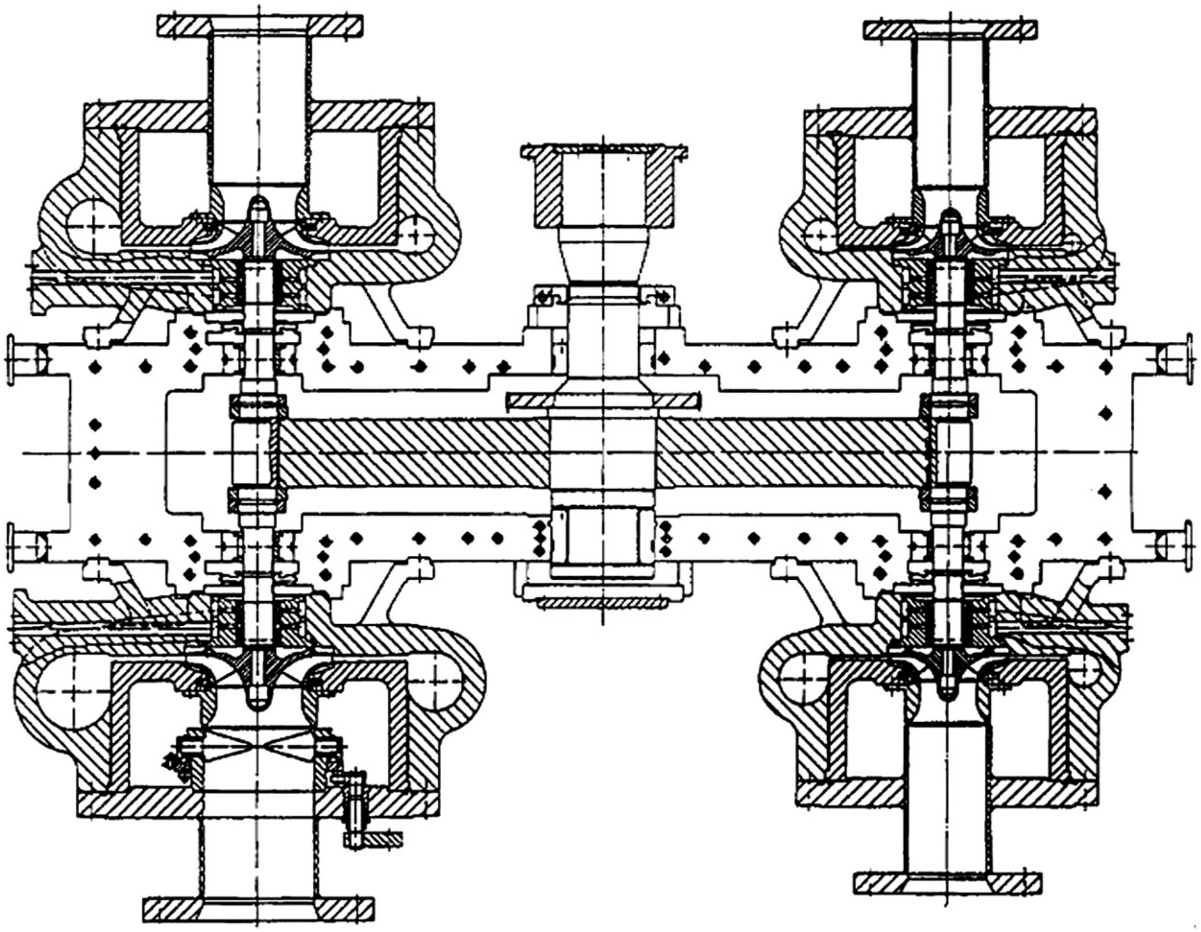


Fig 3.16.1 • 4 stage internal gear compressor (Courtesy of Demag Corp.)

The advantages of an integral gear compressor are the reduced number of stages required and higher efficiency. This is because all stages are intercooled, and the integral speed increasing gear allows all stages to operate in a higher specific speed.

Multistage, integral gear compressor manufacturers have begun to offer these units for hydrocarbon service. Readers are advised to exercise caution and confirm satisfactory field operating experience on similar gases before purchase. The requirements for gas or oil seals, varying gas composition and flow rates and the possibility of fouling can significantly reduce the reliability of this type of compressor opposed to air or inert gas applications.

Best Practice 3.17

Trend individual impeller performance and intercooler temperature rise for all integral geared compressors to ensure maximum operating range and reliability.

It is necessary to have pressure and temperature transmitters at the inlet and outlet of each impeller stage to trend the performance (operating point) of each impeller head and efficiency to determine the need for maintenance before internal conditions (excessive clearances and/or fouling) cause impeller damage.

Most integral geared compressors have a limited flow range for each impeller. Many types use diffuser vanes for increased stage head, which further reduce the flow range.

Operation within the allowable flow range for each impeller (above the surge margin and below the choke flow margin) will depend on the performance of the intercoolers as well as the impeller stages. Ensure that the following items are present in the design and predictive maintenance program:

- Cooler duty is oversized by a minimum of 10%
- The maximum cooling media conditions (water or air) are defined on the intercooler data sheet
- Intercooler gas side differential temperature is trended
- The cooling media operating conditions are within the ranges defined on the intercooler data sheet

Lessons Learned

Many integral geared compressor failures are the result of individual impeller and/or intercooler performance deterioration. Failure to monitor these items will result in low compressor safety and reliability.

Benchmarks

This best practice has been used since the late 1990s when many apparent integral geared compressor failures were encountered, only to find that the root cause was usually fouled intercoolers. This best practice has resulted in above average reliabilities for un-spared integral geared compressors (above 97%).

B.P. 3.17. Supporting Material

Please refer to the supporting material for B.P. 3.16.

Best Practice 3.18

Require a shaft stiffness ratio of 10 or less to ensure a stable rotor free of sub-synchronous vibration frequencies and second critical speed issues.

During the pre-bid phase of the project (before vendor priced proposal), obtain the bearing span dimensions and divide by the shaft diameter under the impellers and take the following action:

- If the ratio is less than 10, confirm that the first and second critical speeds meet the latest edition of API-617 prior to acceptance.
- If the ratio is greater than 10, require vendor experience information for the same pressure ratios and performance conditions, as well as a minimum of 2 installed references that have operated in the field for 2 years or more.

Lessons Learned

I have been involved with many RCFAs of sub-synchronous vibration and critical speed issues where the shaft stiffness ratio exceeded 10. Once the compressor is designed, it is very difficult to correct the issue, since changing the shaft diameter requires that impellers be re-designed and the bearing span cannot be changed.

Benchmarks

This best practice has been used since 1972, when the concept was introduced to me by an engineer that the writer was working with in a joint venture project. Implementing this best practice since that time has resulted in compressor reliabilities that have exceeded 99.7%.

B.P. 3.18. Supporting Material

The term ‘critical speed’ is often misunderstood. In nature, all things exhibit a natural frequency. This is defined as that frequency at which a body will vibrate if excited by an external force. The natural frequency of any body is a function of its stiffness and mass. As mentioned, for a body to vibrate, it must be excited. A classical example of natural frequency excitation is the famous bridge ‘Galloping Gerty’ in the state of Washington. That bridge vibrated to destruction when its natural frequency was excited by prevailing winds.

In the case of turbo-compressor rotors, their natural frequency must be excited by some external force to produce a response that will result in increased amplitude of vibration. One excitation force that could produce this result is the speed of the rotor itself. Thus the term ‘critical speeds’. The term ‘critical speed’ defines the operating speed at which a natural frequency of a rotor system will be excited. All rotor systems have both lateral (horizontal and vertical) and torsional (twist about the central shaft axis) natural frequencies. Only lateral critical speeds will be discussed in this section.

In the early days of rotor design, it was thought that the rotor system consisted primarily of the rotor supported by the bearings. This led to the assumption that only the stiffness of the rotor supported by rigid bearings needed to be considered in the analysis of the natural frequency. Countless machinery problems have proven this assumption to be false over the years. The concept of the ‘rotor system’ must be thoroughly understood. The rotor system consists of the rotor itself, the characteristics of the oil film that support the rotor, the bearing, the bearing housing, the compressor case that supports the bearing, compressor support (base plate), and the foundation. The stiffness and damping characteristics of all of these components together result in the total rotor system that produces the rotor response to excitation forces.

We will examine a typical rotor response case in this section and note the various assumptions, the procedure modeling, the placement of unbalance, the response calculation output and discuss the correlation of these calculations to actual test results.

Critical speeds

The natural frequency of any object is defined by the relationship:

$$F_{NATURAL} = \sqrt{\frac{K}{M}}$$

Where: K = Stiffness

M = Mass

When excited by an external force, any object will vibrate at its natural frequency. If the frequency of the exciting force is equal to the natural frequency of the object, and no damping is present, the object can vibrate to destruction. Therefore, if the frequency of an exciting force equals the natural frequency of an object, the exciting force is operating at the ‘critical frequency’.

Rotor speed is one of the most common external forces in turbo-machinery. When the rotor operates at any rotor system natural frequency, it is said that the rotor is operating at its critical speed. The critical speed of a rotor is commonly designated as NC and the corresponding natural frequencies or critical speeds are: NC_1, NC_2, NC_3, etc.

Every turbo-compressor that is designed must have critical speeds of the rotor system determined prior to manufacture. In this section, we will follow the procedure for the determination of the necessary parameters to define a rotor system’s critical speed. The procedure is commonly known as determination of rotor response. Figure 3.18.1 is a representation of a critical speed map for a rotor system.

It should be understood that all stiffness values are ‘calculated’ and will vary under actual conditions. As an exercise, determine NC_1, NC_2 and NC_3 for the horizontal and vertical directions for each bearing in Figure 3.18.1 (assume bearing 1 and 2 stiffness are the same)

Critical speed	Horizontal (X)	Vertical (Y)
NC_1	3,300 rpm	3,000 rpm
NC_2	9,700 rpm	8,000 rpm
NC_3	16,000 rpm	15,000 rpm

Based on a separation margin of ± 20% from a critical speed, what would be the maximum allowable speed range between NC_1 and NC_2 in Figure 3.18.1?

- Maximum speed 6,600 rpm
- Minimum speed 4,000 rpm

Remember, changing of any value of support stiffness will change the critical speed. Support stiffness, in lbs/inch, is plotted on the x axis. The primary components of support stiffness in order of decreasing increasing influence are:

- Oil support stiffness
- Bearing pad or shell
- Bearing housing
- Bearing bracket
- Casing support foot
- Baseplate
- Foundation

Note that this analysis of the critical speed does not include oil film damping. It is common practice to first determine the ‘undamped critical speeds’ to allow for necessary modifications to the rotor or support system. This is because the effects of stiffness on the location of critical speed are significantly greater than damping. Figure 3.18.1 shows four (4) distinct critical speeds. Operation within ± 20 of actual critical speeds is to be avoided. Also plotted are the horizontal (x) and vertical (y) bearing stiffness for each bearing. Note that these values vary with speed and are the result of changes in the oil stiffness. Therefore, a change in any of the support stiffness components noted above can change the rotor critical speed. Experience has shown that critical speed values seldom change from ± 5% of their original installed values.

If a turbo-compressor with oil seals experiences a significant change in critical speeds, it is usually an indication of seal lock-up. That is, the seal does not have the required degrees of freedom and supports the shaft acting like a bearing. Since the seal span is less than the bearing span,

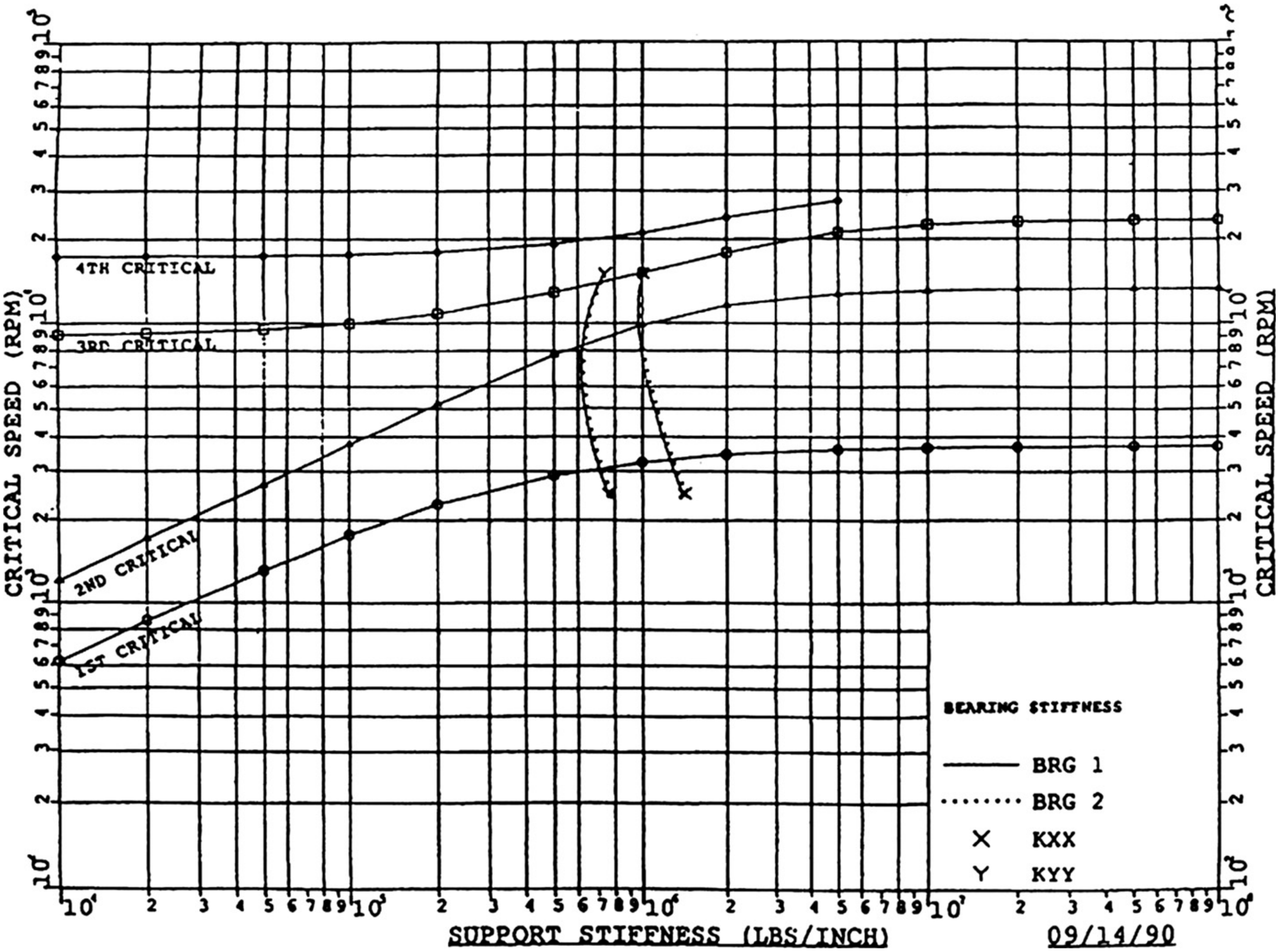


Fig 3.18.1 • Compressor rotor critical speed map — no damping (Courtesy of Elliott Company)

the rotor stiffness 'K' increases and the critical speeds will increase in this case.

The rotor system (input)

Figure 3.18.2 shows a typical turbo-compressor rotor before modeling for critical speed or rotor response analysis. Since the natural frequency or critical speed is a function of shaft stiffness and mass, Table 3.18.1 presents the rotor in Figure 3.18.2 modeled for input to the computer rotor response program.

Table 3.18.1 is an example of a modeled rotor and only includes the rotor stiffness (K) and mass (M).

In order to accurately calculate the rotor critical speeds, the entire rotor system stiffness, masses and damping must be considered. Table 3.18.1 models the oil film stiffness and damping of the journal bearings at different shaft speeds.

Note that it is essential that the type of oil to be used in the field (viscosity characteristics) must be known. End users are cautioned to confirm with the OEM before changing the oil type, as this will affect the rotor response. In addition to

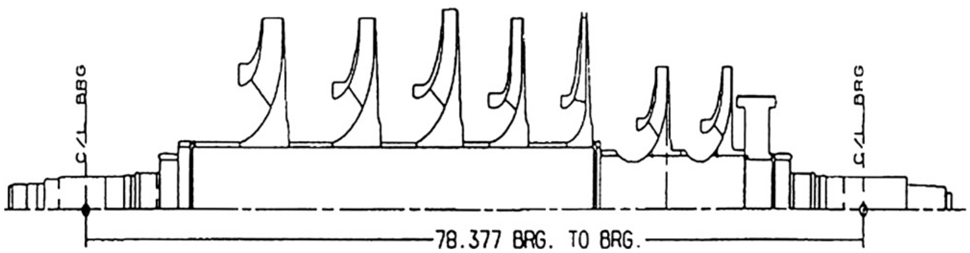


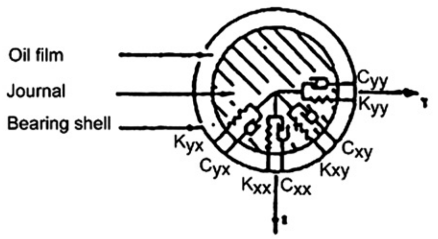
Fig 3.18.2 • Rotor response modeling — rotor (Courtesy of Elliott Co.)

Table 3.18.1 Typical compressor oil film bearing parameters (Courtesy of Elliott Co.)

4 × 1.6"tilt 20.5" TB 3.0" shaft end 7.5—6.5" shaft Bendix coupling

Static bearing load (lbs)	897	diameter (inches)	4.00
Bearing station	12	length (inches)	1.60
Bearing location	thrust	diam assembly clearance (inches)	5.7487E—03
Bearing type	tilt pad	diam machined clearance (inches)	8.7500E—03
Location of load	between pads	inlet oil temperature (deg F)	120.0
Preload	0.343	type of oil	DTE—light (150SSU @100°F)

		Fluid film stiffness		Damping	
Speed (rpm)	50mm No	KXX (lb/in)	KYY (lb/in)	WCXX (lb/in)	WCYY (lb/in)
2500	0.114	1.3871E 06	7.5446E 05	7.7995E 05	4.6249E 05
3000	0.137	1.2984E 06	7.1330E 05	7.8487E 05	4.7587E 05
4000	0.183	1.1769E 06	6.6147E 05	8.0311E 05	5.0825E 05
4500	0.206	1.1341E 06	6.4543E 05	8.1400E 05	5.2564E 05
5500	0.252	1.0703E 06	6.2556E 05	8.3686E 05	5.6116E 05
6613	0.303	1.0230E 06	6.1679E 05	8.6656E 05	6.0354E 05
7000	0.321	1.0109E 06	6.1616E 05	6.7775E 05	6.1885E 05
8000	0.366	9.8751E 05	6.1898E 05	9.0798E 05	6.5935E 05
9000	0.412	9.7305E 05	6.2684E 05	9.4015E 05	7.0111E 05
10000	0.458	9.6556E 05	6.3864E 05	9.7461E 05	7.4430E 05
11000	0.504	9.6360E 05	6.5354E 05	1.0110E 06	7.8878E 05
12000	0.549	9.6610E 05	6.7094E 05	1.0490E 06	8.3434E 05
13000	0.595	9.7225E 05	6.9037E 05	1.0881E 06	8.8080E 05
14000	0.641	9.8144E 05	7.1149E 05	1.1283E 06	9.2801E 05
15000	0.687	9.9317E 05	7.3403E 05	1.1696E 06	9.7586E 05



modeling the rotor and bearings, most rotor response calculations also include the following additional inputs:

- Bearing support stiffness
- Oil film seal damping effects

Of all the input parameters, the effects of bearing and seal oil film parameters are the most difficult to calculate and measure. Therefore, a correlation difference will always exist between the predicted and actual values of critical speed. Historically, predicted values of NC_1 (first critical speed) generally agree within ± 5%. However, wide variations between predicted and actual values above the first critical speed (NC_1) exist for NC_2, NC_3, etc.

When selecting machinery, the best practice is to request specific vendor experience references for installed equipment with similar design parameters as follows:

- Bear span ÷ major shaft diameter
- Speeds
- Bearing design
- Seal design
- Operating conditions (if possible)

Once the rotor system is adequately modeled, the remaining input parameter is the amount and location of unbalance. Since the objective of the rotor response study is to accurately predict

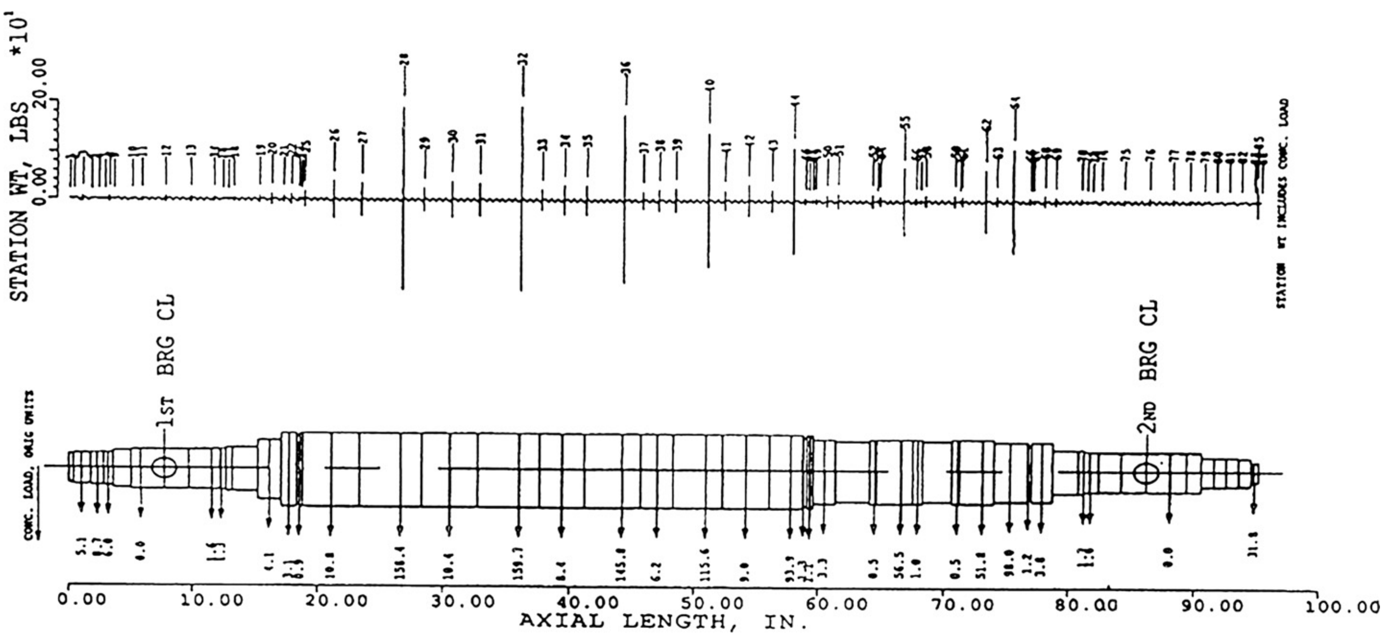


Fig 3.18.3 • Rotor response input data – dimensions, masses and unbalances (Courtesy of Elliott Co.)

the critical speed values and responses, an assumed value and location of unbalances must be defined. Other than bearing and seal parameters, unbalance amount and location is the other parameter with a 'correlation factor'. There is no way to accurately predict the amount and location of residual unbalance on the rotor. Presently, the accepted method is to input a value of 8 × A.P.I. acceptable unbalance limit $\frac{(4W)}{N}$.

This results in a rotor response input unbalance of $\frac{32W}{N}$.

The location of the unbalance is placed to excite the various critical speeds. Typically the unbalances are placed as noted below:

Location	To excite
Mid span	NC_1
Quarter span (2 identical unbalances)	NC_2
At coupling	NC_2, NC_3

Failure to accurately determine the value and location of residual rotor unbalance is one of the major causes of correlation differences between predicted and actual critical speeds.

Rotor response (output)

The output from the rotor response study yields the following:

- Relative rotor mode shapes
- Rotor response for a given unbalance

Figure 3.18.4 shows the relative rotor mode shapes for NC_1, NC_2, NC_3 and NC_4. Usually, the rotor will operate between NC_1 and NC_2.

Rotor mode shape data is important to the designer because it allows determination of modifications to change critical speed values.

For the end user, this data provides an approximation of the vibration at any point along the shaft as a ratio of the measured vibration data. As an example in Figure 3.18.4, determine the vibration at the shaft mid span if the vibration measured by the probe C_2 when operating at NC_1 is 2.00 mils. From Figure 3.18.4, the vibration at the shaft mid span when operating at the first critical speed of 3327 RPM (50 in location) is:

$$\frac{1.00}{.1} \text{ or } 10 \times \text{the bearing vibration}$$

Ten (10) times the value at C_2 or 20.0 mils!

Mode shape data should always be referred to when vibration at operating speed starts to increase and your supervisor asks

'When do we have to shut down the unit?'

or

'Can we raise the radial vibration trip setting?'

In this example, the bearing clearance may be 0.006 or 6 mils. And an honest request would be 'We'll replace the bearing at the turnaround, please run to 7.0 mils vibration'.

Refer to Figure 3.18.4 and remember:

- The compressor must go through NC_1
- The shaft vibration **increases** at NC_1 (usually 2 or 3 times, or more)
- The vibration at center span is approximately **10 times** the probe vibration

Therefore:

Vibration at the mid span during the first critical speed will be:

$= (7.0 \text{ mils}) \times (2.0) \times (10)$

Probe value NC_1 amplification Mode shape difference

$= 140$ mils!!

Normal clearance between the rotor and interstage labyrinths is typically 40 mils! This vibration exposes the diaphragms, which are usually cast iron, to breakage. One final comment; during shutdown, the rate of rotor speed decrease **CANNOT** be controlled as in the case of start-up. It depends on rotor inertia,

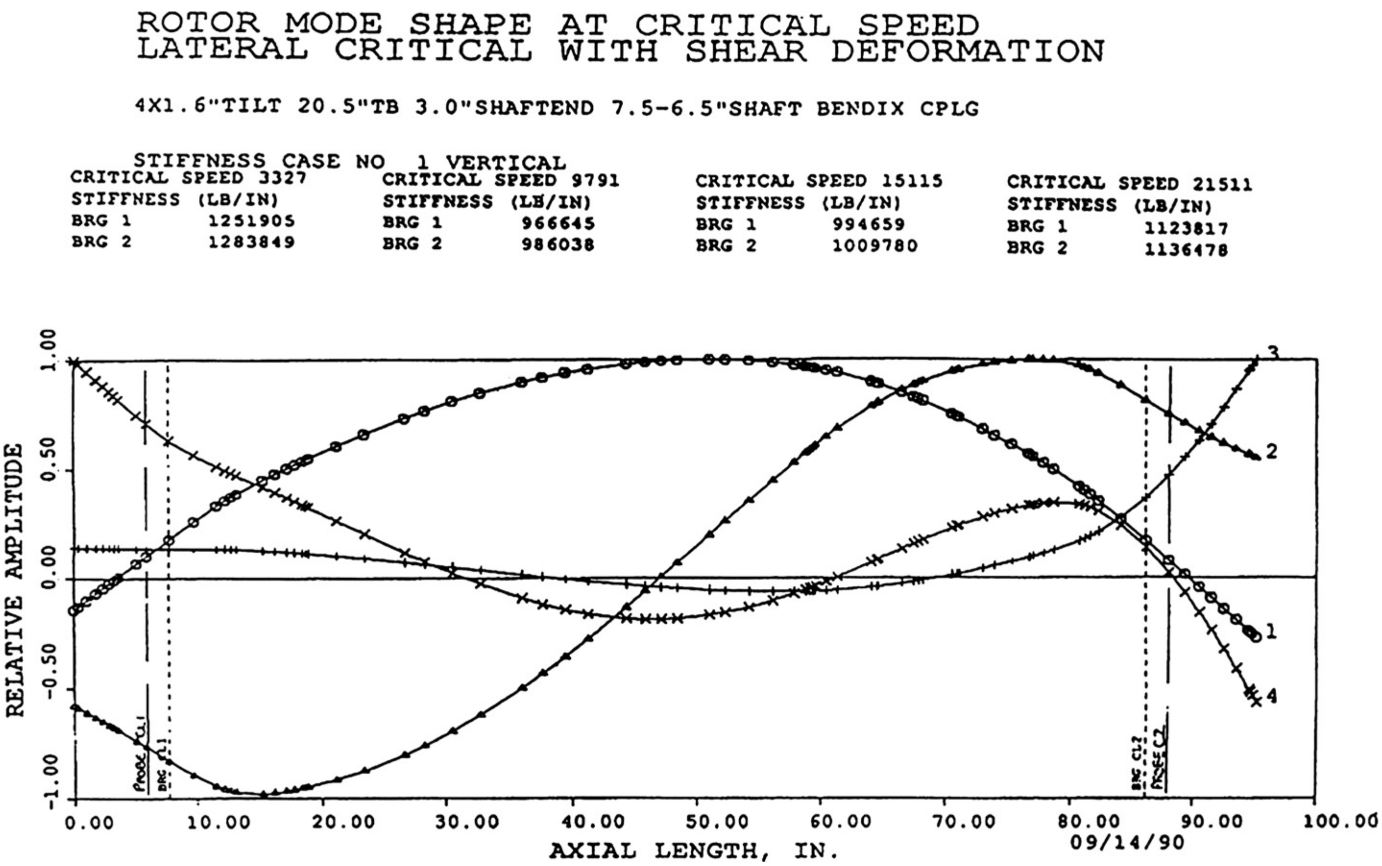


Fig 3.18.4 • Rotor natural frequency mode shapes (Courtesy of Elliott Co.)

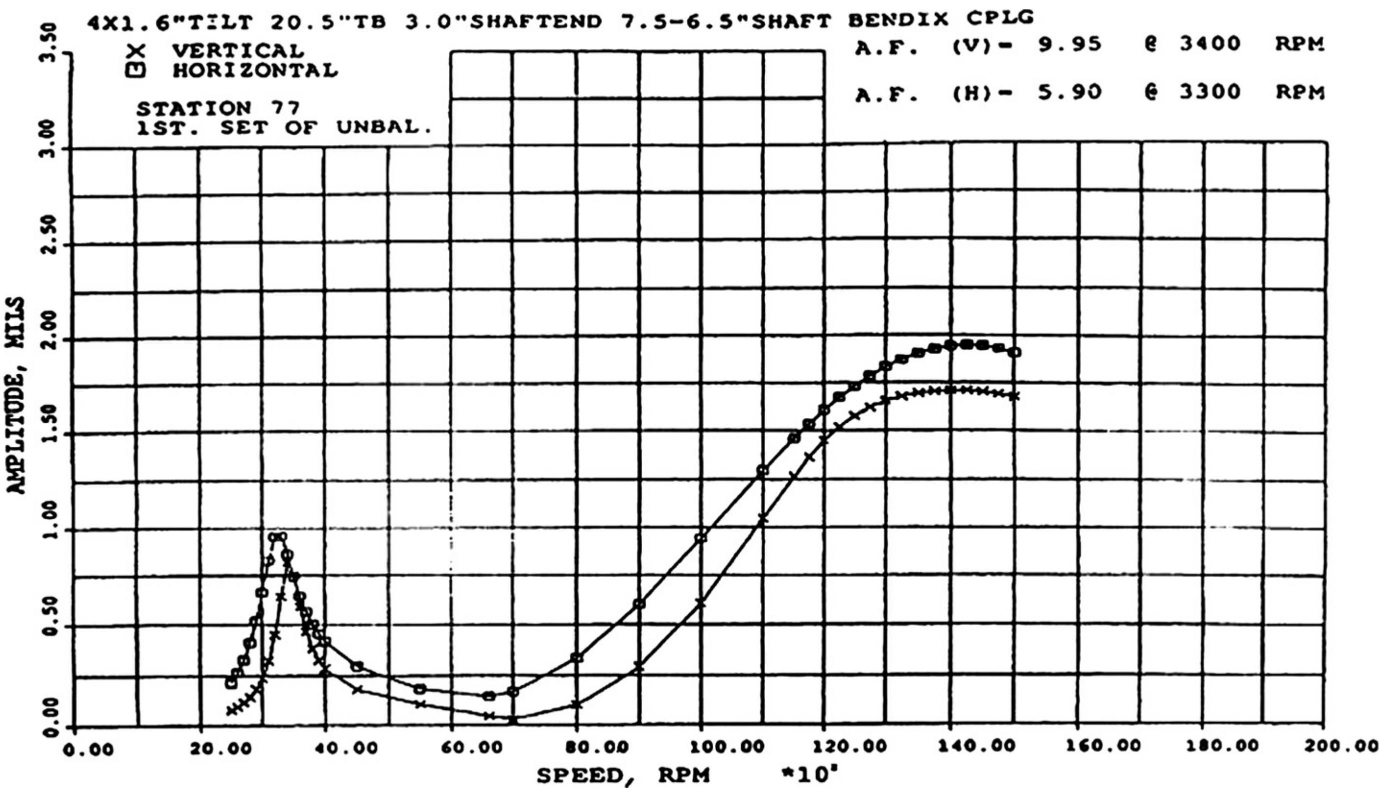


Fig 3.18.5 • Rotor response output at non-drive end bearing (N.D.E) (Courtesy of Elliott Co.)

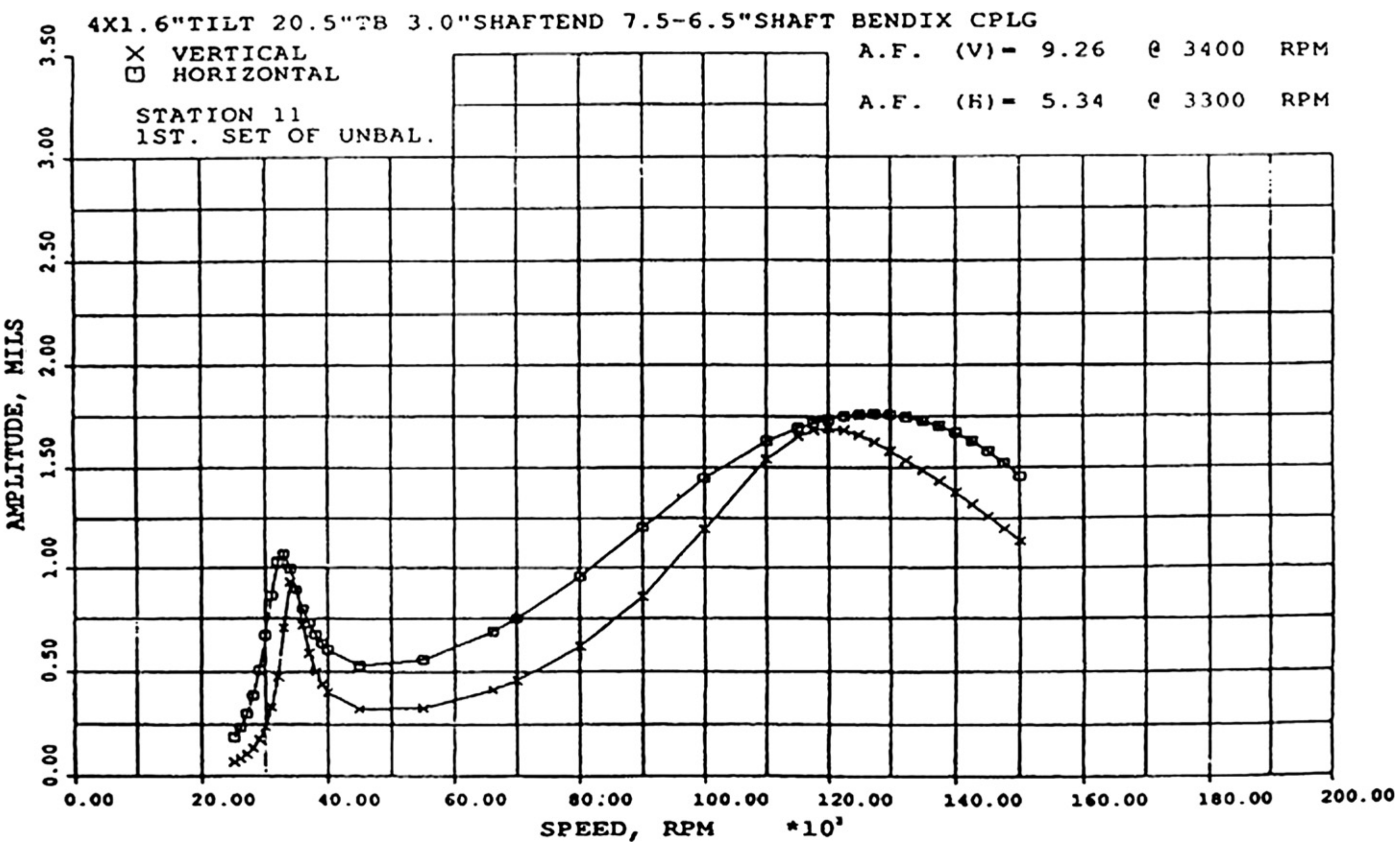


Fig 3.18.6 • Rotor response output drive end bearing (D.E.) (Courtesy of Elliott Co.)

load in the compressor, the process system characteristics and the control and protection system. If the vibration at the probe locations is high, the best advice is to stop the compressor fully loaded which will reduce the time in the critical speed range as much as possible. Yes, the compressor will surge, but the short duration will not normally damage the compressor. Figures 3.18.5 and 3.18.6 present the primary output of a rotor response study.

Rotor response plots display vibration amplitude, measured at the probes, vs. shaft speed for the horizontal and vertical probes. Note that a response curve must be plotted for each set of unbalance locations and unbalance amount.

Figure 3.18.5 shows the rotor response for the non-drive end (N.D.E.) set of probes with the first set of unbalance. Figure 3.18.6 shows the rotor response for the drive end set of probes (D.E.). The operating speed range of this example is 6,000–8,000 rpm.

Measured rotor response

During shop test, the rotor response of every turbo-compressor rotor is measured during acceleration to maximum speed and deceleration to minimum speed. Values are plotted on the same coordinates as for the rotor response analysis. The plot of shaft vibration and phase angle of unbalance vs. shaft speed is known as a **bode plot**.

Bode plots represent the actual signature (rotor response) of a rotor for a given condition of unbalance, support stiffness and unbalance. They indicate the location of the critical speeds, the change of shaft vibration with speed and the phase angle of unbalance at any speed. A bode plot is a dynamic or transient signature of vibration for a rotor system and is unique to that system for the recorded time frame. Bode plots should be recorded during every planned start-up and shutdown of every turbo-compressor. As discussed in this section, the bode plot will provide valuable information concerning shaft vibration and phase angle at any shaft speed.

Best Practice 3.19

Use tilt pad or multi-lobe type bearings and avoid lemon bore (elliptical) and offset sleeve types to ensure acceptable vibration characteristics at all speeds and loads in all between bearing and integral geared compressors.

Lemon bore (elliptical) and offset sleeve type bearings do not prevent vibration whirl and/or whip (excessive vibration at a critical speed frequency) if the attitude (bearing load) angle falls in the major axis of the bearing.

The attitude angle varies with the transmitted power in all integral gear compressors.

Many integral gear compressors still use this type of bearing.

Lessons Learned

The use of lemon bore or offset sleeve type bearings, particularly in integral geared compressors, have caused vibration issues that have resulted in long periods of downtime and large revenue losses in un-spared integral geared compressor installations.

Benchmarks

This best practice has been used since the early 1980s, after experiencing a situation where an offset sleeve bearing had to be replaced during test to meet the vibration specification prior to field operation. The prohibition of lemon bore and offset sleeve bearings since that time has resulted in integral geared compressor reliabilities of greater than 97% and centrifugal compressor reliabilities of greater than 99.7%.

B.P. 3.19. Supporting Material

Hydrodynamic bearing types

Regardless of the type of hydrodynamic bearing, all bearing surfaces are lined with a soft, surface material made from a composition of tin and lead. This material is known as Babbitt. Its melting temperature is above 200°C (400°F), but under load will begin to deform at approximately 160°C (320°F). Typical thickness of Babbitt over steel is 1.5 mm (0.060). Bearing embedded temperature probes are a most effective means of measuring bearing load point temperature and are inserted just below the Babbitt surface. RTDs or thermocouples can be used. There are many modifications available to increase the load effectiveness of hydrodynamic bearings. Among the methods available are:

- Copper backed Babbitt or 'Trimetal' – to aid in heat removal
- Back pad cooling – used on tilt pad bearings to remove heat
- Direct cooling – directing cool oil to maximum load points

A typical straight sleeve hydrodynamic journal bearing is shown in Figure 3.19.1.

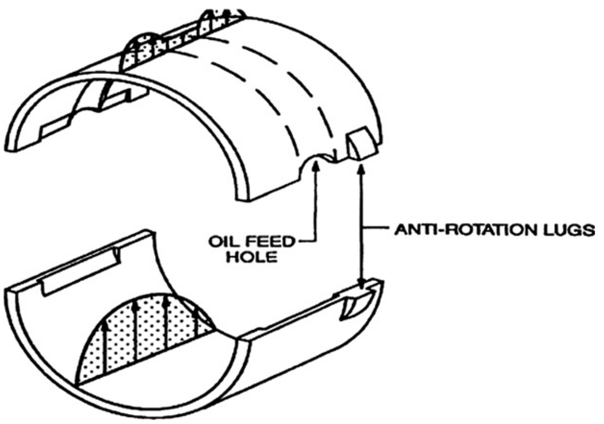


Fig 3.19.1 • Straight sleeve bearing liner (Courtesy of Elliott Co.)

Straight sleeve bearings are used for low shaft speeds (less than 5,000 RPM) or for older turbo-compressor designs. Frequently, they are modified to incorporate a pressure dam, in the direction of rotation. The pressure dam must be positioned in the top half of the bearing to increase the load vector. This action ensures that the tangential force vector will be small relative to the load vector, thus preventing shaft instability. It should be noted that incorrect assembly of the pressure dam in the lower half of the bearing will render this type of bearing unstable. Other alternatives to prevent rotor instabilities are noted in Figure 3.19.2 for use when the shaft speed is high.

Shown are examples of anti-whirl bearings. The most common types of these bearings are the 3 and 4 lobe design. Elliptical and offset bearing designs do prevent instabilities but tend to increase shaft vibration if the load vector passes through the major axis of the bearing. These types of bearings may have to be rotated in the bearing brackets to prevent this occupance.

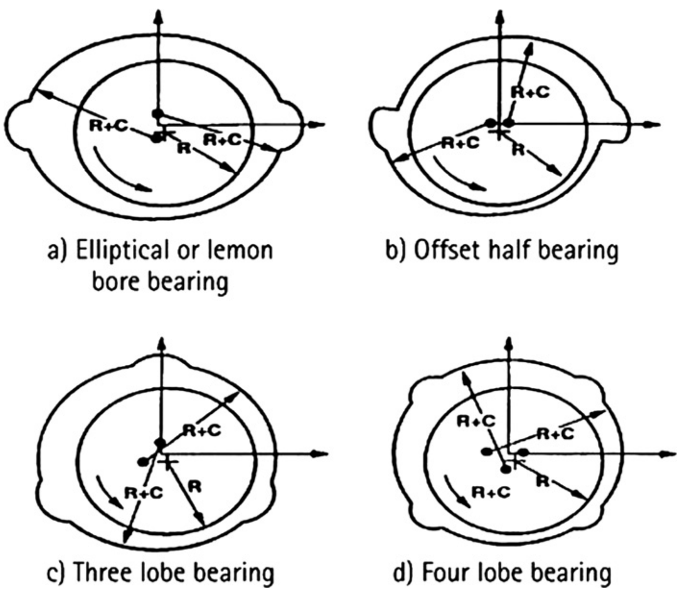


Fig 3.19.2 • Prevention of rotor instabilities

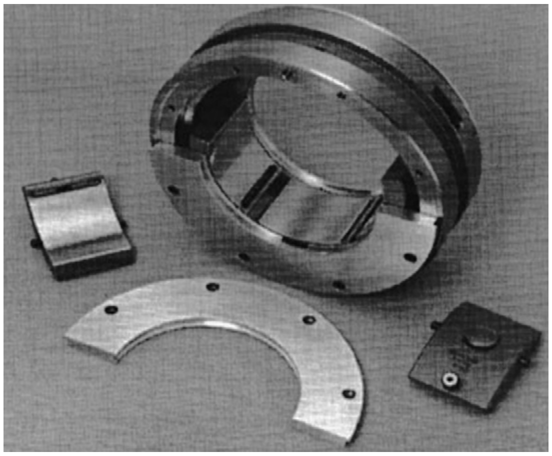


Fig 3.19.3 • Tilting pad journal bearing assembly (Courtesy of Kingsbury, Inc.)

The most common hydrodynamic bearing for higher speed applications is the tilt pad journal bearing shown in Figure 3.19.3. A tilting pad bearing offers the advantage of increased contact area, since the individual pads conform to the shaft orbit. In addition, this type is also a highly effective anti-whirl bearing since the spaces between the pads prevent oil whirl. Most end users specify tilt pad radial and thrust bearings for turbo-compressor applications.

Figure 3.19.4 shows the mechanical frictional losses and oil flow requirements for a tilt pad journal bearing as a function of shaft speed.

Note that the basis for horsepower loss and oil flow is an oil temperature rise of 16.7°C (30°F). This is the normal design ΔT for all hydrodynamic bearings. Also given in this figure is the data necessary to calculate bearing pressure at the load point.

As an exercise calculate the following for this bearing:

- Projected Area

$A_{PROJECTED} = 5" \times 2"$
$= 10$ square inches

- Pressure

$= 3479$ lb force $\div$ 10 square inches
$= 347.9$ psi on the oil film at load point

Condition monitoring

In order to determine the condition of any journal bearing, all the parameters that determine its condition must be monitored.

Figure 3.19.5 presents the eight parameters that determine the condition of a hydrodynamic journal bearing along with typical limits. You are advised to consult the manufacturer's instruction book for vendor recommended limits.

One important parameter noted in Figure 3.19.5 that is frequently overlooked is shaft position. Change of shaft position can only occur if the forces acting on a bearing change or if the bearing surface wears. Figure 3.19.6 shows how shaft position is determined using standard shaft proximity probes.

Regardless of the parameters that are condition monitored, relative change of condition determines if and when action is required. Therefore, effective condition monitoring requires the following action for each monitored condition:

- Establish baseline condition
- Record condition trend
- Establish condition limit

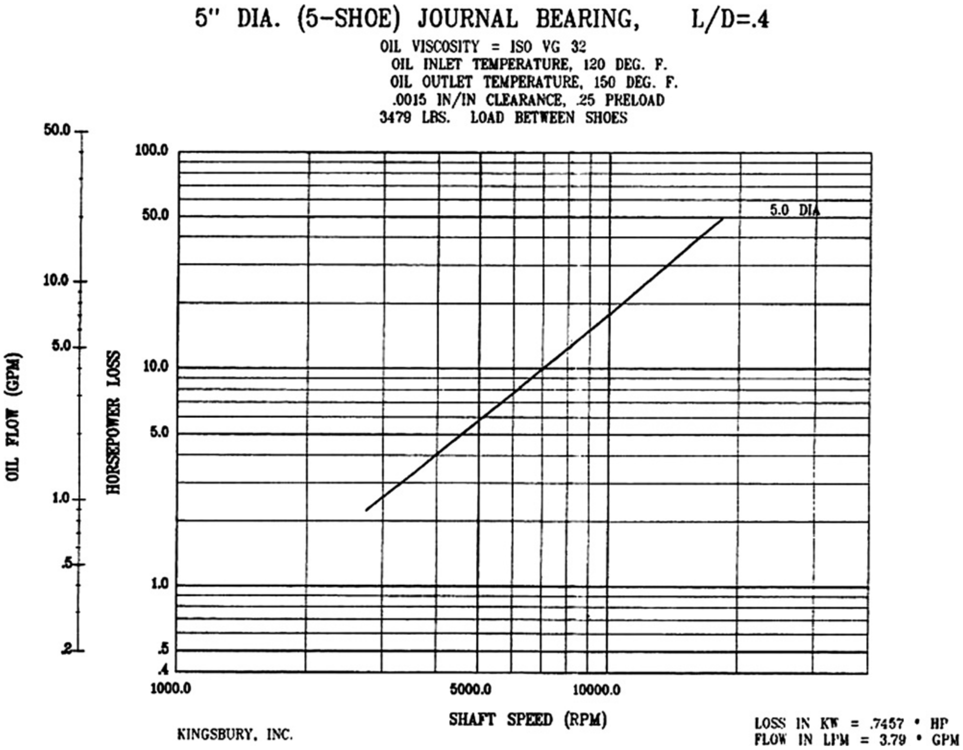


Fig 3.19.4 • Typical journal bearing selection curve (Courtesy of Kingsbury, Inc.)

Parameter	Limits
1. Radial vibration (peak to peak)	2.5 mils (60 microns)
2. Bearing pad temperature	220°F (108°C)
3. Radial shaft position (except for gearboxes where greater values are normal from unloaded to loaded operation)	> 30° change and/or 30% position change
4. Lube oil supply temperature	140°F (60°C)
5. Lube oil drain temperature	190°F (90°C)
6. Lube oil viscosity	Off spec 50%
7. Lube oil flash point	Below 200°F (100°C)
8. Lube oil particle size	Greater than 25 microns

Condition monitoring parameters and their alarm limits according to component:
1. Journal bearing (hydrodynamic)

Fig 3.19.5 • The eight parameters that determine the condition of a hydrodynamic journal bearing along with typical limits

Figure 3.19.7 presents these facts for a typical hydrodynamic journal bearing.

Based on the information shown in this trend, the bearing should be inspected at the next scheduled shutdown. A change in parameters during month 6 has resulted in increased shaft position, vibration and bearing pad temperature.

Vibration instabilities

Vibration is an important condition associated with journal bearings because it can provide a wealth of diagnostic information valuable in determining the root cause of a problem.

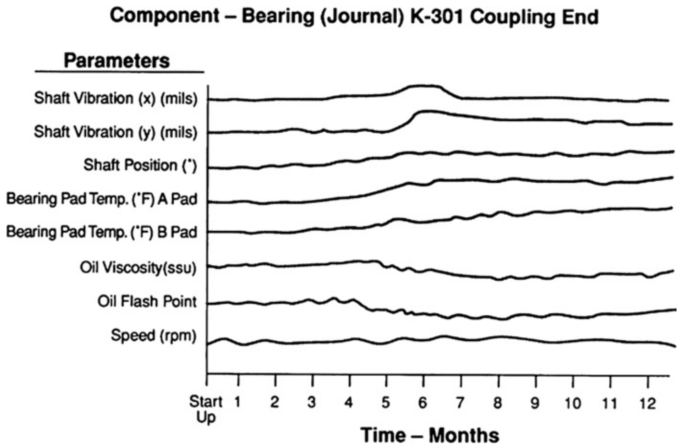


Fig 3.19.7 • Trending data for a typical hydrodynamic journal bearing

Figure 3.19.8 presents important information concerning vibration.

Figure 3.19.9 defines excitation forces with examples that can cause rotor (shaft) vibration. Turbo-compressors generally monitor shaft vibration relative to the bearing bracket using a non-contact or 'proximity probe' system as shown in Figure 3.19.9. The probe generates a D.C. eddy current which continuously measures the change in gap between the probe tip and the shaft. The result is that the peak to peak unfiltered (overall) shaft vibration is read in mils or thousandth of an inch. The D.C. signal is normally calibrated for 200 millivolts per mil. Probe gaps (distance between probe and shaft) are typically 1 mm (0.040 mils) or 8 volts D.C. to ensure the calibration curve is in the linear range. It is important to remember that this system measures shaft vibration relative to the bearing bracket

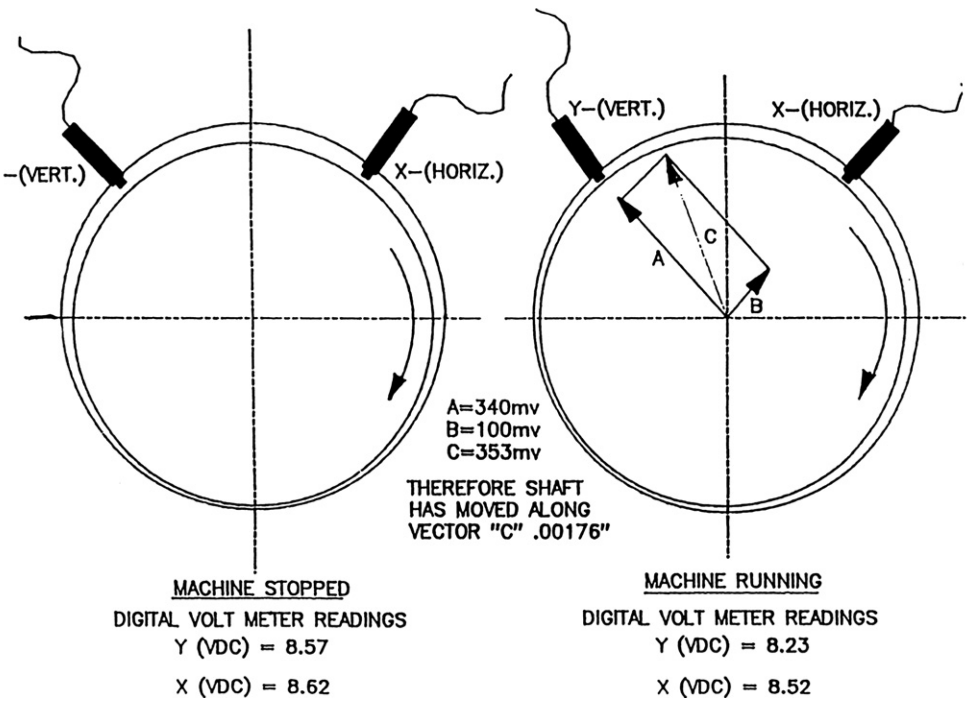


Fig 3.19.6 • Shaft movement analysis (relative to bearing bore) (Courtesy of M.E. Crane, Consultant)

- Vibration is the result of a system being acted on by an excitation.
- This excitation produces a dynamic force by the relationship:

  $F_{DYNAMIC} = Ma$

  Where: $M =$ Mass (Weight/g)

  $g =$ Acceleration due to gravity (386 IN/SEC^2)

  $a =$ Acceleration of mass M (IN/SEC^2)
- Vibration can be:
- Lateral ________ ↕ __________
- Axial→ _______________ ←
- Torsional _____________________

Fig 3.19.8 • Vibration

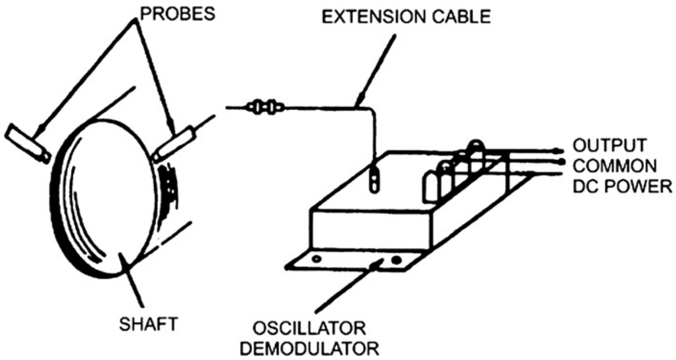


Fig 3.19.10 • Non contact displacement measuring system

and assumes the bearing bracket is fixed. Some systems incorporate an additional bearing bracket vibration monitor and thus record vibration relative to the earth or 'seismic vibration'.

As previously discussed, vibration limits are usually defined by:

$$\text{Vibration (mils p–p)} = \sqrt{\frac{12000}{RPM}}$$

This value represents the allowable shop acceptance level. API recommends alarm and trip shaft vibration levels should be set as follows:

$$V_{ALARM} = \sqrt{\frac{24000}{RPM}}$$

$$V_{TRIP} = \sqrt{\frac{36000}{RPM}}$$

In my opinion, shaft vibration alarm and trip levels should be based on the following parameters as a minimum and should be discussed with the machinery vendor prior to establishing levels:

- Application (critical or general purpose)
- Potential loss of revenue
- Application characteristics (prone to fouling, liquid, unbalance, etc.)
- Bearing clearance
- Speed
- Rotor actual response (Bode Plot)
- Rotor mode shapes (at critical and operating speeds)

Figure 3.19.11 presents a vibration severity chart with recommended action, and a schematic of a shaft vibration and shaft displacement monitor are shown in Figure 3.19.12.

As mentioned above, vibration is measured unfiltered or presents 'overall vibration'. Figure 3.19.13 shows a vibration signal in the unfiltered and filtered conditions. All vibration diagnostic work (troubleshooting) relies heavily on filtered vibration to supply valuable information to determine its root cause.

Figure 3.19.14 presents an example of a radio tuner as an analogy to a filtered vibration signal. By observing the predominant filtered frequencies in any overall (unfiltered) vibration signal, valuable information can be gained to add in the troubleshooting procedure and thus define the root cause of the problem.

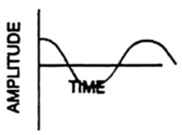
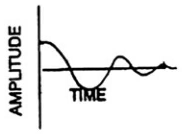
CATEGORY	EXAMPLES	EXCITATION TYPE	VIBRATION TYPE
FORCED VIBRATIONS 	UNBALANCE MISALIGNMENT PULSATION	CONSTANT CONSTANT PERIODIC	LATERAL LATERAL AND AXIAL TORSIONAL
TRANSIENT VIBRATIONS 	SYNCHRONOUS MOTOR START-UP IMPULSE (SHOCK FORCE)	RANDOM RANDOM	TORSIONAL TORSIONAL, AXIAL, RADIAL
SELF EXCITED	INTERNAL RUB OIL WHIRL GAS WHIRL	RANDOM CONSTANT CONSTANT	LATERAL LATERAL LATERAL

Fig 3.19.9 • Excitation forces with examples

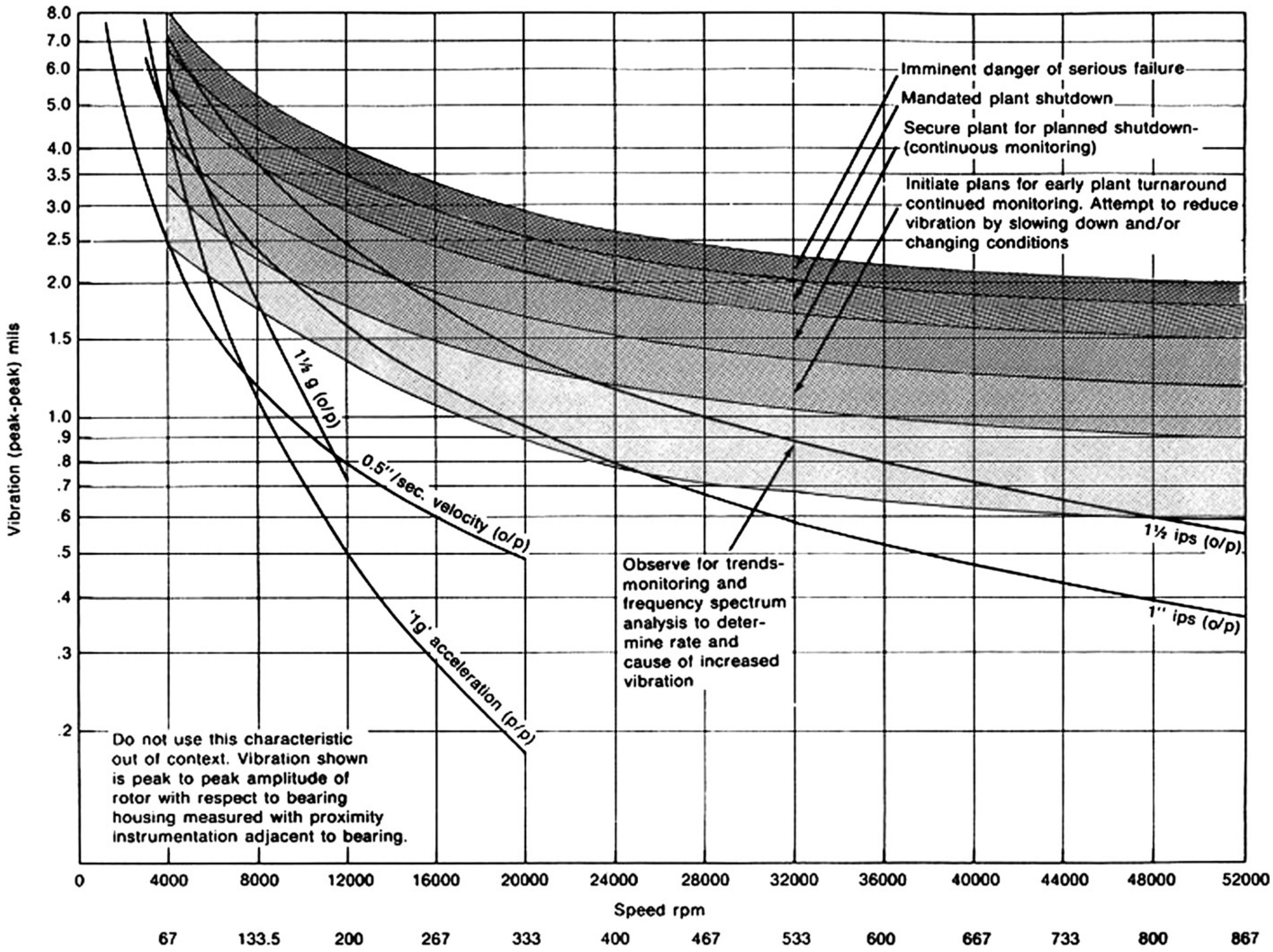


Fig 3.19.11 • Vibration severity chart (Courtesy of Dresser-Rand and C.J. Jackson P.E.)

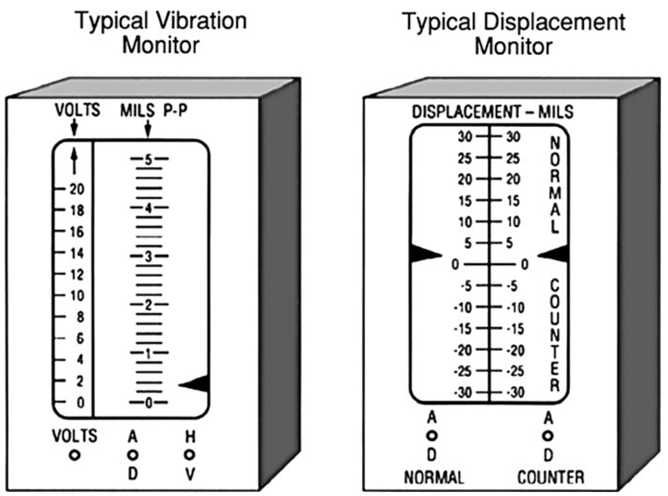
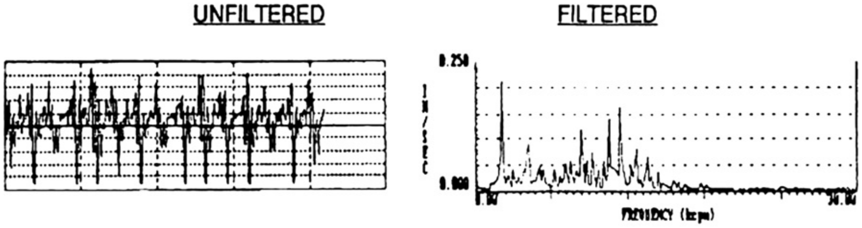


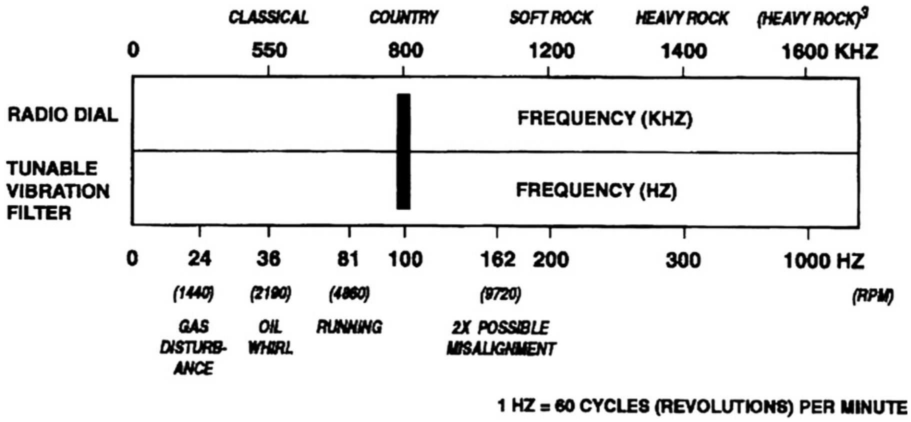
Fig 3.19.12 • Shaft vibration and displacement

ANY VIBRATION SIGNAL IS MADE UP OF ONE OR MORE FREQUENCIES. A TYPICAL UNFILTERED (OVERALL) AND FILTERED SIGNAL ARE SHOWN BELOW:



AN ANALOGY TO A FILTERED SIGNAL IS A RADIO. IN A GIVEN LOCALITY, MANY STATIONS ARE TRANSMITTING SIMULTANEOUSLY. ANY GIVEN STATION IS OBTAINED BY ADJUSTING THE TUNER TO THE CORRECT TRANSMISSION FREQUENCY.

Fig 3.19.13 • Vibration frequency



By Adjusting the Tuner (Filter) to a Selected Station (Frequency) the Desired Program (Vibration Frequency Signal) can be Obtained if it is "On the Air" (Present)

Fig 3.19.14 • Radio tuner/vibration filter analogy

Best Practice 3.20

Always check thrust pad temperature before changing axial displacement alarm and trip settings, to confirm that there is no excessive force being exerted on the thrust bearing assembly.

Hydrodynamic thrust bearings are typically monitored by dual axial shaft displacement probes and thrust pad resistance temperature detectors in two pads on each side of the thrust collar.

To ensure that axial displacement movement is real, always confirm that thrust bearing pad temperatures are greater than 108°C (225°F). If the thrust pad temperature is less than 108°C (225°F), the axial displacement movement is the result of thrust assembly or bearing bracket movements.

If thrust bearing assembly movements are confirmed, i.e. thrust bearing pad temperatures are less than 108°C (225°F), thrust bearing alarms and trip temperatures can be increased to prevent unnecessary trips.

If it is known that movement of the thrust bearing assembly components is causing alarms and trips, obtain the operating thrust value from the equipment vendor, and use a hydraulic jack to load the thrust bearing assembly to operating values when the thrust clearance is being set.

Lessons Learned

The majority of plants that we visit have shutdown on axial displacement when the shutdowns could have been avoided by checking bearing pad temperatures and increasing alarm and trip settings if this is below 108°C.

Benchmarks

I have used this best practice since the 1990s when my article 'Which way is it going?' appeared in the Bently Nevada Orbit Magazine. This best practice has saved many plant shutdowns on axial displacement trips. These unnecessary shutdowns can significantly affect product revenue and quickly reduce machine reliability values.

B.P. 3.20. Supporting Material

In every rotating machine utilizing reaction type blading, a significant thrust is developed across the rotor by the action of the impellers or blades. Also in the case of equipment incorporating higher than atmospheric suction pressure, a thrust force is exerted in the axial direction as a result of the pressure differential between the pressure in the case and atmospheric pressure.

In this section we will cover a specific rotor thrust example, and calculate thrust balance for a specific case. We will see the necessity of employing an axial force balance device, known as a balance drum, in some applications. In many instances, the absence of this device will result in excessive axial (thrust) bearing loadings. For the case of a machine with a balance device, the maintenance of the clearances on this device is of utmost importance. In many older designs, the clearances are maintained by a fixed, close-clearance bushing made out of Babbitt, which has a melting temperature of approximately 175°C (350°F), depending on the pressure differential across the balance drum. If the temperature in this region should exceed this value, the effectiveness of the balance drum will suddenly be lost and catastrophic failures can occur inside the machine. Understanding the function of this device and the potential high axial forces involved in its absence is a very important aspect of condition monitoring of turbo-compressors.

We will also examine various machine configurations, including natural balanced (opposed) thrust, and see how thrust values change even in the case of a balanced machine, as a function of machine flow rate.

Finally, we will examine thrust system condition monitoring and discuss some of the confusion that results with monitoring these machines.

The hydrodynamic thrust bearing

A typical hydrodynamic double acting thrust bearing is pictured in Figure 3.20.1.

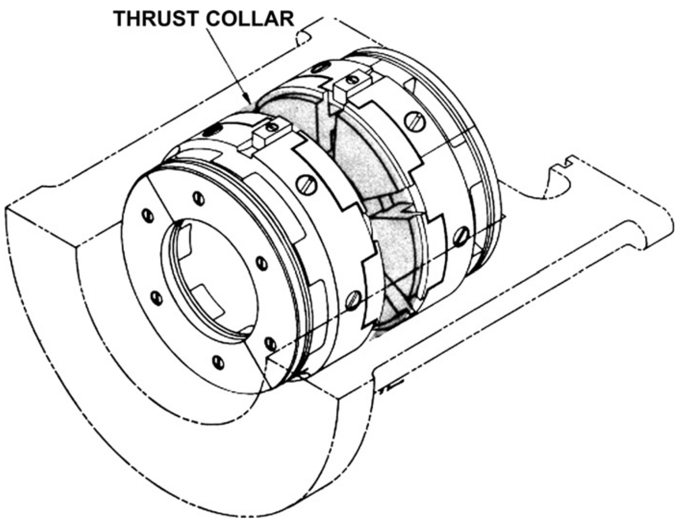


Fig 3.20.1 • Double acting self-equalizing thrust bearing assembly (thrust collar removed) (Courtesy of Elliott Company)

The thrust bearing assembly consists of a thrust collar, mounted on the rotor and two sets of thrust pads (usually identical in capacity) supported by a base ring (Michell type).

The Kingsbury type includes a set of leveling plates between each set of pads and the base ring. This design is shown in Figure 3.20.2.

Both the Michell and Kingsbury types are used. Figure 3.20.3 provides a view of the leveling plates providing the self-equalizing feature in the Kingsbury design. The self-equalizing feature allows the thrust pads to lie in a plane parallel to the thrust collar.

Regardless of the design features, the functions of all thrust bearings are:

- To continuously support all axial loads
- To maintain the axial position of the rotor

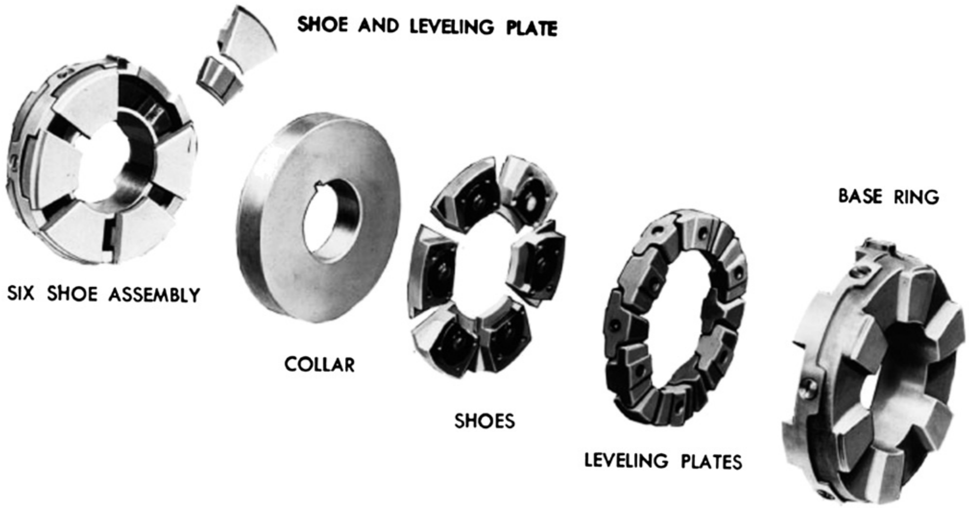


Fig 3.20.2 • Small Kingsbury six-shoe, two direction thrust bearing. Left-hand group assembled, except for one shoe and 'upper' leveling plate. Right-hand group disassembled (Courtesy of Kingsbury, Inc.)

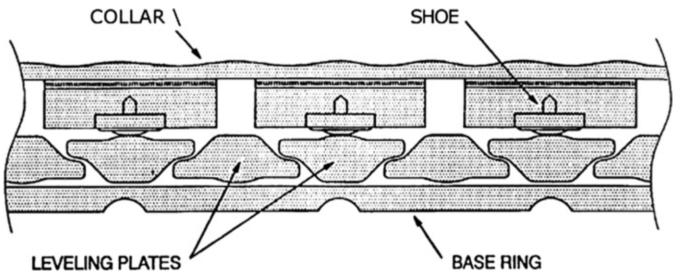


Fig 3.20.3 • Self-equalizing tilt-pad thrust bearing (view – looking down on assembly) (Courtesy of Kingsbury, Inc.)

The first function is accomplished by designing the thrust bearing to provide sufficient thrust area to absorb all thrust loads without exceeding the support film (oil) pressure limit (approximately 500 psi).

Figure 3.20.4 shows what occurs when the support film pressure limit is exceeded.

The oil film breaks down, thus allowing contact between the steel thrust collar and soft thrust bearing pad overlay (Babbitt). Once this thin layer (1/16") is worn away, steel to steel contact occurs resulting in significant turbo-compressor damage.

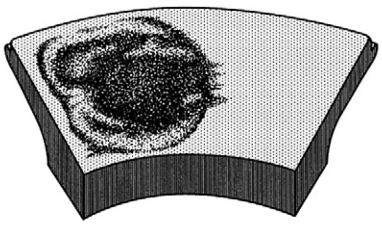


Fig 3.20.4 • Evidence of overload on a tilt-pad self-equalizing thrust bearing pad (Courtesy of Kingsbury, Inc.)

Thrust pad temperature sensors, located directly behind the Babbitt, at the pad maximum load point protect the compressor by tripping the unit before steel-to-steel contact can occur.

Figure 3.20.5 presents different Kingsbury bearing size rated capacities as a function of speed.

Figure 3.20.6 shows how thrust pad temperature and thrust load are related for a given thrust bearing size and shaft speed. Note that the greater the thrust load (psi), the smaller the oil film and hence the greater the effect of oil viscosity on oil flow and heat removal. Based on a maximum load of 3,448 kPa (500 psi), it can be seen from Figure 3.20.6 that a turbo-compressor thrust bearing pad temperature trip setting should be between 127 and 132°C (260 and 270°F).

As well as to support the rotor in an axial direction, the other function of the thrust bearing is to continuously maintain the axial position of the rotor. This is accomplished by locating stainless steel shims between the thrust bearing assembly and compressor axial bearing support plates. The most common thrust assembly clearance with the thrust shims installed is 0.275–0.35 mm (0.011–0.014"). These values vary with thrust bearing size. The vendor instruction book must be consulted to determine the proper clearance.

The following procedure is used to ensure that the rotor is properly positioned in the axial direction.

1. With thrust shims removed, record total end float by pushing rotor axially in both directions, typically 6.35–12.7 mm (0.250–0.500").
2. Position rotor as stated in instruction book.
3. Install minimum number of stainless steel thrust shims to limit end float to specified value.*

Proper running position of the rotor is critical to obtaining optimum efficiency and preventing axial rubs during transient and upset conditions (start-up, surge, etc.)

*An excessive number of thrust shims act as a spring resulting in a greater than specified axial clearance during full thrust load conditions.

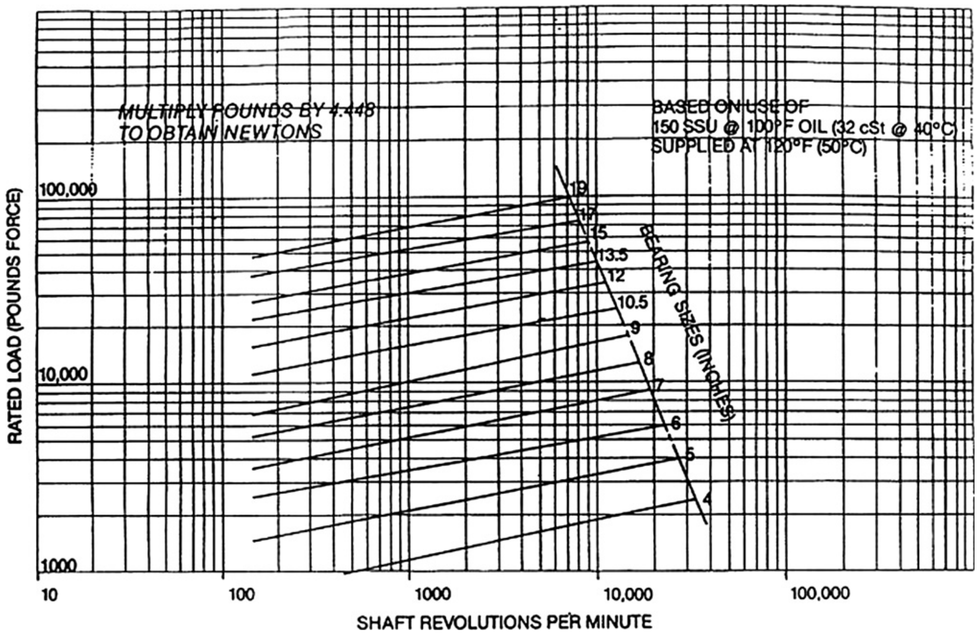


Fig 3.20.5 • Thrust bearing rated load vs. speed (Courtesy of Kingsbury Corp.)

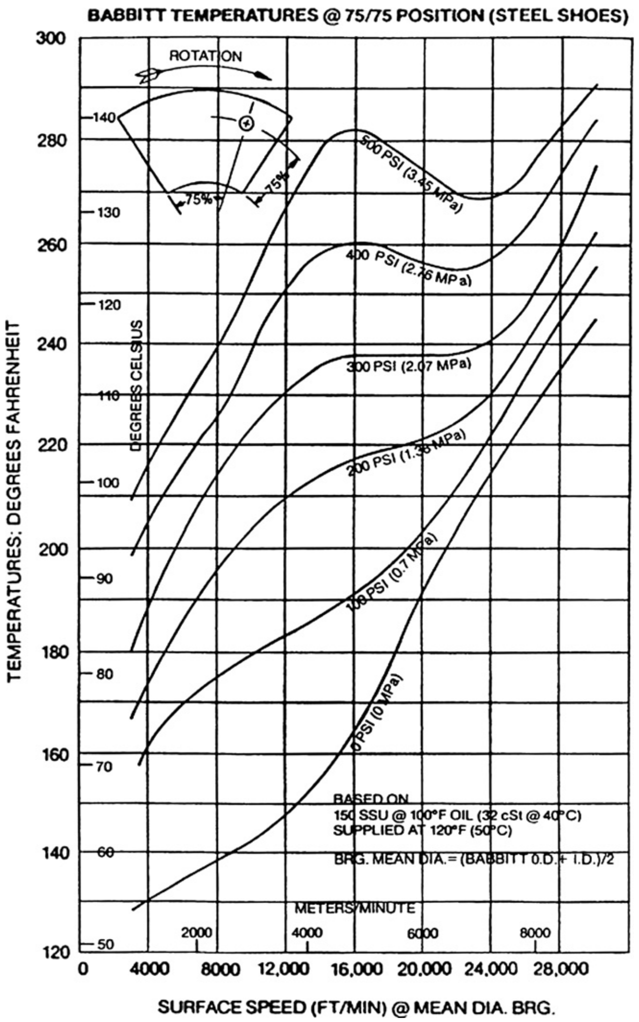


Fig 3.20.6 • The relationship between thrust pad temperature and thrust load (Courtesy of Kingsbury, Inc.)

Impeller thrust forces

Every reaction type compressor blade set or impeller produces an axial force towards the suction of the blade or impeller. Refer to Figure 3.20.7.

In this example, the net force towards the compressor suction is 8,900 N (2,000 lbs) for the set of conditions noted. Note that the pressure behind the impeller is essentially constant at 344.75 kPa (50 psi), but the pressure on the front side of impeller varies from 344.75 to 275.8 kPa (50 to 40 psi), because of the pressure drop across the eye labyrinth. Every impeller in a multistage compressor will produce a specific value of axial force towards its suction at a specific flow rate, speed and gas composition. A change in any or all of these parameters will produce a corresponding change in impeller thrust.

Rotor thrust balance

Figure 3.20.8 shows how a balance drum or opposed impeller design reduces thrust force. The total impeller force is the sum of the forces from the individual impellers. If the suction side of the impellers is opposed, as noted in Figure 3.20.8, the thrust force will be significantly reduced and can approach zero. If the suction side of all impellers are the same (in series), the total impeller thrust force can be very high, and may exceed the thrust bearing rating. If this is the case, a balance drum must be mounted on the rotor as shown in Figure 3.20.8. The balance drum face area is varied such that the opposing force generated by the balance drum reduces the thrust bearing load to an acceptable value. The opposing thrust force results from the differential between compressor discharge pressure (P_F) and compressor suction pressure (P_1) since the area behind the balance drum is usually referenced to the suction of the compressor. This is accomplished by a pipe that connects this chamber to the compressor suction. This line is typically called the 'balance line'.

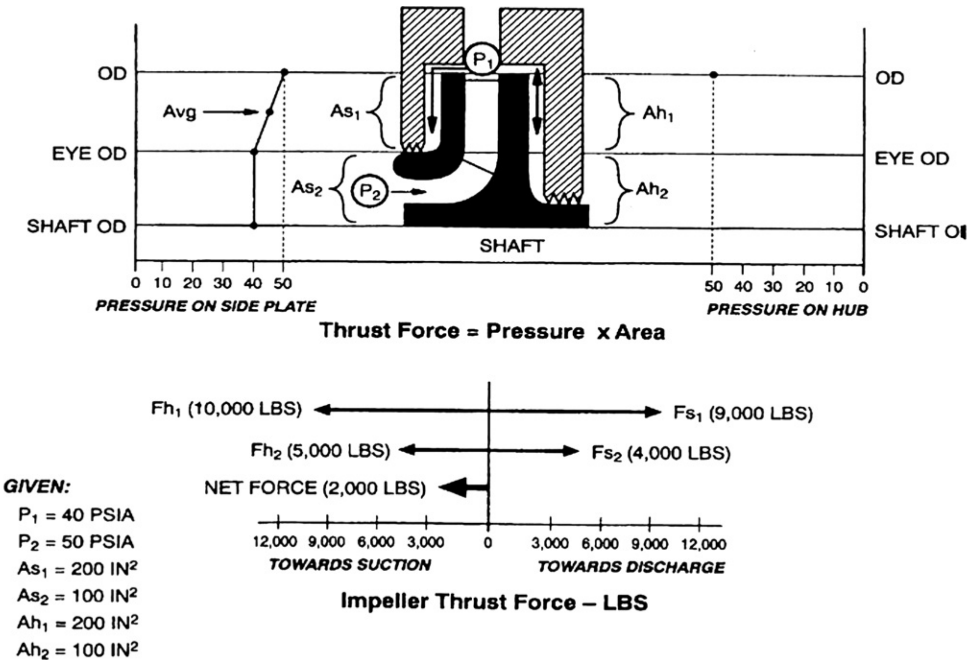


Fig 3.20.7 • Impeller thrust force

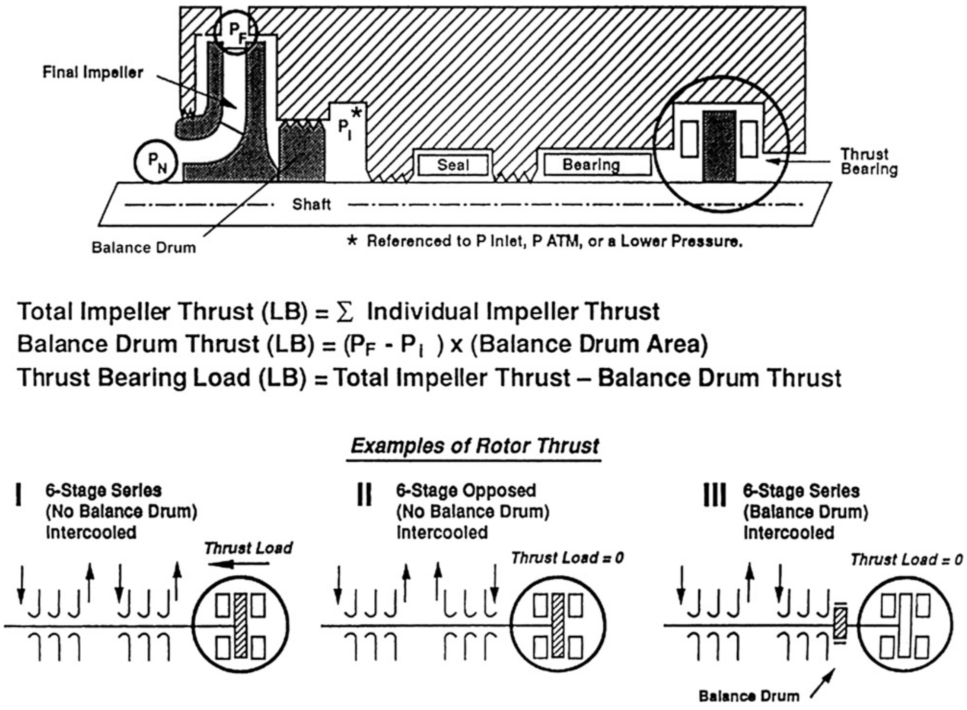


Fig 3.20.8 • Rotor thrust force

It is very important to note that a balance drum is used only where the thrust bearing does not have sufficient capacity to absorb the total compressor axial load. And the effectiveness of the balance drum depends directly on the balance drum seal. Fail the seal (open clearance significantly), and thrust bearing failure can result.

A common misunderstanding associated with balance drum systems is that a balance drum always reduces the rotor thrust to zero. Refer to Figure 3.20.9 and observe that this statement may or may not be true depending on the thrust balance system design. And even if it is, the thrust is zero only at one set of operating conditions.

Figure 3.20.9 shows a rotor system designed four (4) different ways. Note how the thrust **always** changes with the flow rate regardless of the design. Another misconception regarding thrust balance systems is the normal or 'active' direction of thrust. In many cases, the active thrust is assumed to always be towards the suction of the compressor.

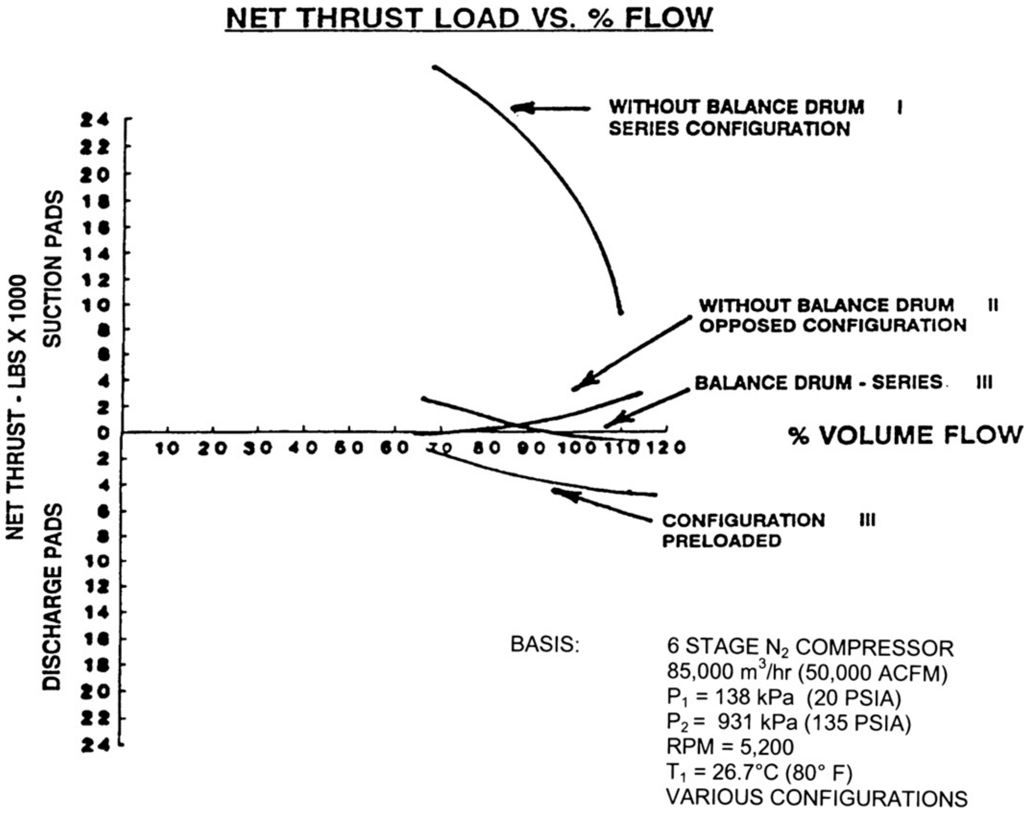


Fig 3.20.9 • Rotor system designed four different ways

Observing Figure 3.20.8, it is obvious that the ‘active’ direction can change when the turbo-compressor has a balance drum or is an opposed design. It is recommended that the use of active thrust be avoided where possible, and that axial displacement monitors be labeled to allow determination of the thrust direction at all times.

Please refer to Figure 3.20.10 which shows a typical thrust displacement monitor.

These monitors detect thrust position by targeting the shaft end, thrust collar or other collar on the rotor. Usually two or three probes (multiple voting arrangement) are provided to eliminate unnecessary compressor trips. The output of the probes is noted on the monitor as either + (normal) or − (counter). However, this information gives no direct indication of the axial direction of the thrust collar. The following procedure is recommended:

1. With compressor shutdown, push rotor towards the suction and note direction of displacement indicator.
2. Label indicator to show direction towards suction of compressor.

Knowing the actual direction of the thrust can be very useful during troubleshooting exercises in determining the root cause of thrust position changes.

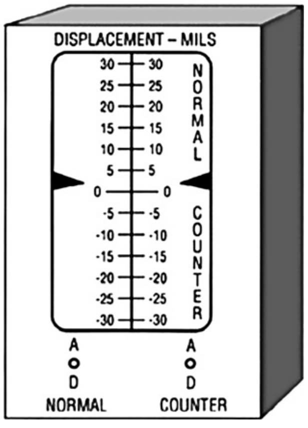


Fig 3.20.10 • Typical axial thrust monitor

Thrust condition monitoring

Failure of a thrust bearing can cause long term and possibly catastrophic damage to a turbo-compressor. Condition monitoring and trending of critical thrust bearing parameters will optimize turbo-compressor reliability.

The critical thrust bearing condition monitoring parameters are:

- Rotor position
- Thrust pad temperature
- Balance line ΔP

Rotor position is the most common thrust bearing condition parameter and provides useful information regarding the direction of thrust. It also provides an indication of thrust load but does not confirm that thrust load is high. Refer to Figure 3.20.10.

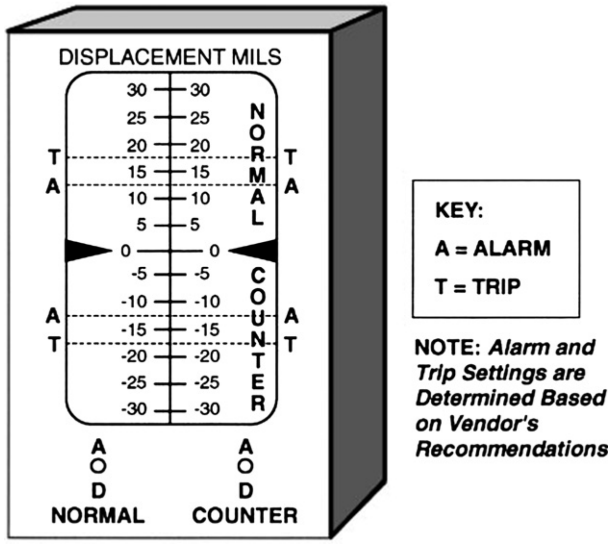


Fig 3.20.11 • Typical axial displacement monitor

All axial displacement monitors have pre-set (adjustable) values for alarm and trip in both thrust directions. Typically, the established procedure is to record the thrust clearance (shims installed) during shutdown and set the alarm and trip settings as follows:

$$\text{Alarm} = \frac{\text{Clearance}}{2} + 10 \text{ mils (each direction)}$$
$$\text{Trip} = \text{Alarm Setting} + 5 \text{ mils (each direction)}$$

The above procedure assumes the rotor is in the mid or zero position of the thrust clearance. An alternative method is to push the rotor by hand to the assumed active position and add appropriate values for alarm and trip.

I personally recommend the first method since an active direction of thrust does not have to be assumed.

As noted, axial displacement monitors only indicate the quantity of thrust load. False indication of alarm or even trip settings can come from:

- Compression of thrust bearing components
- Thermal expansion of probe adaptors or bearing brackets
- Loose probes

It is strongly recommended that any alarm or trip displacement value be confirmed by thrust pad temperature if possible prior to taking action. Please refer back to Figure 3.20.6 and note that the thrust pad temperature in the case of thrust pad overload is approximately 121°C (250°F). If an axial displacement alarm or trip signal is activated, **observe** the corresponding thrust pad temperature. If it is below 104°C (220°F), take the following action:

- Observe thrust pads. If no evidence of high load is observed (pad and back of pad) confirm calibration of thrust monitor and change settings if necessary.

The last condition monitoring parameter for the thrust system is balance line pressure drop. An increase of balance line ΔP will indicate increased balance drum seal leakage and will result in higher thrust bearing load. Noting the baseline ΔP of the balance line and trending this parameter will provide valuable information as to the root cause of a thrust bearing failure. In many field case histories, the end user makes many thrust bearing replacements before an excessive balance drum clearance is discovered as the root cause of the thrust bearing failure. It is a good practice to always check the balance line ΔP after reported machine surge. Surging will cause high internal gas temperatures which can damage the balance drum seal.

Best Practice 3.21

Monitor balance devices by trending balance line differential pressure to ensure that the balance system components are serviced only during a turnaround.

Balance device replacement requires complete disassembly of the compressor which can result 5–7 days downtime. Modern large (mega) plants can lose as much as $5MM per day in revenue.

The only method of condition monitoring the balance system is to monitor balance line or seal equalization line (in back to back compressors) differential pressure. Balance chamber pressure monitoring does not ensure accurate results since the suction pressure can change.

If balance or equalization line differential pressure increases at the same time that thrust bearing pad temperature and axial displacement are increasing, the balance device should be replaced at the next turnaround.

Lessons Learned

Thrust bearing assemblies are frequently changed without considering balance system differential pressure trends, only to find that balance device deterioration is the root cause and compressor disassembly is required, forcing a 5–7 day loss of revenue.

Benchmarks

This best practice has been used since 1990, and has required the installation of differential pressure transmitters to enable monitoring of balance line or equalization line differential pressure. This action has limited balance device maintenance to turnarounds and produced compressor reliabilities which exceed 99.7%

B.P. 3.21. Supporting Material

Refer B.P. 3.20 for supporting material on impeller thrust forces and balance device design.

Best Practice 3.22

Oil seals should be designed or modified to incorporate a flow through port if the difference between operating seal reference pressure and start-up pressure is greater than 1,375 kPa (200 psi).

All compressor oil seals incorporate an atmospheric bushing. The atmospheric bushing flow is directly proportional to the seal reference gas pressure since the seal system provides sealing oil at some constant differential pressure above the reference pressure.

In many seal designs, the atmospheric bushing clearance provides the only means of seal cooling and is designed to provide sufficient cooling oil flow for the rated operating reference gas pressure.

If the reference pressure at start-up is less than 1,375 kPa (200 psid) than the operating reference pressure, insufficient atmospheric bushing flow will be present during start-up, and bushing wear can result and cause a compressor trip on low differential seal oil pressure.

Incorporating a flow through port into the seal design will allow sufficient cooling oil flow during low reference gas conditions that will occur during start-up.

Lessons Learned

Compressor applications using oil seals in high suction pressure applications frequently have low seal MTBFs. RCFA investigations show that low pressure start-up conditions (nitrogen purge operation, etc.) with seals not using a flow through design is the root cause of low MTBFs.

Benchmarks

This best practice has been used since 1990, during the design phase to require flow-through type seal designs for high suction pressure compressor applications where there is a difference of more than 1,375 kPa (200 psi) between start-up and operating reference pressures.

Seal designs can be modified in the field to incorporate flow-through designs, but require redesign of the seal oil system and seal housing modifications in the field.

Use of flow through oil seals has resulted in seal MTBFs in excess of 100 months, and compressor reliabilities in excess of 99.7%.

B.P. 3.22. SUPPORTING MATERIAL

There are numerous types of fluid seal systems since the types of seals utilized, sealing fluids and sealing pressures vary widely. Regardless of the type of seal used, the function of a critical equipment seal system is as follows:

> 'To continuously supply clean fluid to each specified seal interface point at the required differential pressure, temperature and flow rate'.

A typical seal system for a centrifugal compressor is shown in Figure 3.22.1.

The system shown is for use with clearance bushing seals. Let's examine this figure by proceeding through the system; from the seal oil reservoir, through the compressor shaft seal and back through the reservoir. As previously discussed, the concept of sub-systems can be useful here. The seal oil system shown can be divided into four major sub-systems:

A The supply system
B The seal housing system
C The atmospheric drain system
D The seal leakage system

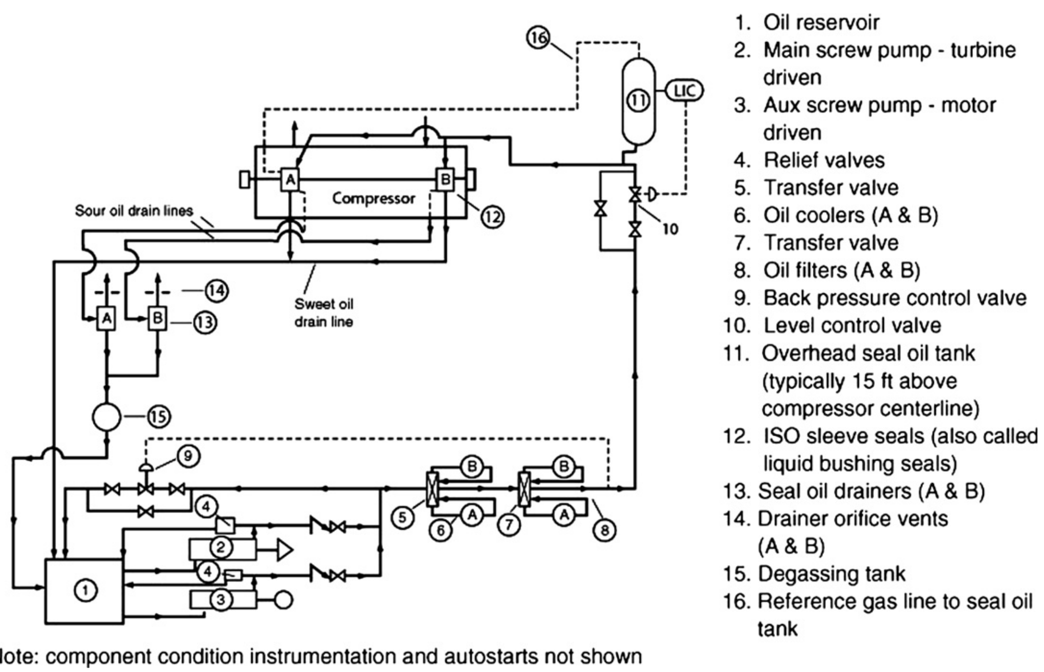


Fig 3.22.1 • API 614 lube/seal oil system for ISO-sleeve seals (Courtesy of Elliott Co.)

A. The supply system

This system consists of the reservoir, pumping units, exchangers, transfer valves, temperature control valves, and filters. The purpose of this sub-system is to continuously supply clean, cool sealing fluid to the seal interfaces at the correct differential pressure.

B. The seal housing system

This system is comprised of two different seals; a gas side bushing, and an atmospheric bushing. The purpose of the seal housing system is to positively contain the fluid in the compressor and not allow leakage to the atmosphere. The seal fluid is introduced between both seal interfaces, thus constituting a double seal arrangement. Refer to Figure 3.22.2 for a closer examination of the seal.

The purpose of the gas side bushing seal is to constantly contain the reference fluid and minimize sour oil leakage. This bushing can be conceived as an equivalent orifice. This concept is similar to bearings previously discussed, with the exception that the referenced downstream pressure of the gas side bushing can change. In order to ensure a constant flow across this 'orifice', the differential pressure must be kept constant. Therefore, every compressor seal system is designed to maintain a constant differential against the gas side seal. The means of obtaining this objective will be discussed as we proceed.

The other seal in the system is the atmospheric bushing, whose purpose is to minimize the flow of seal liquid to an amount that will remove frictional heat from the seal. This bushing can be conceptualized as a bearing, since the downstream pressure is usually atmospheric pressure. In systems that directly feed into a bearing, the atmospheric bushing downstream pressure will be constant, at approximately 138 kPa (20 psi). However, the upstream supply pressure will vary with the pressure required by the sealing media in the compressor.

As an example, if a seal system is designed to maintain a constant differential of 34.5 kPa (5 psi) per square inch between the compressor process gas and the seal oil supply to the gas side bushing, the supply pressure with zero process gas pressure would be 34.5 kPa (5 psi) to both the gas side bushing and atmospheric bushing. Therefore, gas side bushing and the atmospheric bushing differential would both be equal to 34.5 kPa (5 psi). If the process gas pressure were increased to 138 kPa (20 psi), the seal oil system would maintain a differential of 34.5 kPa (5 psi) across the gas side seal, and the supply pressure to the gas side bushing and atmospheric bushing would be 172 kPa (25 psi). In this case, the differential across the gas side bushing would remain constant at 34.5 kPa (5 psi), but the atmospheric bushing differential pressure would increase from 34.5 to 172 kPa (5 to 25 psi). As a result, a primary concern in any seal liquid system is the assurance that the atmospheric bushing receives proper fluid flow under all conditions. After the seal fluid exits the seal chamber, it essentially returns through two additional subsystems.

C. The atmospheric draining system

The flow from the atmospheric bushing, if it does not directly enter the bearing system, will return to the seal oil reservoir. In addition, flow from any downstream control valve will also return through the atmospheric drain system to the seal oil reservoir. Both these streams should be gas free since they should not come in contact with the process gas.

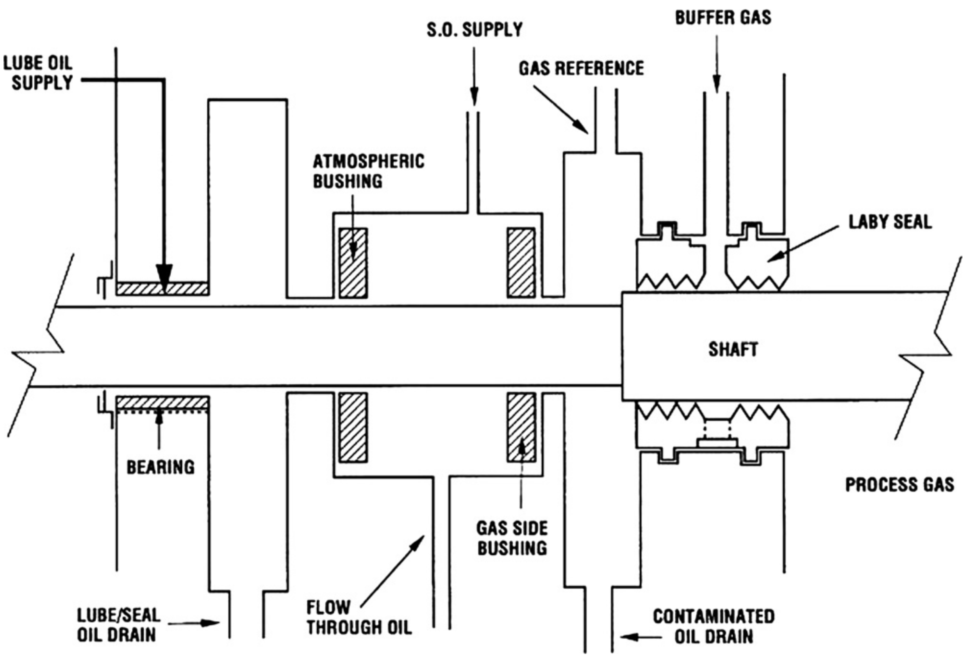


Fig 3.22.2 • Bushing seal schematic
(Courtesy of M.E. Crane, Consultant)

D. The seal leakage system

The fluid that enters the gas side bushing is controlled to a minimum amount, such that it can be either discarded or properly returned to the reservoir after it is degassed. Typically, this amount is limited to less than 77 liters (20 gallons) per day, per seal. Since this liquid is in contact with the high speed shaft it is atomized and combines with sealing gas to enter the leakage system. This system consists of:

- An automatic drainer
- A vent system
- Degassing tank (if furnished)

The function of each component is as follows:

The drainer

The drainer contains the oil-gas mixture from the gas side seal. The liquid level under pressure in the drainer is controlled by an internal float, or external level control valve, to drain oil back to the reservoir or the de-gassing tank, as required.

The vent system

The function of the seal oil drainer vent system is to ensure that all gas side seal oil leakage is directed to the drainer. This is accomplished by referencing the drainer vent to a lower pressure than the pressure present at the gas side seal in the compressor. The drainer vent can be routed back to the compressor suction, suction vessel or a lower pressure source.

The degassing tank

This vessel is usually a heated tank, with ample residence time (72 hours or greater) to sufficiently de-gas all seal oil, such that it will be returned to the reservoir and meet the seal oil specification (viscosity, flashpoint, dissolved gasses, etc.). These items will be discussed in detail later.

We will now proceed to discuss each of the major sub-systems in detail, defining the function of each such that the total operation of a seal system can be simplified.

The supply system

Referring to definition of a seal oil system, it can be seen that its function is identical to that of a lube oil system, with one exception. This is that the seal fluid must be delivered to the seals at the specified differential pressure. Let's examine this requirement further.

Refer again to Figure 3.22.2 which shows an equivalent orifice diagram for a typical compressor shaft seal. Notice that the atmospheric bushing downstream pressure is constant (atmospheric pressure). However, the gas side bushing pressure is referenced to the compressor process pressure. This pressure can and will vary during operation. If it were always constant, the requirement for differential pressure control would not be present in a seal system and it would be identical to a lubricating system. Another way of visualizing the system is to understand that the lube system utilizes differential pressure control as well, but the reference pressure (atmospheric pressure) is constant, and consequently all control valves need only control lube oil pressure. However seal systems require some means of constant differential pressure control (reference gas pressure to seal oil supply pressure). This objective can be accomplished in many different ways. Referring back to Figure 3.22.1 it can be seen that the supply system function is identical to that of a lube oil system with the exception that the liquid is referenced to a pressure that can vary, and must be controlled to maintain a constant differential between the referenced pressure and the seal system supply pressure. The sizing of the seal oil system components is also identical to that of the lube oil system components. Refer back and observe that the heat load and flow required of each seal is determined in a similar way to that of the bearings. Seals are tested at various speeds, and a necessary flow is determined to remove the heat of friction under various conditions. The seal oil flow requirements and corresponding heat loads are then tabulated, and pumps exchangers, filters, and control valves are sized accordingly.

The seal oil reservoir is sized exactly the same way as the lube oil reservoir in our previous example. The only major difference between the component sizing of a seal and a bearing is that the seal flows across the atmospheric bushing will change with differential pressures. As previously explained, any liquid compressor seal incorporates a double seal arrangement. The gas side seal differential is held constant by system design. The atmospheric side seal differential varies with varying seal reference (process) pressure. Therefore, the total flow to the seals will vary with process pressure and must be specified for maximum and minimum values when sizing seal system components. Remembering the concept of an equivalent orifice, a compressor at atmospheric conditions will require significantly less seal oil flow than it will at high pressure 1,380 kPa (200 psi) conditions. This is true since the differential across the atmospheric seal and liquid flow will increase from a low value to a significantly higher value, while the gas side bushing differential and liquid flow will remain constant provided seal clearances remain constant.

Many seal system problems have been related to insufficient seal oil flow through the atmospheric bushing at low suction pressure conditions. Close attention to the atmospheric drain cavity temperature is recommended during any off design (low suction pressure condition) operation.

The seal housing system

Regardless of the type, the purpose of any seal is to contain the fluid in the prescribed vessel (pump, compressor, turbine, etc.). Types and designs of seals vary widely. Figure 3.22.3 shows a typical mechanical seal used for a pump.

Since the contained fluid is a liquid, this seal utilizes that fluid to remove the frictional heat of the seal and vaporize the liquid, thus attaining a perceived perfect seal. A small amount of vaporized liquid constantly exits the pump across the seal face. It is a fact that all seals leak. This is the major reason that many pump applications today are required to utilize seal-less pumps

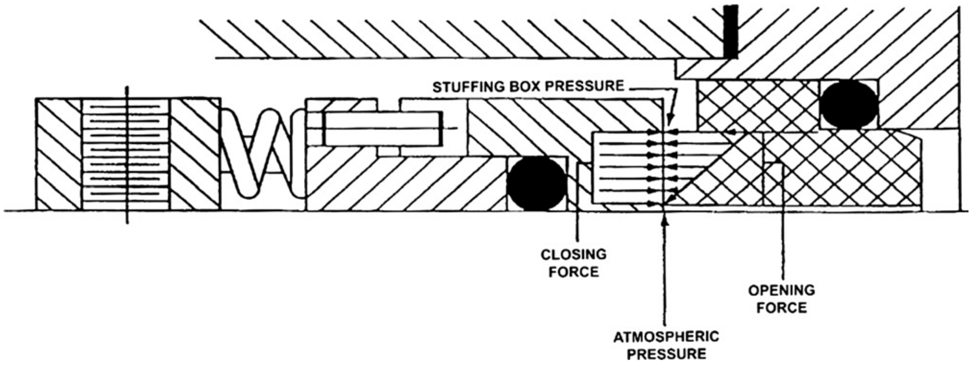


Fig 3.22.3 • Typical pump single mechanical seal

to prevent emission of toxic vapors. The following is a discussion of major types of seal combinations used in centrifugal compressor seal applications.

Gas seals

A typical gas seal is shown in Figure 3.22.4.

Gas seals have recently drawn attention since their supply systems appear to be much simpler than those of a traditional liquid seal system.

Since gas seals utilize the sealed gas or a clean buffer gas, a liquid seal system incorporating pumps, a reservoir and other components, is not required. However, one must remember that the sealing fluid still must be supplied at the proper flow rate, temperature and cleanliness. As a result, a highly efficient, reliable source of filtration, cooling, and supply must be furnished. If the system relies upon inert buffer gas for continued operation, the supply source of the buffer gas must be as reliable as the critical equipment itself. Gas seal configurations vary and will be discussed in detail in the next section. They can take the form of single, tandem (series), or multiple seal systems. The principle of operation is to maintain a fixed minimum clearance between the rotating and non-rotating face of the seal.

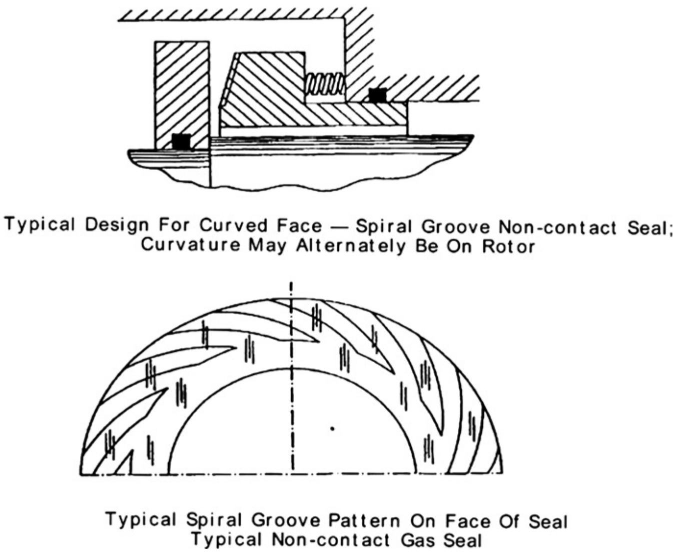


Fig 3.22.4 • Typical gas seal (Courtesy of John Crane Co.)

The seal employed is essentially a contact seal with some type of lifting device to maintain a fixed minimum clearance between the rotating faces. It is essential that the gas between these surfaces be clean since any debris will quickly clog areas and reduce the effectiveness of the lifting devices, consequently resulting in rapid damage to the seal faces.

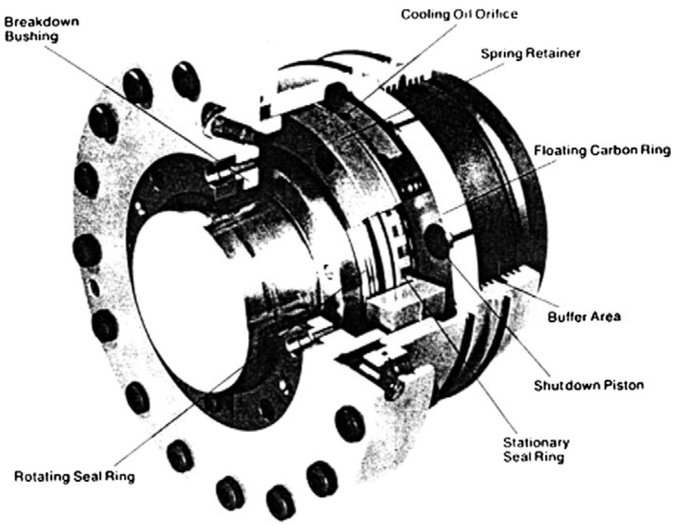
Liquid seals

Traditionally, the type of seal used in compressor service has been a liquid seal. Since the media that we are sealing against is a gas, a liquid must be introduced that will remove the frictional heat of the seal and ensure proper sealing. Therefore, all compressor liquid seals take the form of a double seal. That is, they are comprised of two seals with the sealing liquid introduced between the sealing faces. Refer to Figure 3.22.5. To ensure proper lubrication of both the gas side (inboard) and atmospheric side (outboard) seals, the equivalent 'orifices' of each seal must be properly designed, such that the differential pressure present provides sufficient flow through the seal to remove the heat of friction at the maximum operating speed. The type of gas side seal used in Figure 3.22.5 is a contact seal similar to that used in most pump applications. This seal provides a minimum of leakage (five to ten gallons per day per seal) and provides reliable operation (continuous operation for over three years.) As will be discussed below, the specific types of seals used in the double seal (liquid) configuration can vary.

Liquid bushing seals

A liquid bushing seal can be used for either a gas side or an atmospheric side seal application. Most seals utilize a liquid bushing seal for an atmospheric bushing application. A typical bushing seal is shown in Figure 3.22.6.

The principle of a bushing seal is that of an orifice. That is, a minimum clearance between the shaft and the bushing surface to minimize leakage. The bushing seal is designed such that the clearance is sufficient to remove all the frictional heat at the maximum power loss condition of that bushing with the available fluid differential across the bushing. It is important to realize that while acting as a seal, the bushing must not act as a bearing. That is, it must have degrees of freedom (float) to ensure that it does not support the load of the rotor. Since its configuration is similar to a bearing, if not allowed freedom of movement, it can act as an equipment bearing and result in



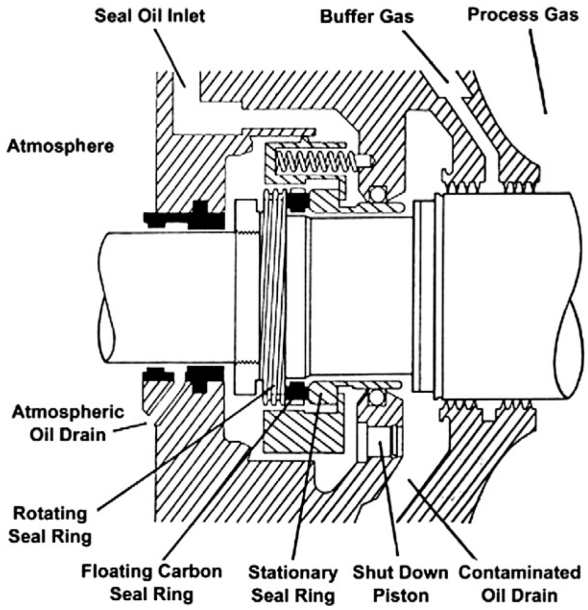


Fig 3.22.5 • ISO carbon seal (Courtesy of Elliott Co.)

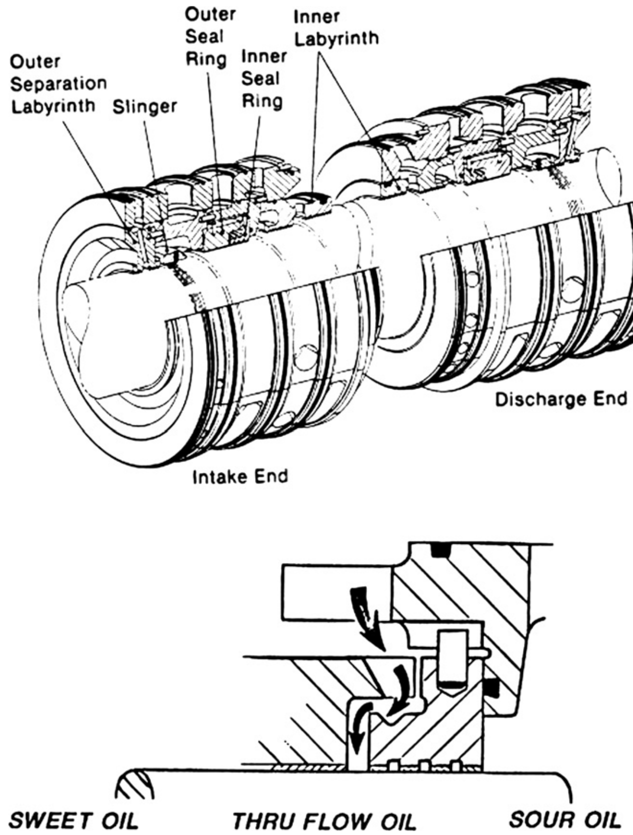


Fig 3.22.6 • Bushing seal – top: oil film seal: bottom: seal oil flow (Courtesy of Dresser-Rand)

a significant change to the dynamic characteristics of equipment, which has the potential to cause damage to the critical equipment. In order to achieve the objectives of a bushing seal, clearances are on the order of 0.0005" diametrical clearance per inch of shaft diameter.

Liquid bushing seals are also used for gas side seals, but their leakage rate will be significantly larger than that of a contact seal, since they are essentially an orifice. When used as a gas side bushing, therefore, the system must be designed to minimize the differential across the bushing. As a result, the differential control system utilized must be accurate enough to maintain the specified oil/gas differential under all operating conditions. The typical design differential across a gas side bushing seal is on the order of five to ten psid. The accurate control of this differential is usually maintained by a level control system.

Referring back to Figure 3.22.6, one can see that functioning of the bushing seal totally depends on maintaining a liquid interface between the seal and shaft surface. Failure to achieve this results in leakage of gas outward through the seal. It must be fully understood that all bushing seals must continuously maintain this liquid interface to ensure proper sealing. All systems incorporating gas side bushing seals must have the seal system in operation whenever pressurized gas is present inside the compressor case. If a liquid interface is not maintained, gas will migrate across the atmospheric bushing seal and proceed through the system returning back to the supply system. There have been cases in such system designs where failure to operate the seal system when the compressor is pressurized have resulted in effectively turning the gas side bushing into a filter for the entire process gas system! This resulted in the supply side of the seal oil system being filled with extensive debris, which required lengthy flushing and system cleaning operations prior to putting the unit back into service. Remember, any system incorporating a gas side bushing seal must be designed such that the entrance of process gas into the supply system is prohibited at all time. This can be accomplished by either:

- Continuous buffer gas supply
- A check valve installed as close as possible to seals in the seal oil supply header
- Rapid venting and isolation of the compressor case on seal system failure

In the second and third cases above, supply seal oil piping must be thoroughly checked for debris prior to re-start of the compressor. It is our experience that many bushing seal

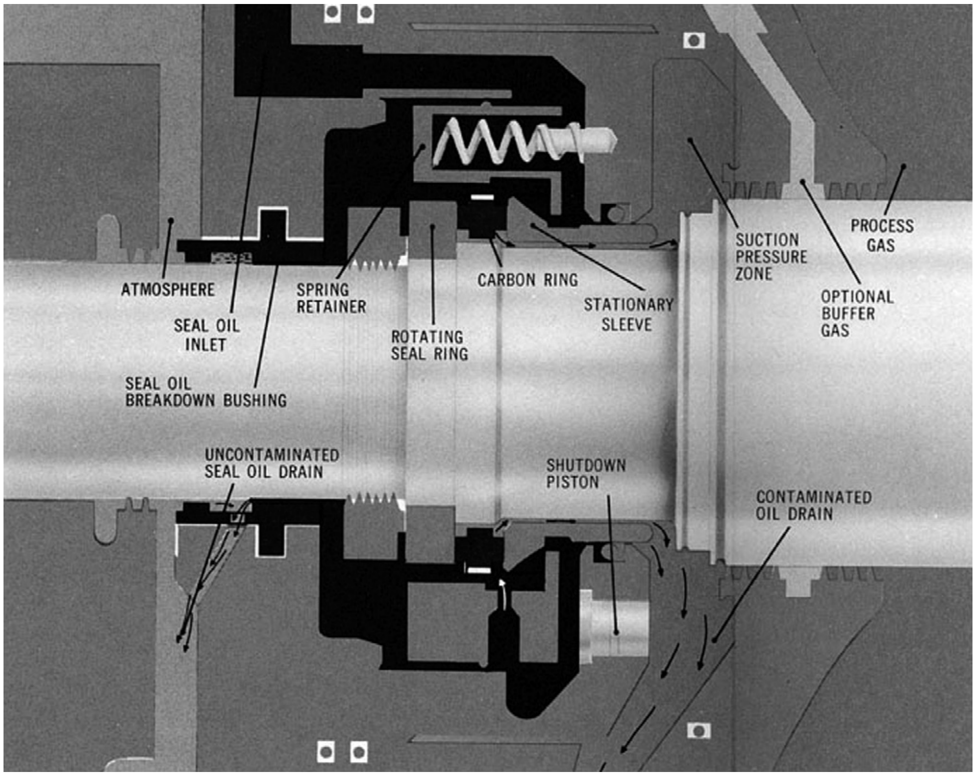


Fig 3.22.7 • Compressor contact seal (Courtesy of Elliott Co.)

system problems have resulted from improper attention to the above facts.

Contact seals

Figure 3.22.7 shows a typical compressor contact seal. As mentioned, these seals are similar in design to pump seals. In order to remove the heat of friction for this type of seal, a sufficient differential pressure above the reference gas must be maintained. Typical differentials for contact seals vary between 240 and 345 kPa (35 and 50 psi) differential pressure. Leakage rates with a properly installed seal can be maintained between 19 to 38 liters (5 to 10 gallons) per day per seal.

A limitation in the use of contact seals is the shaft speed; since the contact seal operates on a surface perpendicular to the axis of rotation, the rubbing speed of the seal surface is critical, and as a result, contact seals are speed-limited. Typical maximum speeds are approximately 12,000 revolutions per minute. Above those speeds, bushing seals are used, since the sealing surface is maintained at a lower diameter and correspondingly lower rubbing speed. The maximum limit of differential pressure across contact seals is controlled by the materials of construction, and is approximately 1,380 kPa (200 psi) differential. As a result, contact seals are usually used for gas side seal applications. They are very seldom utilized for atmospheric seal applications since they are differential pressure limited.

Since the differential pressure required across the seal face is relatively high compared to a bushing seal, contact seals utilize differential pressure control as opposed to level control for most bushing seals. This fact will be discussed in the next section.

Restricted bushing seals

The last type of seal to be discussed is a restricted bushing type seal. This type of seal is shown in Figure 3.22.8.

This particular type of restricted bushing seal utilizes a small pumping ring in the opposite direction of bushing liquid flow to compensate for the relatively large leakage experienced with bushing seals by introducing an opposing pumping flow in the opposite direction. Seals of this type can be designed for practically zero flow leakages. However, it must be pointed out that in variable speed applications, the pumping capability of the trapped seal ring must be calculated for both minimum and maximum speeds. Failure to do so can result in the actual pumping of gas from the compressor into the sealing system. It is recommended that such seals be designed to leak a small amount at maximum operating speed. Any retrofits of equipment employing this type of seal should be investigated when higher operating speeds are anticipated. A restricted bushing seal is used exclusively for gas side service.

In summary, the basic types of liquid seals used for compressor applications can be either; open bushing types, contact types, or restricted bushing types. Contact types are used primarily on the gas side. Liquid bushing types are used on either the gas or atmospheric side. Restricted bushing types are used exclusively on the gas side.

We will now investigate various seal system designs using various seal combinations employing the types of seals that have been discussed in this section.

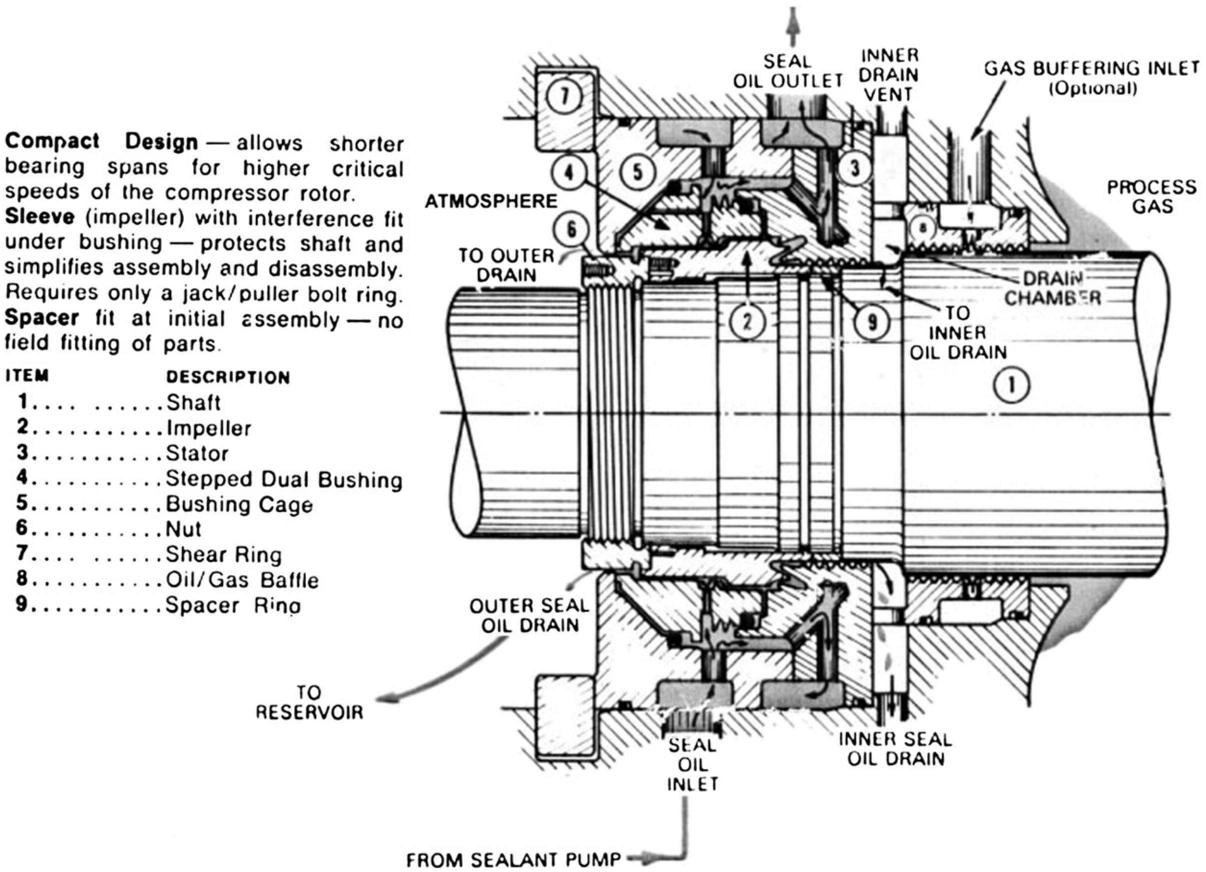


Fig 3.22.8 • Turbo-compressor ‘trapped bushing seal’ (Courtesy of A.C. Compressor Corp.)

Seal supply systems

As can be seen from the previous discussion, the type of seal system will depend on the type of seal utilized. We will now examine five different types of seal systems, each utilizing a different type of main compressor shaft seal system. As we proceed through each type, the function of each system will become clear.

Example 1: Contact type gas side seal – bushing type atmospheric side seal with cooling flow

This system incorporates a contact seal on the gas side and a bushing seal on the atmospheric side of each end of the compressor. The inlet pressure of the seal fluid on each end is referenced to the suction pressure of the compressor. It should be noted that some applications employ different reference pressures on each end of the compressor. The reference pressure should be taken off the balance drum end, or high pressure end of the compressor, to ensure that the oil to gas differential pressure is always at a minimum acceptable value. Therefore, the low pressure end may experience a slightly higher oil to gas differential than the reference end of the seal. Refer to Figure 3.22.9.

Proceeding through the seal, the seal oil supply, which is referenced to the gas reference pressure, enters the seal chamber. The differential across the gas side contact seal is maintained by a differential pressure control valve located downstream of the seal. Seal oil flows in three separate directions:

- Through the seal chamber (cooling flow)
- Through the gas side contact seal 38.5–77 liters/day (10–20 gallons/day)
- Through the atmospheric seal

Let’s examine the variants of flows across the equivalent orifice of each portion of this configuration.

The gas side contact seal will experience a constant flow, which for purposes of discussion, can be assumed to be zero gallons per minute (since the maximum flow rate will usually be on the order of ten gallons per day).

The atmospheric side bushing seal flow will vary based upon the referenced gas pressure. At low suction pressure conditions, this flow will be significantly less than it will be under high pressure conditions. The seal system design must consider the maximum reference pressure to be experienced in the compressor case to ensure that sufficient seal oil flow is available at maximum pressure conditions.

The seal chamber through flow in this seal design is used to remove any excess frictional heat of the seals and is regulated by the downstream control valve. As an example, let us assume the following values were calculated for this specific seal application.

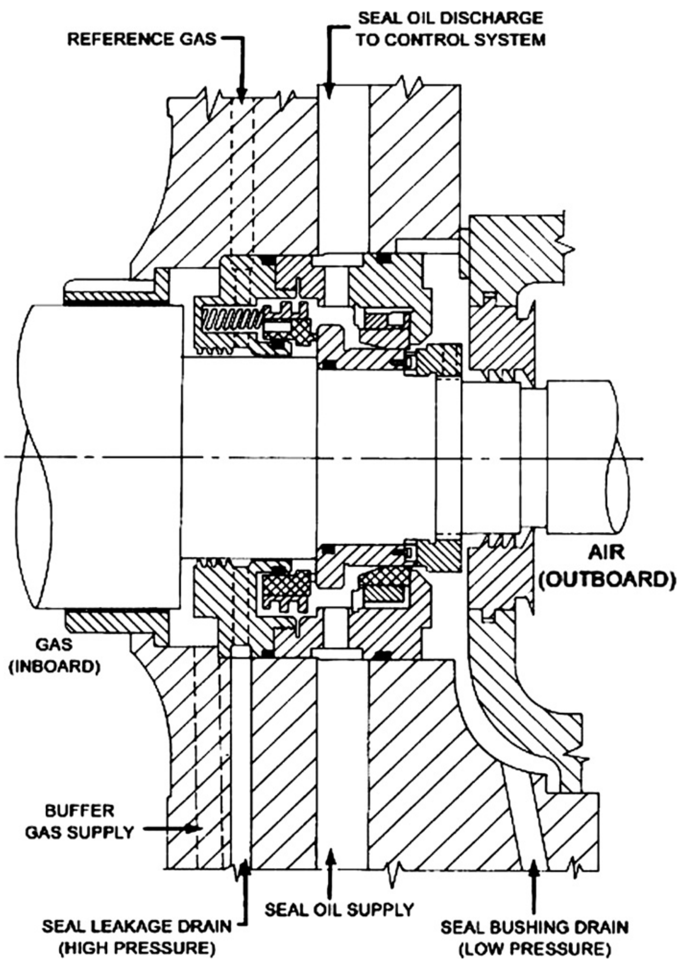


Fig 3.22.9 • Compressor shaft seal (Courtesy of IMO Industries)

1. Gas Side Seal Flow = 0
2. Atmospheric Side Seal Flow
   Reference Pressure = 0
   Seal Flow = 19.25 liters/min (5 GPM)
   Reference Pressure = 1,380 kPa (200 PSIG)
   Seal flow = 46 liters/min (12 GPM)
3. Flow Through Flow
   Minimum = 11.5 liters/min (3 GPM) – occurring at high ATM bushing flow = 46 liters/min (12 GPM)
   Maximum = 46 liters/min (12 GPM) – occurring at low ATM bushing flow = 11.5 liters/min (3 GPM)
4. *Seal Oil Supply Flow* in both cases = 57.8 liters/min (15 GPM)

As shown in the previous example, the required seal oil supply at maximum operating speed required to remove frictional heat is 57.8 liters/min (15 gal/min). At start-up, low suction pressure conditions, the control valve must open to allow an additional ten gallons per minute flow through to the seal chamber. At maximum operating pressure, however, the valve only passes a flow of three gallons per minute since 46 liters/min (12 gal/min) exit through the atmospheric bushing. This type of system is less sensitive to low suction pressure operation, since flow through oil will remove frictional heat around the atmospheric bushing.

Example 2: Contact type gas side seal – bushing type atmospheric seal with orificed through flow

The only difference between this type of system and the previous system is that the back pressure is maintained constant by a permanently installed through flow orifice. As a result, the differential pressure control valve is installed on the inlet side of the system. The process gas reference is still the same as before, that is, to the highest pressure side of the compressor. Figure 3.22.10 shows this type of system.

Let us examine the previous example case for this system and observe the differences.

1. Gas Side Seal Flow = 0
2. Atmospheric Side Seal Flow
   Reference Pressure = 0
   Seal Flow = 19.25 liters/min (5 GPM)
   Reference Pressure = 1,380 kPa (200 PSIG)
   Seal Flow = 46 liters/min (12 GPM)
3. Flow Through Flow (Orifice)
   Minimum = 2 liters/min (0.5 GPM)
   Maximum = 11.5 liters/min (3 GPM)

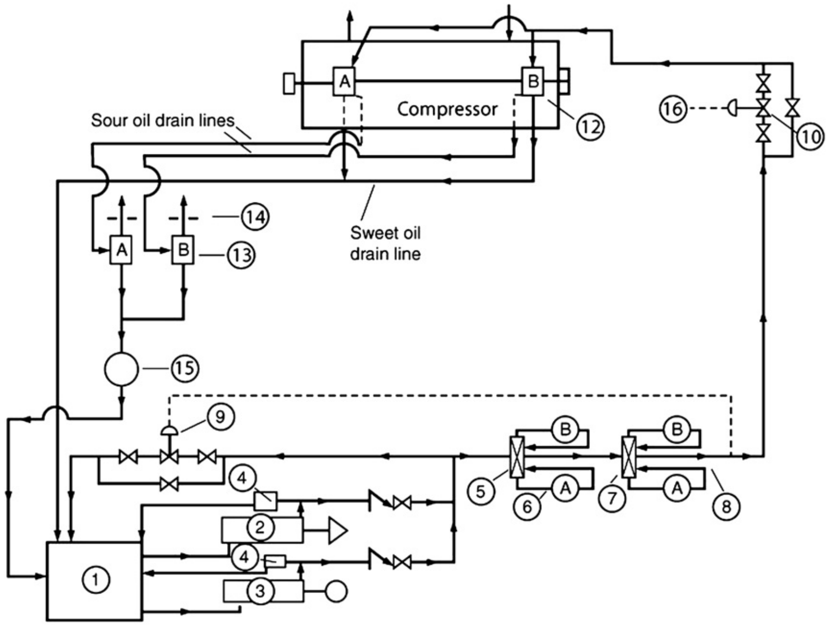
As can be seen, this system is more susceptible to high temperature atmospheric bushing conditions at low suction pressures, and must be observed during such operation to ensure integrity of the atmospheric bushing. In this system, the control valve will sense supply oil pressure to the seal chamber and control a constant set differential, approximately 241 kPad (35 psid), between the reference gas pressure and the supply pressure. If continued low pressure operation is anticipated with such a system, consideration should be given to a means of changing the minimum flow and maximum flow orifice for various operation points. Externally piped bypass orifices could be arranged such that a bypass line with a large orifice for minimum suction pressure conditions could be installed and opened during this operation. It is important to note, however, that the entire supply system must be designed for this flow condition and the control valve must be properly sized to ensure proper flow at this condition. In addition, the low pressure bypass line must be completely closed during normal high pressure operation.

Example 3: Bushing gas side seal – bushing atmospheric side seal with no flow through provision

Figure 3.22.11 shows this type of seal system. Here, the differential control valve becomes a level control valve sensing differential from the level in an overhead tank and is positioned upstream of the unit. Both bushings can be easily conceived as equivalent orifices. The gas side bushing flow will remain constant regardless of differential. The atmospheric bushing flow will vary according to seal chamber to atmospheric pressure differential.

The atmospheric bushing must therefore be designed to pass a minimum flow at minimum pressure conditions that will remove frictional heat and thus prevent overheating and damage to the seal. Since a gas side bushing seal is utilized, a minimum differential across this orifice must be continuously maintained.

Utilizing the concept of head, the control of differential pressure across the inner seal is maintained by a column of liquid.



1. Oil reservoir
2. Main screw pump - turbine driven
3. Aux screw pump - motor driven
4. Relief valves
5. Transfer valve
6. Oil coolers (A & B)
7. Transfer valve
8. Oil filters (A & B)
9. Back pressure control valve
10. Differential pressure control valve
11. Overhead seal oil tank (typically 15 ft above compressor centerline)
12. ISO sleeve seals (also called liquid bushing seals)
13. Seal oil drainers (A & B)
14. Drainer orifice vents (A & B)
15. Degassing tank
16. Reference gas line

Note: component condition instrumention and autostarts not shown

Fig 3.22.10 • API 614 Lube-seal oil system for ISO-carbon seals (Courtesy of Elliott Company)

As an example, if the required gas side seal differential of oil to gas is 34.5 kPad (5 psid), by the liquid head equation:

$$\text{Head} = \frac{0.102 \times 34.5}{0.85} = 4.1 \text{ meters}$$

$$(\text{Head} = \frac{2.311 \times 5 \text{ psid}}{0.85} = 13.6 \text{ ft.})$$

Therefore maintaining a liquid level of 4.1 meters (13.6 ft) above the seal while referencing process gas pressure will ensure a continuous 34.5 kPad (5 psid) gas side bushing differential. In this configuration, the control valve, which senses its signal from the level transmitter, will be sized to continuously supply the required flow to maintain a constant level in the overhead tack. As an example, consider the following system changes from start-up to normal operation.

In this example, a change from the start-up to operating condition will increase gas reference pressure on the liquid level in the overhead tank, and would tend to push the level downward. Any movement of the level in the tank will result in an increasing signal to the level control valve to open, thus increasing the pressure (assuming a positive displacement pump) to the overhead tank and reestablishing the pre-set level.

In the above example, at 1,380 kPa (200 psi) reference pressure, the bypass valve would close considerably. To increase the pressure supply of the seal oil from 34.5 to 1,413 kPa (5 to 205 psi), the difference of bypass flow through the valve, of 37 lpm (7 gpm), is equal to the increased flow through the

Table 3.22.1 Seal System Flow

Item	Start-up condition	Normal operation
Compressor suction pressure	0 kPa (PSIG)	1,380 kPa (200 PSIG)
Overhead tank reference pressure	0 kPa (PSIG)	1,380 kPa (200 PSIG)
Gas side seal bushing flow	0 LPM (GPM)	0 LPM (GPM)
Atmospheric side seal bushing flow	19.3 LPM (5 GPM)	46 LPM (12 GPM)
Seal pump flow	77 LPM (20 GPM)	77 LPM (20 GPM)
Bypass valve flow	57.8 LPM (15 GPM)	30.8 LPM (8 GPM)

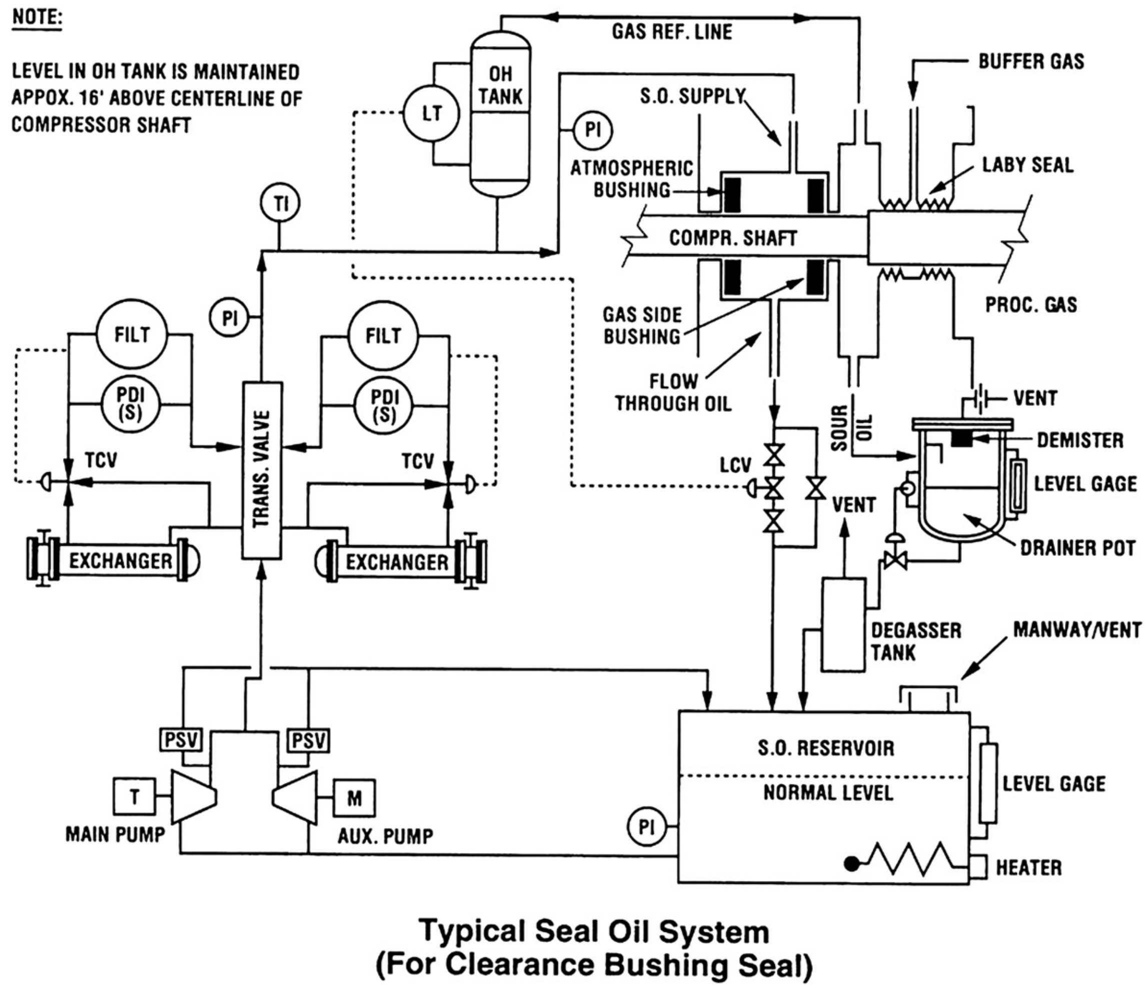


Fig 3.22.11 • Typical seal oil system (Courtesy of M.E. Crane, Consultant)

atmospheric bushing at this higher differential pressure condition. Utilizing the concept of equivalent orifices, it can be seen that the additional differential pressure across the atmospheric bushing orifice is compensated for by reducing the effective orifice area of the bypass control valve. This is accomplished by sensing the level in the head tank and maintaining it at a constant value by opening the seal oil supply valve.

As in the case of the orificed through flow example above, this configuration is susceptible to high atmospheric bushing temperatures at low suction pressures and must be monitored during this condition. Repeated high temperatures during low suction pressure conditions should give consideration to re-sizing of atmospheric bushing clearances during the next available turnaround. The original equipment manufacturer should be consulted to ensure correct bushing sizing and supply system capability.

Example 4: Gas side bushing seal – atmospheric side bushing seal with through flow design

Refer to Figure 3.22.11. The only difference between this and the previous system is that a through flow option is added, to allow sufficient flow through the system during changing pressure conditions. The bypass valve in the previous system is replaced in this system by a level control valve referenced from a head tank level transmitter, and is installed downstream of the seal chamber. This system functions in exactly the same way as the system in Example 1, the only difference being that a level control valve in this example replaces the differential control valve in that example. Both valves have the same function; that is, to control the differential in the seal chamber between the seal oil supply and the referenced gas pressure. A level control valve is utilized here, however, since a bushing seal requires a significantly lower differential between the seal oil supply pressure and the gas reference pressure.

Consider the following example. Assume that a differential control valve would be used as opposed to a level control valve for the system in Figure 3.22.11. For the start-up case, the differential control valve would have to maintain a differential of 34.5 kPa (5 psi) over the reference gas. When the reference gas pressure was 0, the oil upstream pressure to the valve would be approximately 34.5 kPa (5 psi). For the operating case, maintaining the same 34.5 kPa (5 psi) differential, the upstream pressure across the valve would be approximately 1,413 kPa (205 psi) instead of 34.5 kPa (5 psi). Consequently, the valve position would change significantly, but still would have to control the differential accurately to maintain 34.5 kPa (5 psi). Reduction of this pressure by any amount below 34.5 kPa (5 psi) could result in instantaneous bushing failure. However, if a level

control valve were installed, the accuracy of the valve would be measured in inches of oil instead of psi. Any level control system could control the level within two inches, which would be only 0.41 kPa (0.06 psid) variation in pressure differential!

This example shows that the most accurate way of controlling differential pressure for systems requiring control of small differential values is to use level instead of differential control. This system would be designed such that the combination of the atmospheric flow and the through flow through the seal would be equal to the flow from the pump.

Example 5: Trapped bushing gas side seal: atmospheric side bushing seal with flowthrough design

This system follows exactly the same design as that described in Example 4. The only difference would be in the amount of flow registered in the seal oil drainer. A trapped bushing system is designed to minimize seal oil drainer pot leakage. Typical values can be less than five gallons/day.

Seal supply system summary

All of the above examples have dealt with a system incorporating one seal assembly. It must be understood that most systems utilize two or more seal system assemblies. Typical multi-stage compressors contain two seal assemblies per compressor body, and many applications contain upwards of three compressor bodies in series, or six seal assemblies. Usually each compressor body is maintained at the suction pressure to that body, therefore three discreet seal pressure levels would be required and three differential pressure systems would be utilized. The concepts discussed in this section follow through regardless of the amount of seals in the system. Sometimes, the entire train, that is, all the seals referenced to the same pressure. In this case, one differential seal system could be used across all seals.

In conclusion, remembering the concept of an orifice will help in understanding the operation of these systems. Remember, the gas side bushing is essentially zero flow, the atmospheric side bushing flow varies with changing differential across the seal and any seal chamber through flow will change either as a result of differential across a fixed orifice or the repositioning of the control valve.

Seal liquid leakage system

This seal system sub-system's function is to collect all of the leakage from the gas side seal and return it to the seal reservoir at specified seal fluid conditions. Depending upon the gas condition in the case, this objective may or may not be possible. If the gas being compressed has a tendency to change the specification of the seal oil to off-specification conditions, one of two possibilities remain:

- *Introduce a clean buffer gas* between the seal to ensure proper oil conditions
- *Dispose of the seal oil leakage*

In most cases, the first alternative is utilized. Once the seal oil is in the drainer, a combination of oil and gas are present. A vent may be installed in the drainer pot to remove some of the gas, or a degassing tank can be incorporated.

This concludes the overview section of seal oil systems. As can be seen from the above discussion, it is evident that the design of a seal oil system follows closely to that of a lube system. The major difference is that the downstream reference pressure of the components (seal) varies, whereas in the case of a lube system it does not. In addition, the collection of the expensive seal oil is required in most cases and a downstream collector, or drainer system, must be utilized. Other than these two exceptions, the design of the seal system is very similar to that of a lube system and the same concepts apply in both cases.

Best Practice 3.23

Define all dry gas seal and system requirements in the specification and ensure they are complied with for seal MTBFs greater than 100 months.

Use site, company and industry dry gas seal and system best practices (see supporting material).

Attach the DGS best practices to the data sheet and include a P&ID of the required system.

Do not allow any exceptions, since the best practices required will result in optimum seal system safety and reliability for the life of the process unit.

Lessons Learned

In 2006, FAI experienced over 50 dry gas seal and system reliability issues by being involved with the Root Cause Failure Analysis. All issues were the result of the end user not properly specifying the system for the plant operation during the design phase.

In all cases, the failures were in critical un-spared services and resulted in a minimum of 3 days of downtime for each case.

Benchmarks

This best practice has been used since the mid-1990s in all projects and has resulted in dry gas seal MTBFs in excess of 100 months.

B.P. 3.23. Supporting Material

The reliability of critical equipment is dependent on the reliability of each component in every auxiliary system that is connected with it. How do we maximize critical equipment reliability? The easiest way is to eliminate the auxiliary systems. Imagine the opportunity to eliminate all of the components; pumps, filters, reservoirs, etc. and thereby increase reliability and, hopefully, the safety of the equipment. The gas seal, as used in compressor applications, affords the opportunity to achieve these objectives. However, the gas seal is still part of a system and the entire gas seal system must be properly specified, designed, maintained and operated to achieve the objectives of optimum safety and reliability of the critical equipment. In this chapter, the principles of gas seal design will be discussed and applied to various gas seal system types. In addition, best practices will be discussed for saturated gas systems as well as shutdown philosophies.

System function

The function of a gas seal system is naturally the same as a liquid seal system. The function of a fluid seal system, remembering that a fluid can be a liquid or a gas, is to continuously supply clean fluid to each specified seal interface point at the required differential pressure, temperature, and flow rate. Therefore, one would expect the design of a gas seal and a liquid seal to be very similar, which, in fact, they are. Then why are their systems so different?

Comparison of a liquid and gas sealing system

Figure 3.23.1 shows a liquid sealing system. Compare this system to Figure 3.23.2, which shows a gas seal system, if the same compressor were retrofitted for a gas seal. WOW – what a difference. Why are there such a small number of components for the gas seal system? As an aid, refer to Figure 3.23.3, which shows a typical pump liquid flush system as specified by the American Petroleum Institute. This system incorporates a liquid mechanical seal and utilizes pump discharge liquid as a flush for the seal. Refer now to Figure 3.23.2 and observe the similarities. It should be evident that a gas seal system is simpler in compressor applications than a liquid seal system merely because the gas seal utilizes the process fluid. This is exactly the same case for a pump. By using the process fluid, and not a liquid, one can eliminate the need to separate liquid from a gas, thereby totally eliminating the need for a liquid supply system and the need for a contaminated liquid (sour oil) drain system.

Referring back to Figure 3.23.1, we can see that the following major components are eliminated:

1. The seal oil reservoir
2. The pumping units
3. The exchangers
4. The temperature control valves
5. The overhead tank
6. The drain pot
7. The degassing tank
8. All control valves
9. A significant amount of instrumentation

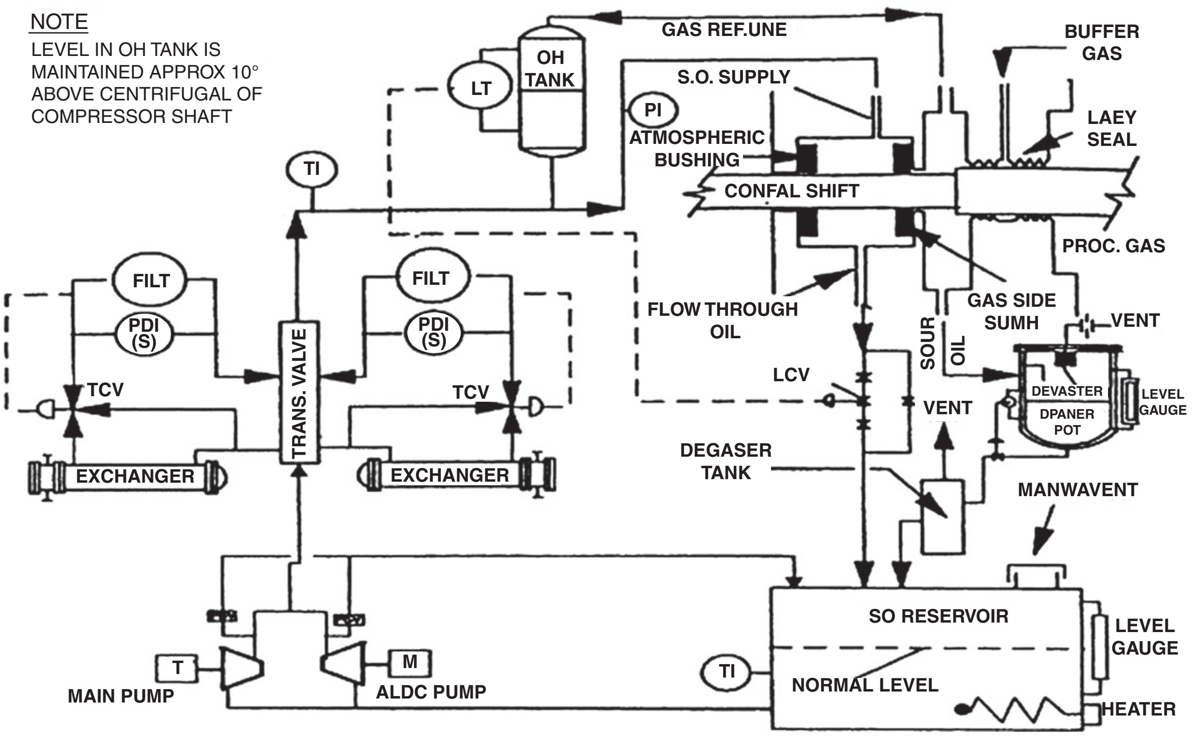


Fig 3.23.1 • Typical seal oil system for clearance bushing seal (Courtesy of M.E. Crane, Consultant)

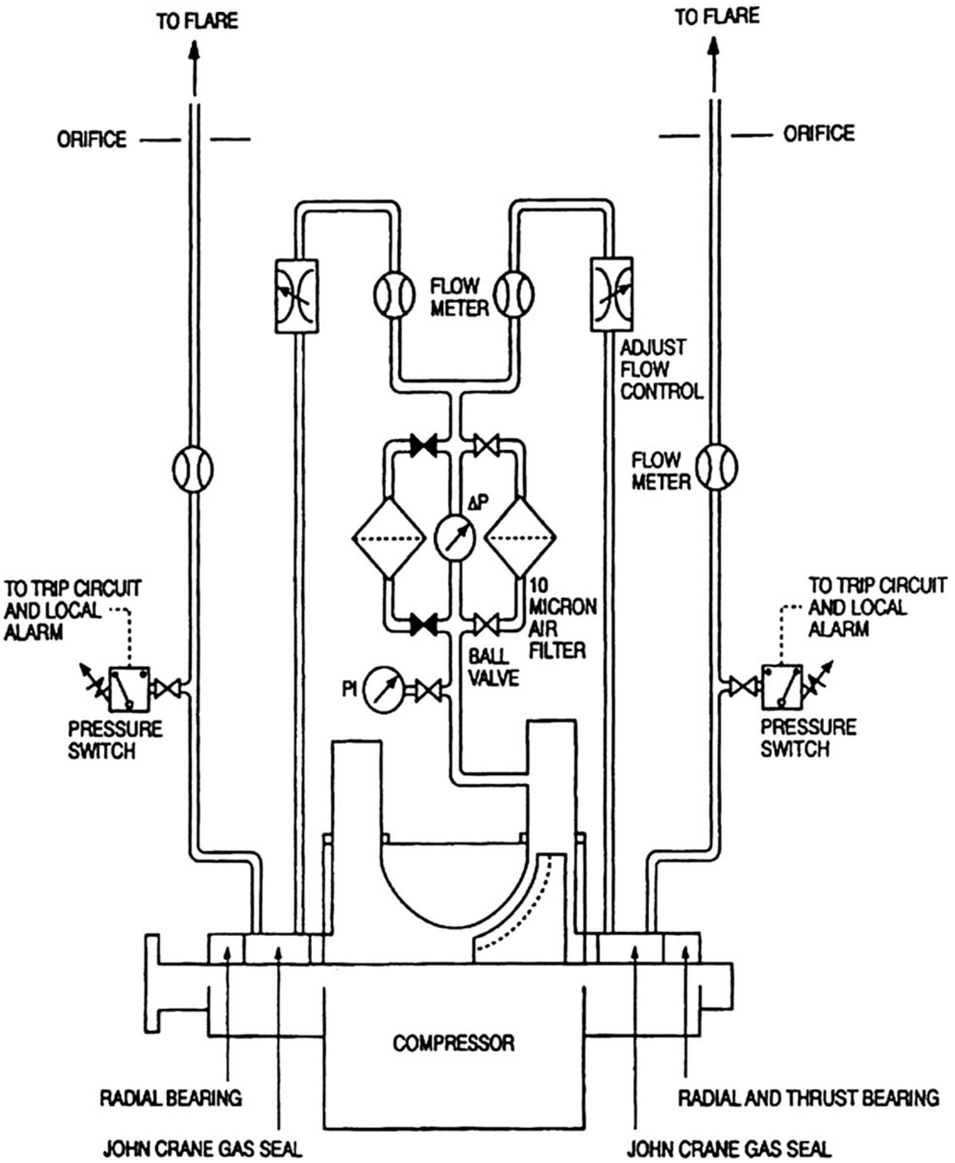


Fig 3.23.2 • Typical gas seal system for dry air and or inert gas (Courtesy of John Crane Co.)

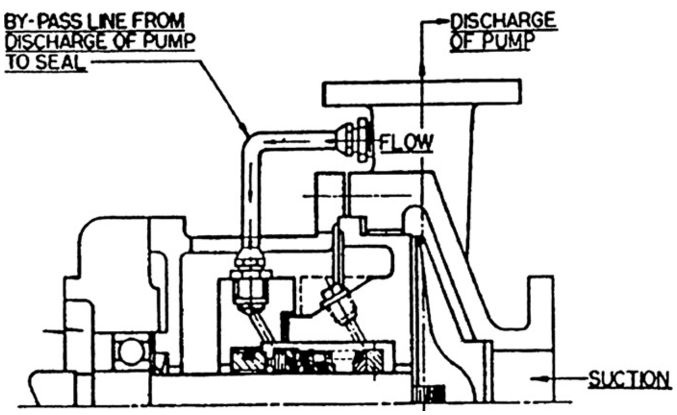


Fig 3.23.3 • Liquid seal flush (Courtesy of John Crane Co.)

Referring back to the function definition of the gas seal system, all requirements are met. 'Continuously supplying fluid' is met by utilizing the discharge pressure of the compressor. The requirements for 'specified differential pressure, temperature and flow rate' are met by the design of the seal itself, which can accommodate high differential pressures, high temperatures, and is sized to maintain a flow rate that will remove frictional heat necessary to maintain seal reliability. The only requirement not met is that of supplying a clean dry fluid, and this can be seen in Figure 3.23.2. This requirement is met by using a dual system coalescing filter.

When one considers all the advantages, the next question to ask is; "Okay, what are the disadvantages?" Naturally, there are disadvantages. However, proper design of the gas seal system can minimize and eliminate many of them. Do not forget that the requirements for any system mandate proper specification, design, manufacture, operation and maintenance. One can never eliminate these requirements in any critical equipment system.

Considerations for system design

As mentioned above, there are disadvantages to a gas seal system; which are not insurmountable but must be considered in the design of such a system. These considerations are as follows:

Sensitivity to dirt – since clearances between seal faces are usually less than 0.0005 inch and seal design is essential to proper operation, the fluid passing between the faces must be clean (5–10 microns maximum particle size). If it is not the small grooves (indentations) necessary for seal force separation will become plugged thus causing face contact and seal failure.

Sensitivity to saturated gas – saturated fluids increase the probability of groove (indentation) blockage.

Lift-off speed – as will be explained below, a minimum speed is required for operation. Care must be taken in variable speed operation to ensure that operation is always above this speed. It is recommended that the seal test be conducted for a period at turning gear speed to confirm proper 'lift off' followed by seal face inspection.

Positive prevention of toxic gas leaks to atmosphere – since all seals leak, the system must be designed to preclude the possibility of toxic of flammable gas leaks out of the system. This will be discussed in detail below.

Possible oil ingestion from the lube system – a suitable separation seal must be provided to eliminate the possibility of oil ingestion from the bearings. Whenever a gas seal system is utilized, the design of the critical equipment by definition incorporates a separate lube oil and seal system. Consideration must be given during the design or retrofit phases to the separation between the liquid (lube) and gas seal system.

'O' ring (secondary seal components) design and maintenance – most seal vendors state that 'O' ring life is limited and they should be changed every five years. This applies to both operating and spare seals. My experience has shown that dry gas 'O' ring seals can exceed this limit. It is recommended that seal vendors should provide references for similar applications prior to making a decision to change out the seals after five years.

If all of the above considerations are incorporated in the design of a gas seal system, its reliability has the potential to exceed that of a liquid seal system, and the operating costs can be reduced.

Before moving to the next section, however, one must consider that relative reliability of gas and liquid seal systems are a function of proper specification, design, etc. as mentioned previously. A properly designed liquid seal system, which is well operated and maintained, can achieve the reliabilities of a gas seal system. Also, when one considers the operating costs of the two systems, various factors must be considered. While the loss of costly seal oil is eliminated, with a gas seal system (assuming oil ingestion from the lube system does not occur) the loss of process gas, while minimal, can be expensive. It is argued that the loss of process gas from a liquid seal system through drainer vents and degassing tank vents is also significant. While this may be true in many cases, a properly specified, designed and operated liquid seal system can minimize process gas leakage such that it is equal or even less than that of a gas seal.

There is no question that gas seal systems contain far fewer components and are easier to maintain than liquid seal systems. These systems will be used extensively in the years ahead. The intention of this discussion is to point out that existing liquid seal systems that cannot be justified for retrofit, or cannot be retrofitted easily, can be modified to minimize outward gas leakage and optimize safety and reliability.

Dry gas seal design

Principles of operation

The intention of this sub-section is to present a brief detail of the principles of operation of a dry gas seal in a conceptual form. The reader is directed to any of the good literature available on this subject for a detailed review of gas seal design.

Refer to Figure 3.23.4, which shows a mechanical seal utilized for pump applications, and Figure 3.23.5, which shows a dry gas mechanical seal utilized for a compressor application. The seal designs appear to be almost identical. Close attention to Figure 3.23.5, however, will show reliefs of the rotating face of the seal. Considering that both seals operate on a fluid may give some hint as to why the designs are very similar. The objective of seal design is to positively minimize leakage while removing

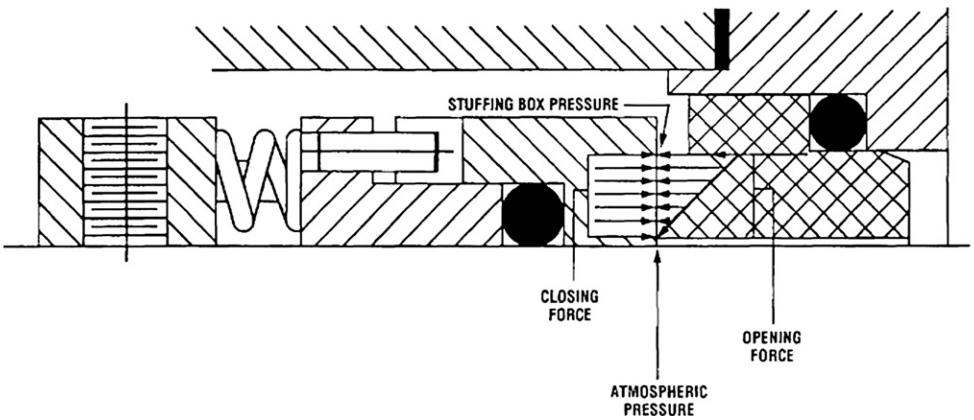


Fig 3.23.4 • Typical pump single mechanical seal

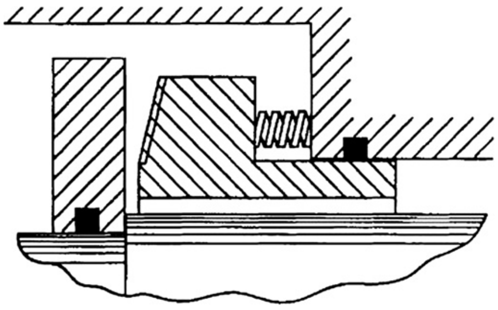


Fig 3.23.5 • Typical design for curved face – spiral groove non-contact seal; curvature may alternately be on rotor (Courtesy of John Crane Co.)

frictional heat, in order to obtain reliable, continuous operation of the seal. In a liquid application, the heat is removed by the fluid which passes between the rotating and stationary faces and the seal flush and changes from a liquid to gaseous state (heat of vaporization). This is precisely why all seals are said to leak, and explains the recent movement in the industry to seal-less pumps in toxic or flammable service. If the fluid between the rotating faces now becomes a gas, its capacity to absorb frictional heat is significantly less than that of a liquid. Therefore an 'equivalent orifice' must continuously exist between the faces to reduce friction, and allow a sufficient amount of fluid to pass and thus take away the heat. The problem obviously is how to create this 'equivalent orifice'. There are many different designs of gas seals. However, regardless of design, the dynamic action of the rotating face must create a dynamic opening force that will overcome the static closing forces acting on the seal to create an opening and hence 'equivalent orifice'.

Refer to Figure 3.23.6, which shows a typical gas dry seal face. Notice the spiral grooves in this picture; they are typically machined at a depth of 100–400 micro inches. When rotating, these vanes create a high head, low flow, impeller that pumps gas into the area between the stationary and the rotating face, thereby increasing the pressure between the faces. When this pressure is greater than the static pressure holding the faces together, the faces will separate, thus forming an equivalent orifice. In this specific seal design, the annulus below the vanes forms a tight face such that under static (stationary) conditions, zero leakage can be obtained if the seal is properly pressure-balanced. Refer to Figure 3.23.7 for a force diagram that shows how this operation occurs.

In Figure 3.23.6, the rotation of the face must be counter-clockwise to force the gas into the passages and create an opening (F_o) force. This design is known as a 'uni-directional' design and requires that the faces always operate in this direction. Alternative face designs are available that all rotation in either direction and they are known as 'bidirectional' designs.

Ranges of operation

Essentially, gas seals can be designed to operate at speeds and pressure differentials equal to or greater than those of liquid seals. Present state-of-the-art (2010) limits seal face differentials to approximately 17,250 kPa (2,500 psi) and rubbing speeds to approximately 122 meters/second (400 feet/second). Temperatures of operation can reach 538°C (1,000°F). Where seal face differential exceeds these values, seals can be used in series (tandem) to meet specifications provided sufficient axial space is available in the seal housing.

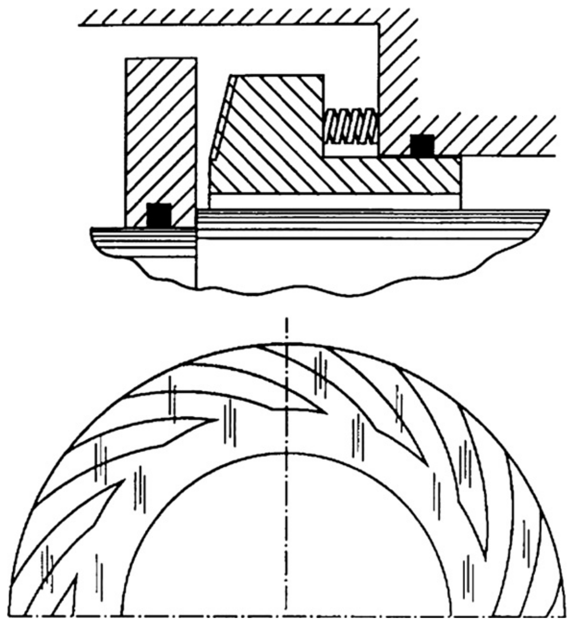


Fig 3.23.6 • Dry gas seal. Top: typical design for curved face – spiral groove non-contact seal; curvature may alternately be on rotor; Bottom: Typical spiral groove pattern on face of seal typical non-contact gas seal (Courtesy of John Crane Co.)

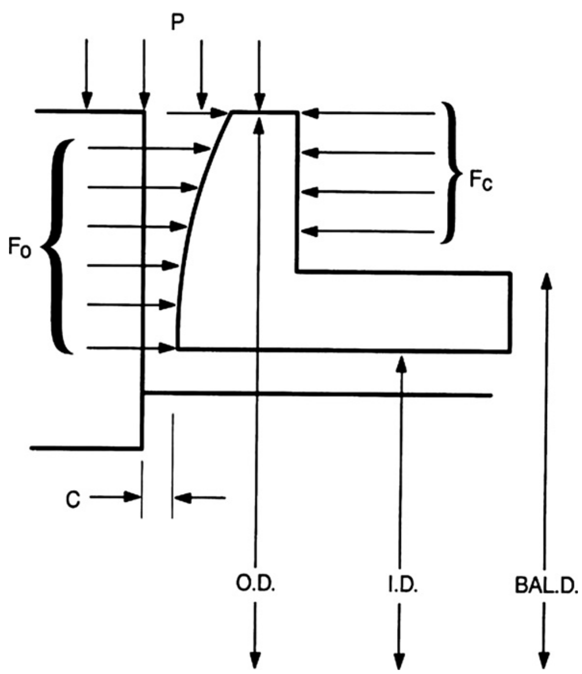


Fig 3.23.7 • Hydrostatic force balance on seal stator ($F_C = F_O$) (Courtesy of John Crane Co.)

Leakage rates

Since the gas seal when operating forms an equivalent orifice, whose differential is equal to the supply pressure minus the seal reference pressure, there will always be a certain amount of leakage. Refer to Figure 3.23.8 for leakage graphs.

It can be stated in general, that for most compressor applications with suction pressures on the order of 3,450 kPa (500 psi) and below, leakage can be maintained at the order of one standard cubic foot per minute per seal. For a high pressure application (17,750 kPa (2,500 psi)), differential leakage values can be as high as 8.5 Nm^3/hr (5 scfm) per seal. As in any seal design, the total leakage is equal to the leakage across the seal faces and any leakage across secondary seals ('O' rings, etc.). There have been reported incidences of explosive 'O' ring failure on rapid decompression of systems incorporating gas seals, thus resulting in excessive leakage. Consideration must be given to the system, in order to tailor system decompression times that meet the requirements of the secondary seals. As previously mentioned, all gas seals will leak, but not until the face 'lifts-off'. This speed known, oddly enough, as 'lift off speed' is usually less than 500 rpm. Caution must be exercised in variable speed applications, to ensure that the system prevents the operation of the variable speed driver below this minimum lift-off value. One recommendation concerning instrumentation is to provide one or two thermocouples in the stationary face of each seal to measure seal face temperature. This information is very valuable in determining lift-off speed and condition of the grooves in the rotating seal face. Any clogging of these grooves will result in a higher face temperature and will be a good indication of requirement for seal maintenance.

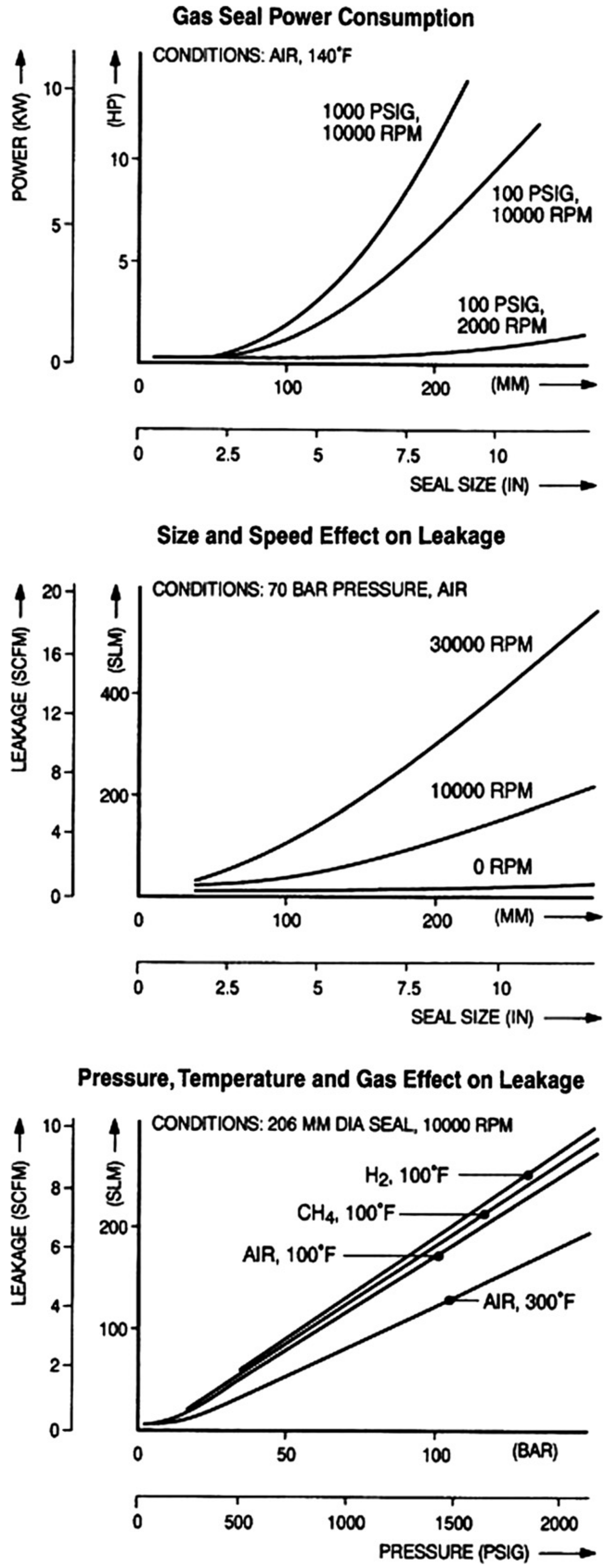


Fig 3.23.8 • Dry gas seal leakage rates (Courtesy of John Crane Co.)

Gas seal system types

As mentioned in this section, in order to ensure the safety and reliability of gas seals, the system must be properly specified and designed. Listed below are typical gas seal system applications in use today.

Low/medium pressure applications – air or inert gas

Figure 3.23.2 shows such a system, which is identical to that of a liquid pump flush system incorporating relatively clean fluid that meets the requirements of the seal in terms of temperature and pressure. This system takes the motive fluid from the discharge of the compressor through dual filters (ten microns or less) incorporating a differential pressure gauge and proportions equal flow through flow meters to each seal on the compressor. Compressors are usually pressure balanced such that the pressure on each end is approximately equal to the suction pressure of the compressor. The clean gas then enters the seal chamber and has two main paths:

A. Through the internal labyrinth back to the compressor. Note that the majority of supplied gas takes this path for cooling purposes (99%).

B. Across the seal face and back to either the suction of the compressor or to vent.

Since the gas in this application is inert, it can be vented directly to the atmosphere, or can be put back to the compressor suction. It must be noted, however, that this port is next to the journal bearing. Therefore, a means of positively preventing

entry of lube oil into this port must be provided, in order to prevent the loss of lube oil, or prevent the ingestion of lube oil into the compressor if this line is referenced back to the compressor suction. A suitable design must be incorporated for this bushing. Typically called a disaster bushing, it serves a dual purpose of isolating the lube system from the seal system and providing a means to minimize leakage of process fluid into the lube system in the event of a gas seal failure. In this system, a pressure switch upstream of an orifice in a vent line is used as an alarm and a shutdown to monitor flow. This switch uses the concept of an equivalent vessel, in that increased seal leakage will increase the rate of supply versus demand flow in the equivalent vessel (pipe) and result in a higher pressure. When a high flow is reached, the orifice and pressure switch setting are thus sized and selected to alarm and shutdown the unit if necessary. As in any system, close attention to changes in operating parameters are required. Flow meters must be properly sized and maintained clean such that relative changes in the flows can be detected in order to adequately plan for seal maintenance.

High pressure applications

In this application, for pressures in excess of 6,895 kPa (1,000 psi), a tandem seal arrangement or series seal arrangement is usually used. Since failure of the inner seal would cause significant upset of the seal system, and large amounts of gas escaping to the atmosphere, a backup seal is employed. Refer to Figure 3.23.9, which shows a triple gas tandem seal. For present designs up to 17,250 kPa (2500 psi), double tandem seals are proven and used.

The arrangement is essentially the same as low/medium pressure applications, except that a backup seal is used in place of the disaster bushing. Most designs still incorporate a disaster bushing between the backup seal and the bearing cavity known these days as the barrier seal. Attention in this design must be given to control of the inter-stage pressure between the primary and backup seal. Experience has shown that low differentials across the backup seal can significantly decrease its life. As in the case of liquid seals, a minimum pressure in the cavity between the seals of 172–207 kPa (25–30 psi) is usually specified. This is achieved by properly sizing the orifice in the vent or reference line back to the suction to ensure this pressure is maintained. All instrumentation and filtration are identical to that of the previous system.

Dual seal and system options for toxic and/or flammable gas applications

There are many field proven options available today for use in toxic and/or flammable gas applications. In this section we will discuss the following systems:

- Tandem seals for dry gas applications
- Tandem seals for saturated gas applications
- Tandem seals with interstage labyrinth and nitrogen separation gas
- Double seal system for dry gas or saturated gas applications

Tandem seals for dry gas applications

The tandem seal arrangement for this application is shown in Figure 3.23.10, and a schematic of this seal in the compressor seal housing is shown in Figure 3.23.11. Gas from the compressor discharge enters the port that is closest to the compressor (labyrinth end), and the majority of the gas enters the compressor through this labyrinth. To ensure that process gas, which is not treated by the dry gas system, does not enter the seal chamber, velocities across the labyrinth should be maintained between 6–15 m/sec (20–50 ft/sec). Taking labyrinth

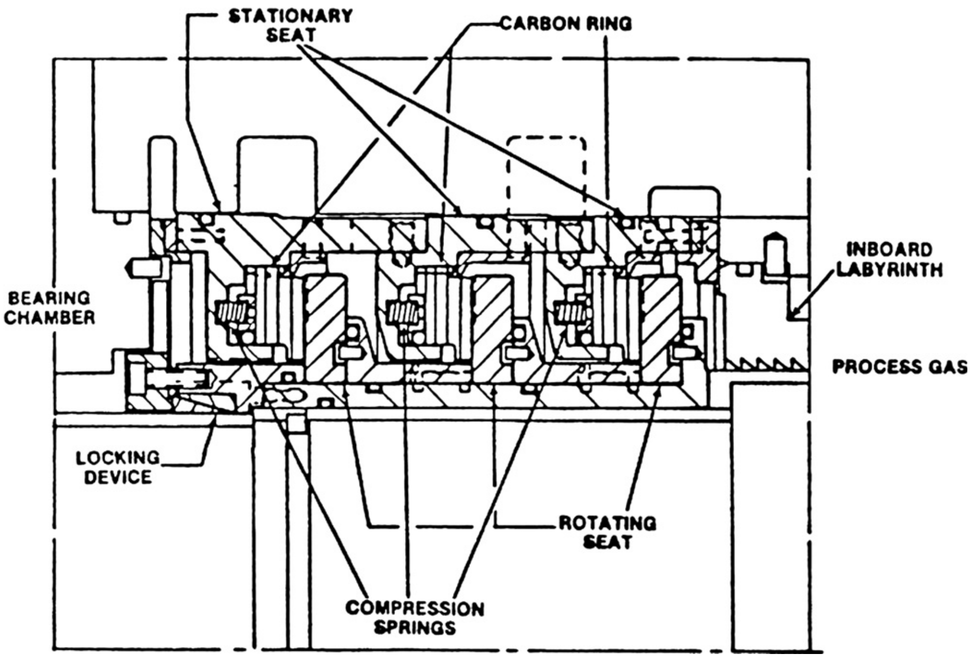


Fig 3.23.9 • Dry gas seal: tandem dry gas seal arrangement (Courtesy of Dresser-Rand Corp.)

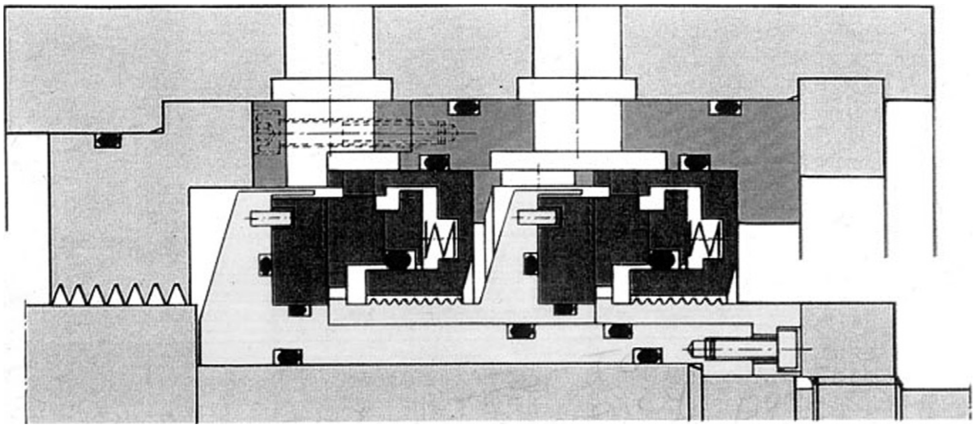


Fig 3.23.10 • Tandem seal (Courtesy of Flowserve Corp.)

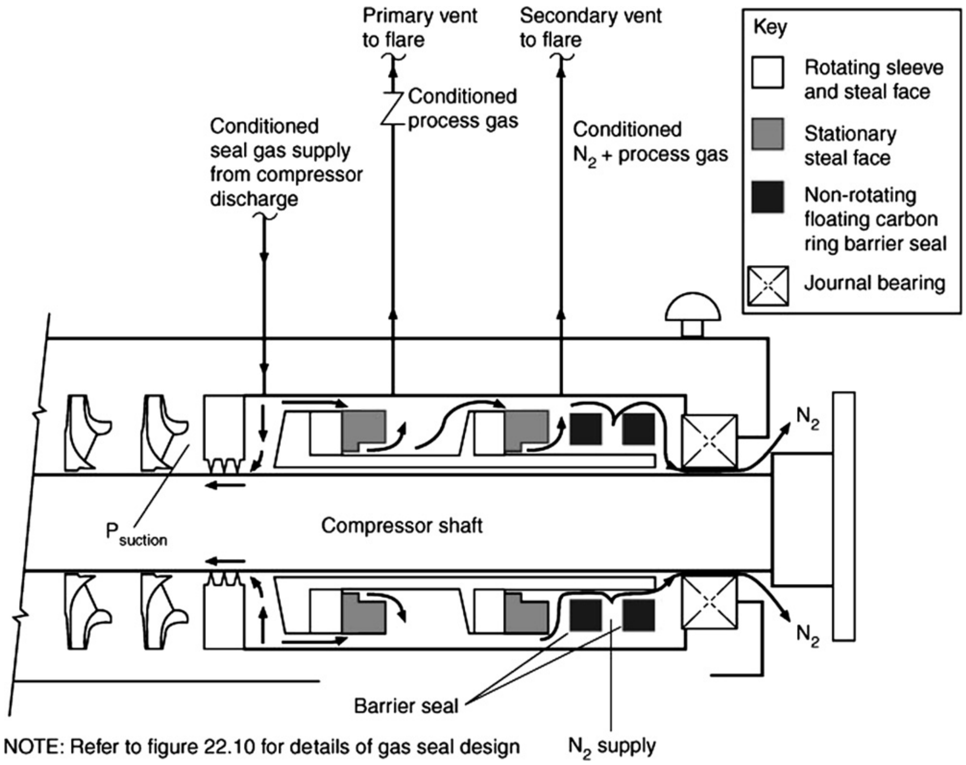


Fig 3.23.11 • Tandem seal and barrier seal typical housing arrangement

wear into consideration, the design should be closer to 15m/sec (50 ft/sec). Approximately 1.7–3.4 Nm^3/hr (1–2 scfm) flow (standard cubic feet per minute) leak across the first tandem seal faces (∗primary seal) and exit through the primary vent. Based on the backpressure of the primary vent system, 1.7 Nm^3/hr (1 scfm) or less will pass through the second tandem seal faces (secondary seal) and exit through the secondary vent. To ensure that oil mist from the bearing housing does not enter the dry gas seal chamber and that seal gas does not escape to atmosphere, an additional barrier seal is used and provided with pressurized nitrogen at approximately 35 kPa (5 psi).

A typical seal system for this arrangement is shown in Figure 3.23.12. As previously mentioned, dry gas seal reliability depends on the condition of the gas entering the seal faces. The function of the seal gas supply system for any dry gas seal option is to continuously supply clean, dry gas to the seal faces. During start-up, when the compressor is not operating with sufficient pressure to supply the seals, an alternate source of gas or a gas pressure booster system should be provided. These items are shown in Figure 3.23.12 and are typical for any type of dry gas seal application. Note that the following options exist regarding the primary, secondary vent and barrier seal instrumentation and components:

- Primary seal vent triple redundant (two-out-of-three voting) flow or differential pressure alarm and shutdown.

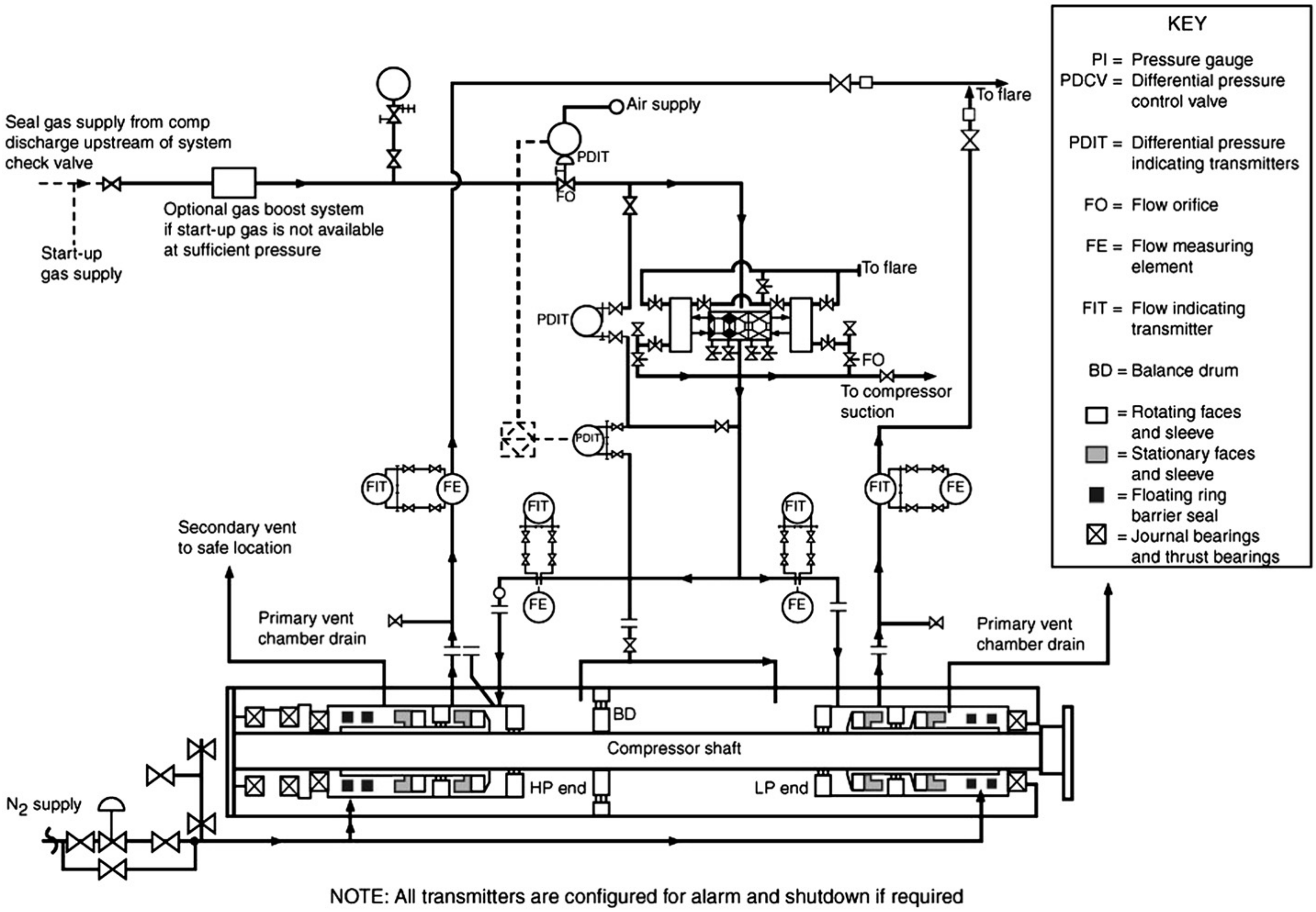


Fig 3.23.12 • Typical tandem seal system for dry process gas

- Primary seal vent rupture discs in parallel with vent line to rupture at a set pressure and prevent excessive pressure to the secondary seal on primary seal failure.
- Spring loaded exercise valves in the primary vent line to exert a backpressure on the primary seal to close the faces in the event of dynamic 'O' ring hang-up.
- Secondary vent line flow or differential pressure alarms and trips.
- Barrier seal supply pressure alarm and permission not to start the lube oil system until barrier seal minimum pressure is established.

Tandem seals for saturated gas applications

The tandem seal arrangement for this application can be exactly the same as that shown in Figures 3.23.10 and 3.23.11 for the dry gas application. The changes required for a saturated gas are solely in the seal system. A typical system is shown in Figure 3.23.13 and incorporates a cooler, separator and heater in addition to the normal components used for a dry gas application to ensure that saturated gas does not enter the seal chamber. Typical values for the cooler are to reduce the gas temperature to 30°F below its saturation temperature. The typical dimensions for the separator vessel, complete with a demister, are 460 mm (18 inches) diameter and 1.8 meters (6 feet) high. The typical requirements for the heater are to reheat the gas to 30°F above its saturation temperature. Temperature transmitters are provided upstream and downstream of the cooler, and downstream of the heater. As a precaution, in the event of cooler or heater malfunction, a dual filter/coalescer, complete with a drain back to the suction, is provided.

Tandem seals with interstage labyrinth

The present (2010) industry 'best practice' tandem seal arrangement for dry or saturated gas applications is shown in Figure 3.23.14. This arrangement features a labyrinth between the primary and secondary seals. This action ensures that gas vented from the secondary seal will always be nitrogen since the nitrogen supplied between the primary and secondary seals is differential pressure controlled to always be at a higher pressure than the primary seal vent, thus assuring that only nitrogen will be in the chamber between the primary and secondary seals. Figure 3.23.15 shows a typical nitrogen supply system used with this tandem seal configuration.

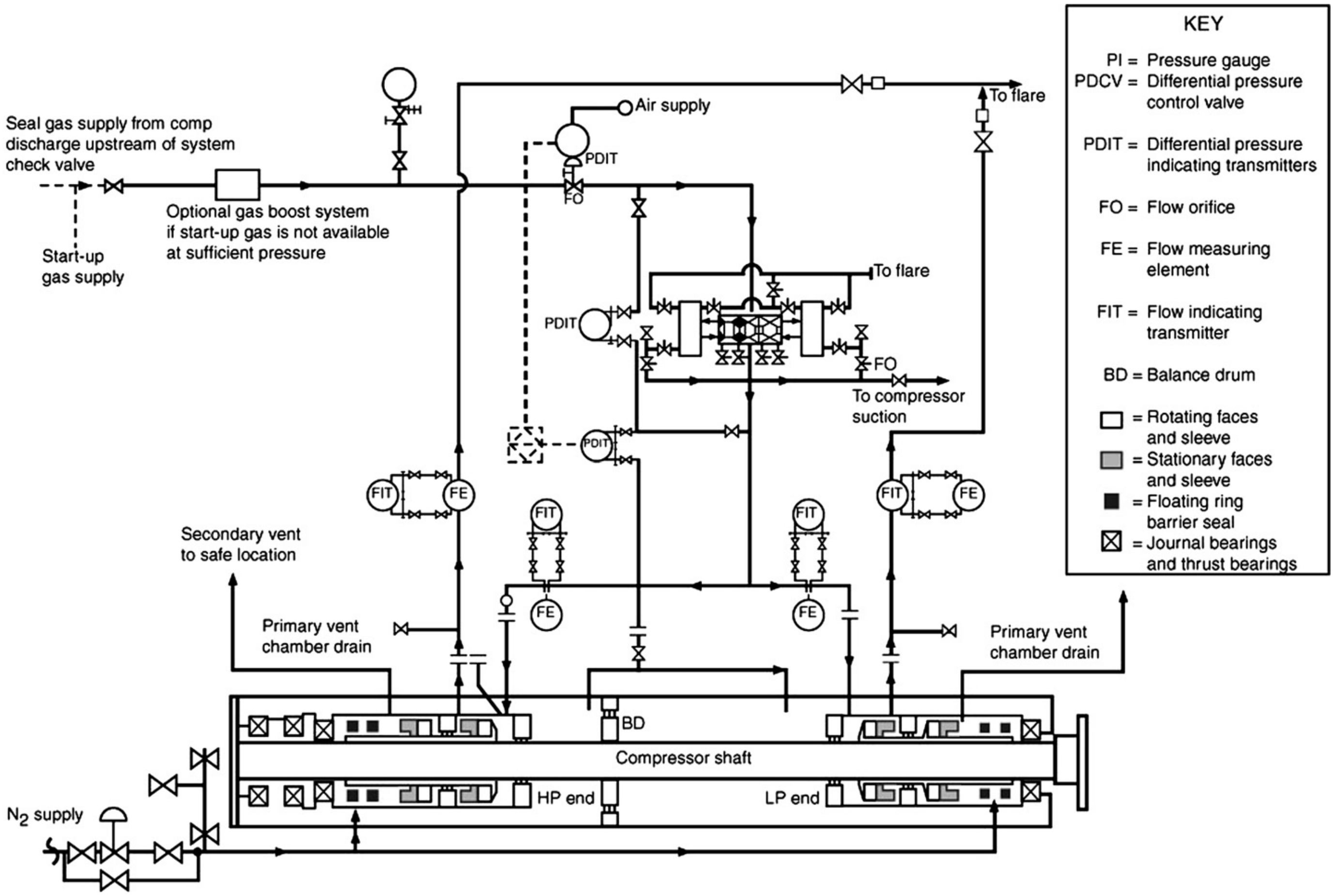


Fig 3.23.13 • Typical tandem seal system for saturated process gas

Double seal system for dry gas or saturated gas application

Figure 3.23.16 depicts a double seal, which is used in either dry gas or saturated gas applications where the process gas is not permitted to exit the compressor case. For this application process gas can be used, after it is conditioned, or an external source can be used if it is compatible with the process gas. If the gas used between the seals is toxic or flammable, a suitable barrier seal, provided with nitrogen, as shown in Figure 3.23.10, must be used. The seal systems previously shown will be used for the supply of conditioned gas to the seals as required by the condition of the seal gas (dry or saturated).

Summary

Since there are significant advantages to the use of dry gas seals, many units are being retrofitted in the field to incorporate this system. In many cases, significant payoffs can be realized.

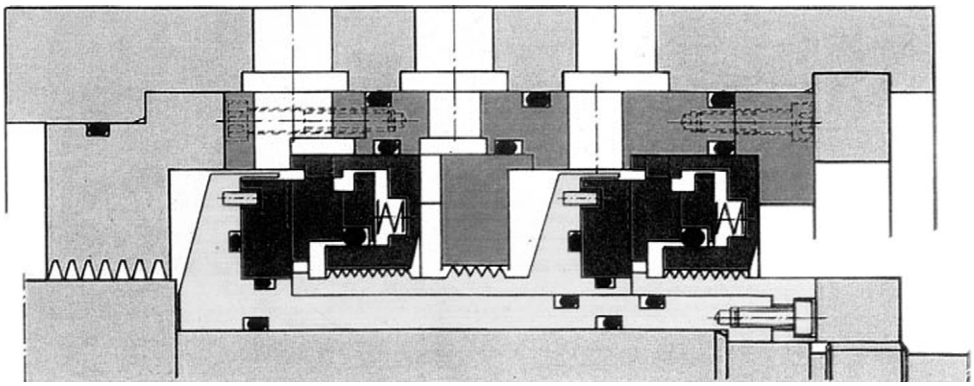


Fig 3.23.14 • Tandem seal with interstage labyrinth (Courtesy of Flowserve Corp.)

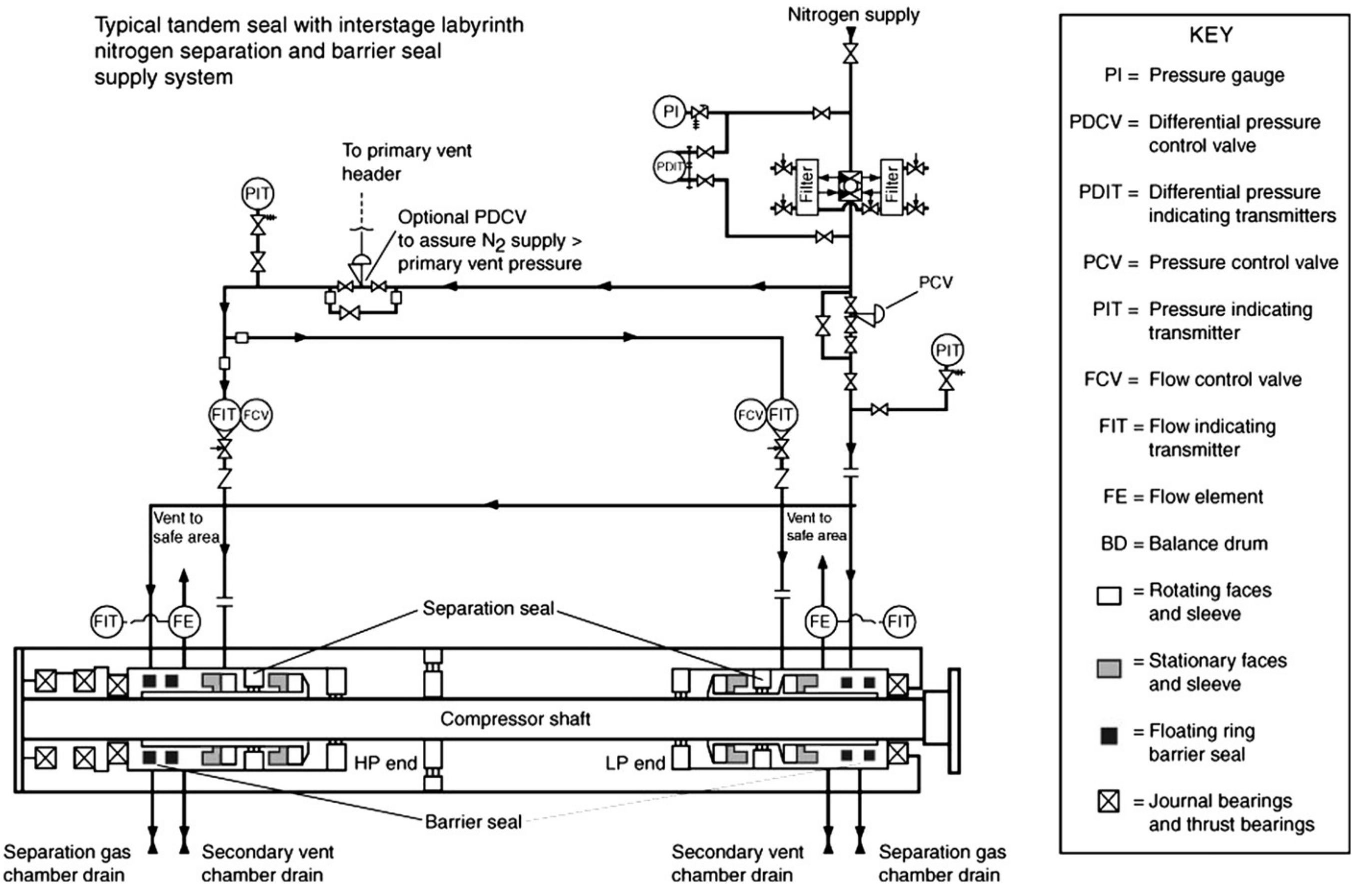


Fig 3.23.15 • Typical tandem seal system with an interstage labyrinth-nitrogen supply

If a unit is to be retrofitted, it is strongly recommended that the design of the gas seal be thoroughly audited to ensure safety and reliability. As mentioned in this section, retrofitting from a liquid to a gas seal system renders the unit a separate system type unit, that is, a separate lube and gas seal system. Naturally, loss of lube oil into the seal system will result in significant costs and could result in seal damage or failure by accumulating debris between the seal rotating and the stationary faces. The adequate design of the separation barriers between the lube and seal face must be thoroughly examined and audited to ensure reliable and safe operation of this system. Many unscheduled field shutdowns and safety situations have resulted

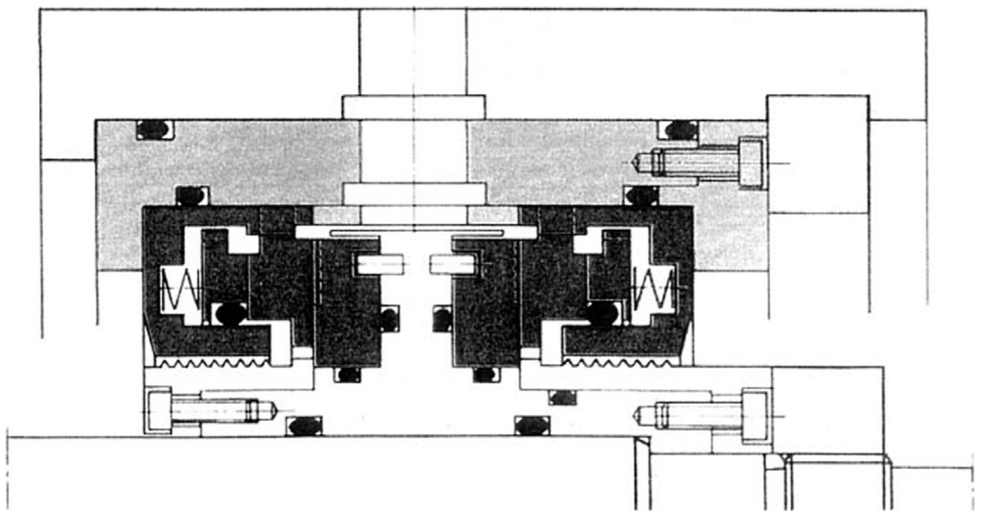


Fig 3.23.16 • Double seal (Courtesy of Flowserve Corp.)

from the improper design of the lube system, seal system separation labyrinth. In addition to the above considerations, a critical speed analysis, rotor response and stability analysis (if the operating discharge pressure is above 3,450 kPa (500 psi)) should always be conducted when retrofitting from liquid to dry gas seals.

Best Practice 3.24

For operating pressures above 13,800 kPa (2,000 psi), require full pressure and full load centrifugal compressor shop testing only if the vendor does not have 2 years' operating field experience on similar gases at the pressure levels specified.

At high field operating pressures >13,800 kPa (2,000 psi), gas disturbances can result in serious operational problems and prevent continued operation.

Since high pressure gas densities will cause these issues, be sure to specify a full pressure and load shop test and not only a full load test which may not match field pressure levels, if the vendor cannot show field operating experience at the specified pressure levels.

Lessons Learned

Correcting a gas disturbance problem in the field can result in long operational delays (weeks), losses of millions of dollars of revenue and may require multiple field modifications.

Benchmarks

This best practice has been used since 1990. Very few full pressure and full load tests are necessary today if vendors are selected who have high pressure field operating experience.

B.P. 3.24. Supporting Material

If a full pressure and load test is required, justify the full pressure test on the basis of a minimum of two weeks of lost production in the first year of plant operation.

Testing phase

The testing phase is the last phase in terms of vendor and subsupplier design and manufacturing involvement in the project; and it is the last chance to ensure the optimum availability of the finished product.

Remember that all of the equipment addressed in this section is most likely custom designed, and no matter how much accrued design and manufacturing experience is present, the possibility of some abnormality, hopefully minor, is high. Therefore, it is imperative that this phase be carefully observed and witnessed by the end user team. Figure 3.24.1 lists important facts surrounding this phase of the project.

The shop test is an opportunity to:

- Confirm vendor proper design and manufacture
- To match field conditions
- To witness assembly and disassembly using job special tools
- To have plant personnel observe test, assembly, etc. and take pictures for purposes of emergency field maintenance excellence
- Review the instruction book
- Have the assigned vendor service engineer observe the equipment he will install
- Review all vendor field procedures

Fig 3.24.1 • The shop test

A shop test checklist is included at the end of this B.P. that will be valuable for planning and executing the shop test phase. Yes — there certainly are many opportunities to ensure equipment reliability during shop test, but there are also a lot of potential lost opportunities if they are not justified to the project team early, during the pre-FEED phase of the project. The potential lost shop test phase opportunities are noted in Figure 3.24.2.

The following opportunities will be lost if they are not justified at project inception:

- Possible full load test
- Unproven component tests
- Attendance at test by plant personnel
- Use of special tools
- Vendor permission for pictures
- Agreement that assigned vendor field service specialists will be present for tests
- Agreement that the instruction book is reviewed
- Agreement for formal field construction meeting to clearly define all vendor procedures from receipt of equipment on site to initial run in of equipment

Fig 3.24.2 • Potential lost test opportunities

The success of the shop test depends on a good test plan that is reviewed by the end user and contractor and modified as requested, well in advance of the test. Figure 3.24.3 presents these facts.

- The agenda is issued for review 2 months prior to test
- It incorporates agreed VCM scope
- Compressor performance test conditions are per ASME PTC-10 requirements
- A sample of test calculations and report format is included
- Vendor concurs with all end user and contractor comments prior to test

Fig 3.24.3 • Shop test agenda review – key facts

I began my career in rotating equipment on the test floor, and I can still remember how we would see the witnesses come in with an intent to completely participate in the entire test only to leave for a long 'test lunch' an hour or so later. Why did this occur? Usually because the concerned end user and contractor witnesses did not have the opportunity to review the test set-up and the procedure prior to the test. As a result, I have always been a proponent of a pre-test meeting.

Is it always required? I think it is, but the detail and timing of the meeting depends on certain factors, noted in Figure 3.24.4.

- If the equipment is prototype
- If the equipment is complex
- If a full load test is required
- If the test facility is new

Fig 3.24.4 • When is a pre-test meeting required?

If it is decided to conduct a pre-test meeting, the key facts are noted in Figure 3.24.5.

- Conduct the meeting prior to the test day
- Send the agenda to the vendor well in advance
- A typical agenda outline:
  - Confirm test agenda requirements
  - Confirm all test parameter acceptance limits
  - Confirm instrument calibration
  - Review test set-up or concept drawing
  - Review data reduction methods
  - Confirm all test program agreements

Fig 3.24.5 • Pre-test meeting – key facts

Figures 3.24.6 to 3.24.8 define recommended test activity for the mechanical, auxiliary equipment and performance shop tests respectively.

- Per API and project requirements
- Confirm all components are installed
- Confirm all accessories are installed
- Monitor progress of test, look for leaks, etc.
- Do not accept test until all requirements are met

Fig 3.24.6 • Mechanical test – key facts

- Must be per API and project requirements
- Confirm that the test agenda is followed
- Confirm all components are installed
- Confirm that all required instruments are installed
- Monitor the progress of the test – look for leaks, etc
- Do not accept until all requirements are met

Fig 3.24.7 • Auxiliary system test – key facts

- Per ASME PTC-10 requirements
- Reconfirm test speed is per PTC-10
- Confirm all instruments are calibrated and installed
- Confirm test gas purity
- Agree that conditions are stable prior to each test point
- Confirm vendor's calculations for each test point
- Do not accept until all test requirements are met

Fig 3.24.8 • Compressor performance test – key facts

At the conclusion of all test activities, there is still important work to be performed. These items are defined in Figure 3.24.9.

- Confirm performance results, corrected to field conditions
- Confirm mechanical test acceptance
- Confirm auxiliary system test acceptance
- Inspect components and confirm acceptance
- Agree to any corrective action in writing
- Accept or reject test – any corrective action requires a retest!

Fig 3.24.9 • Post test – key facts

What happens if the test is not successful? Approximately 50% of the tests that I have either run or participated in over my career have not been successful in regards to one component or more not meeting test requirements. Possible rejected test action is noted in Figure 3.24.10.

- Immediately provide details to the project team
- Confirm if field conditions can handle the abnormality
- Determine if the 'as tested' machine will meet all reliability requirements
- If the decision is to reject, inform the vendor and detail the reasons
- Do not accept unrealistic delivery delays

Fig 3.24.10 • Rejected test action

Finally, do not forget the importance of test report requirements. The test report is a most important document that represents the 'baseline performance of the unit' and will be a benchmark for field operation acceptability. Test report – key facts are noted in Figure 3.24.11.

- The shop test is the field baseline!
- The test report must be detailed and complete
- Review the preliminary contents of the report before leaving the test floor
- Obtain the actual test results
- When the final report is received, check the results obtained at test against the final report
- Immediately contact the vendor if there are any differences

Fig 3.24.11 • Test report – key facts

Shop test checklist

1. Scope

- ☐ Appropriate industry specs included (ANSI, API, NEMA, etc.)
- ☐ In-house and/or E&C specs included
- ☐ Project specific requirements
- ☐ Performance test ☐ All rotors ☐ One rotor
- ☐ Test (equivalent) conditions
- ☐ Field (actual) conditions
- ☐ Mechanical test ☐ All rotors ☐ One rotor
- ☐ Test (equivalent) conditions
- ☐ Field (actual) conditions
- ☐ Unit test of all equipment (string test)
- ☐ No load ☐ Includes auxiliary systems
- ☐ Full load ☐ Does not include auxiliary systems
- ☐ Use of job couplings and coupling guards
- ☐ Testing of instrumentation, control and protection devices
- ☐ Auxiliary system test
- ☐ Lube oil ☐ Test press ☐ Full press
- ☐ Control oil ☐ Test press ☐ Full press
- ☐ Seal oil ☐ Test press ☐ Full press
- ☐ Seal gas ☐ Test press ☐ Full press
- ☐ Fuel ☐ Test press ☐ Full press
- ☐ Flow measurement required
- ☐ Time base recording of transient events required
- ☐ Use of all special tools during test (rotor, removal, coupling, etc.)
- ☐ Shop test attendance (includes assembly and disassembly)
- ☐ Site reliability ☐ Site maintenance ☐ Site operations
- ☐ Review of instruction book during shop test visit
- ☐ Test agenda requirements
- ☐ Mutually agreed limits for each measured parameter
- ☐ Issue for approval 2 months prior to contract test date

2. Pre-test meeting agenda

- ■ Meet with test department prior to test to:
- ☐ Confirm test agenda requirements
- ☐ Confirm all test parameters have mutually agreed established limits
- ☐ Review all instrument calibration procedures
- ☐ Review test set-up drawing
- ☐ Review data calculation (data reduction) methods
- ☐ Define work scopes for site personnel (assembly and disassembly witness, video or still frame pictures, etc.)
- ■ Confirm assigned vendor service engineers will be in attendance for:
- ☐ Assembly
- ☐ Disassembly
- ☐ Test

Shop test activity

- ☐ Review and understand test agenda prior to test
- ■ Immediately prior to test meet with assigned test engineer to:
  - ☐ Review schedule of events
  - ☐ Designate a team leader
  - ☐ Confirm test team leader will be notified prior to each event
  - ☐ 'Walk' test set-up to identify each instrumented point
  - ☐ Confirm calibration of each test instrument
  - ☐ Obtain documents for data reduction check – if applicable (flow meter equations, gas data, etc.)
- ■ During test (Note: coordinate with test personnel to avoid interference)
  - ☐ Review 'as measured' raw data for consistency
  - ☐ 'Walk' equipment – look for leaks, contract instrument, piping and baseplate vibration, etc.
  - ☐ Use test team effectively – assign a station to each individual
  - ☐ Ask all questions now, not later, while an opportunity exists to correct the problem
  - ☐ Check vendor's data reduction for rated point – if applicable
- ■ After test
  - ☐ Inspect all components as required by the test agenda (bearings, seals, labyrinths, RTD wires, etc.)
  - ☐ Review data reduction of performance data corrected to guarantee conditions
  - ☐ Review all mechanical test data
  - ☐ Generate list of action (if applicable) prior to acceptance of test
  - ☐ Approve or reject

Best Practice 3.25

Do not allow centrifugal compressor mechanical tests to be run under vacuum conditions

This is specified because this type of test does not expose the components to the following field conditions:

- Partial shaft torque
- Thermal expansion of rotor
- Partial thrust load

Some of these issues have been:

- Vibration due to insufficient axial gap between components caused by thermal expansion of the rotor
- Thrust bearing high pad temperatures
- Axial rotor to diaphragm rubs

Lessons Learned

Centrifugal compressors that have had mechanical tests run in a vacuum have encountered field problems that would have been exposed under partial pressure mechanical tests.

Benchmarks

This best practice has been followed since the 1970s, and has resulted in trouble free start-ups and centrifugal compressor reliabilities of greater than 99.7%.

B.P. 3.25. Supporting Material

Mechanical test requirements prescribed by API 617 for centrifugal compressors can be met under full load, limited load or no load (vacuum). No load testing (vacuum testing) can be accomplished with lowest set up time and operating cost. Given the recent amount of compressor shop load, vacuum mechanical testing has become the choice of compressor suppliers. Since API 617 does not outlaw shop mechanical vacuum tests, end-users are left to either accept these tests (sometimes as a surprise) or to take a proactive approach in the job specifications to prohibit vacuum mechanical tests.

If mechanical vacuum testing has been performed for years (since the 1960s), then why is it a concern? Today, with mega process units and the high cost of products, daily plant production revenues have never been higher. A day's production loss can exceed millions of dollars. Shop testing therefore must closely mirror the field conditions that the compressor will experience, to ensure that the installed unit will achieve the highest level of reliability.

Shop mechanical tests conducted under vacuum do not duplicate any field conditions other than shaft speed. Impellers do not experience torque, temperature or thrust loads. Rotor assemblies are not subjected to thermal expansion. Journal bearings are not subjected to aerodynamic forces. Thrust bearings are not subjected to axial loads. Seals are not subjected to operating sealing pressures.

In addition, many centrifugal compressor vendors are using vaned or low-solidity diffuser vanes to increase head produced and compressor efficiency. While these components usually achieve their objectives, they can produce aerodynamic instability, which can result in high sub-synchronous vibration levels and cause vibration trips.

Vacuum testing cannot determine whether aerodynamic instabilities will be present since the impellers will not be subjected to flow conditions. It can be argued that if a performance test is conducted in accordance with ASME PTC-10, the compressor will be tested under some load, which will serve to eliminate the concerns noted above. However, since API 617 does not set limits on the performance test equivalent speed, a performance test is usually run at speeds far from design (50% – 65% of rated speed). Lower operating speeds do not duplicate field aerodynamic forces on the rotor of centrifugal forces.

To ensure optimum field reliability, users should always require that a mechanical shop test be run under a minimum of 10% load and that the shop-testing program will subject the rotor to operating temperatures. It has been our experience that the most costly revenue-reducing field problems are related to rotor thermal expansion issues and aerodynamic instabilities.

Thanks to the extensive rotor-dynamic requirements of API 617, field rotor response issues have been significantly reduced in recent years and are not of concern if all rotor system design issues are met. An argument can be made for only mechanically testing at 10% load to ensure aerodynamic forces will not be present during field operation. Considering the present amount of industry experience with high-density compressor applications (reinjection, recycle and synthesis gas), we offer the following guidelines for determining if a full pressure mechanical test should be performed.

A full-pressure shop mechanical test is required if:

- The vendor does not have experience with this pressure level at similar gas densities.
- If experience at similar densities is not with the gas path components to be used for the project (vaned diffusers and aerodynamic stability devices).

One final note regarding full-pressure tests: these tests, designed to demonstrate that the compressor will be free of aerodynamic instabilities, have been confused with full-load tests. Full-load tests do not necessarily duplicate field operating pressures, and therefore do not ensure field operation will be free of aero instabilities.

Best Practice 3.26

Surge protection system design – incorporate a dedicated bypass line intercooler in large systems to minimize the volume of trapped gas for optimum surge system response.

Using a dedicated surge line intercooler allows the surge line takeoff and check valve to be connected immediately after the discharge flange.

This design results in the smallest volume of gas that can re-enter the compressor during a surge cycle and provides the quickest surge system response.

A fast system response allows the surge control line to be set close to the actual surge line which results in the largest possible compressor operating range.

Incorporate this design during the pre-FEED phase of the project to ensure implementation.

Lessons Learned

Using the aftercooler for recycle gas heat removal, as opposed to a dedicated surge line cooler, results in a large volume of trapped gas.

It is found that this will:

- Reduce surge system response
- Require increased surge to surge control line margin
- Reduce the compressor operating range
- Possibly require an additional hot gas bypass line to prevent surge during emergency shutdowns

A recent gas booster project that used the aftercooler and not a dedicated surge line cooler required a large surge to surge control line margin of 20% to prevent surge during emergency shutdowns (ESDs) thus reducing the operating range of the compressor.

Benchmarks

This best practice was recently implemented for a gas booster project and justified based on total cost (not having to use an additional hot gas bypass line) and reliability (minimum surge system hardware).

B.P. 3.26. Supporting Material

System objectives

The surge system objectives are to ensure that the gas velocity in each impeller stage in any dynamic compressor exceeds the critical velocity that will cause stall. Refer to Figure 3.26.1.

- Ensure the relative gas velocity in each impeller stage exceeds the critical velocity that will cause stall

Fig 3.26.1 • The surge system objective

Available options

In order to ensure that impeller velocity always exceeds the stated critical velocity, a number of options are available. These options mainly fall into three (3) categories. Refer to Figure 3.26.2.

- Shutdown unit before critical flow is reached
- Increase the relative gas velocity:
  - By impeller speed increase
  - By adjusting inlet gas angle
- Decrease system resistance:
  - Process system decrease
  - Direct gas flow through blow off or recycle line

Fig 3.26.2 • Options to achieve the objective

1. Shut down the unit before the critical flow is reached.
Naturally, when one considers that most dynamic compressors are in critical service, that is un-spared, this option is not possible.
2. Increase the relative gas velocity.
This objective can be achieved by either increasing the speed of the impellers or by adjusting the inlet angle of the gas relative to the impeller vanes. Both alternatives result in the increase of turbo-compressor energy and should provide the higher velocity through the impeller. Note, however, that this alternative may not be effective under certain system resistance characteristics.
3. Decrease the system resistance.
This can be achieved by either directly reducing process system resistance, fully opening a throttle valve, or by directing gas flow through a blow off or a recycle line. Of the two alternatives available, the latter is more efficient since it does not affect the energy in the process system.

Given the applications in which turbo-compressors are employed, it can be readily observed that option three (3) is the most cost-effective way to protect turbo-compressors against surge and corresponding mechanical damage.

System design considerations

Accepting the fact that reduced system resistance is the most cost effective way to eliminate surge damage, a number of surge system design considerations should be mentioned. These considerations fall into two categories; compressor considerations and process system considerations. The considerations are presented in Figure 3.26.3.

- Compressor considerations
- Fast system response
- Control line automatic adjustment
- Ability to adjust control line to gas density and operating point changes
- Recycle stream cooling
- Check valve location
- Process system considerations
- Gradual process flow changes
- Flow reversals
- Start-up procedure
- Shutdown procedure

Fig 3.26.3 • System design considerations

A history of system types

Figure 3.26.4 presents a general chronological listing of surge system history.

- ‘Surgeless’ applications
- Surge alarm systems
- Single parameter systems – flow
- Biased systems – flow and differential pressure
- Modern systems
- Compensation
- System back-ups
- Controller output options

Fig 3.26.4 • Surge system history

In my experience, many so called ‘surgeless’ applications have been observed. These are applications where the process designer assesses the system and concludes that there is no possibility to surge the machine. Often, this assessment is based on system resistance alone, the conclusion being that, since there are no variable resistance points in the process system (coolers, reactors that can foul or control valves), the turbo-compressor cannot surge. If the molecular weight and gas inlet temperature are constant, and the turbo-compressor will not be subjected to mechanical damage or fouling, this assessment is true. If any of the above mentioned possibilities can occur, however, the turbo-compressor in question can be forced into surge as a result of its inability to produce the energy required for throughput flow. As previously discussed, the operating point of any compressor is the equilibrium between required process system energy and the compressor produced energy. Both can change, and do.

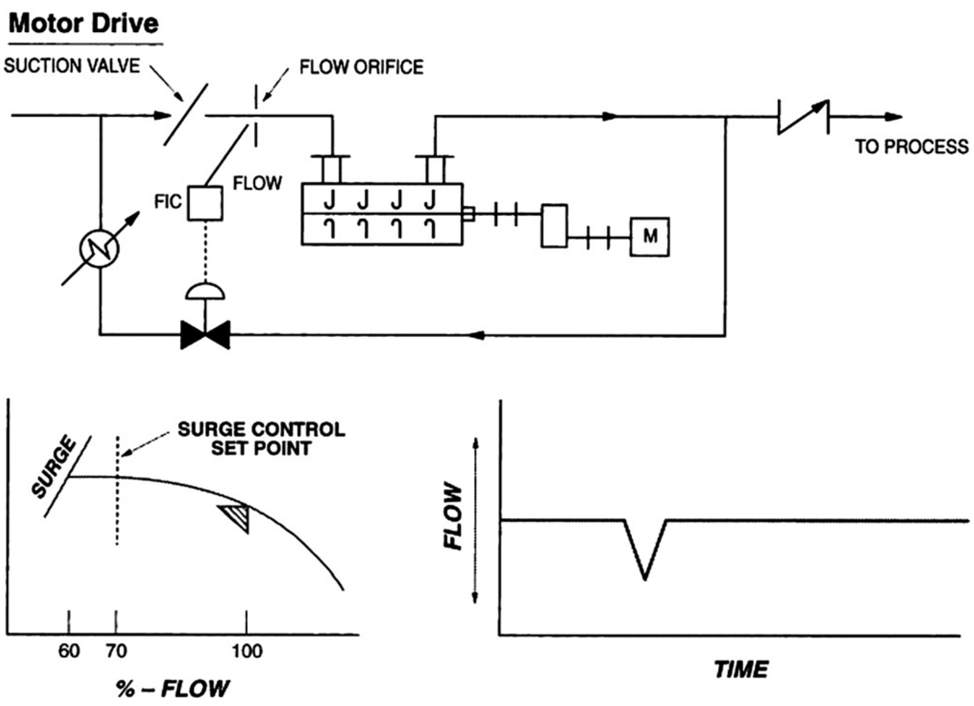


Fig 3.26.5 • A single parameter surge protection system

The second system encountered has been surge alarm type systems. Again, the process designer assesses a particular system and determines that the possibility of surge is minimal. In addition, the application, hopefully, is such that a shutdown can be achieved without significant revenue loss. Therefore, the surge system is simply an alarm or shutdown system that will stop compressor operation in the event of surge, thus protecting the machine. It should be mentioned, however, that a full assessment should be made to determine the cost impact of such a shutdown, and the potential problems incurred on the start-up. A properly installed surge system can eliminate these unnecessary shutdowns.

Before leaving this subject, the need for a reliable of the surge detection system should be mentioned. Frequently, these detection systems are simply differential pressure switches. Depending on the application, surge can be either very violent or very light. However, in the case of low pressure applications or low molecular weight applications the effect of surge (flow and pressure pulsations) will be minimal and the sensitivity of the surge sensor device must be investigated to ensure that such an event is detected.

Figure 3.26.5 shows a single parameter surge protection system. These systems have been employed extensively, and in simple terms, consist of a flow measuring device and an anti-surge valve. These systems can work if properly sized and installed. However, it must be understood that a one parameter system will not be economical in that a rather large area of operation as in the cases of suction throttle constant speed devices and variable speed drives will not be allowed. The result will be unnecessary re-circulation and a loss of product revenue. The next system shown in Figure 3.26.6 is the standard, two parameter, or biased system that has been employed almost exclusively in the industry over the years.

This system consists of flow measurement and differential pressure measurement across the compressor, and results in a surge control line that roughly parallels the actual surge line of the compressor. It allows operation across a wider range than the single parameter system and, if properly sized, can adequately protect the compressor.

The final system in the evolution of surge controls is the modern microprocessor system, which affords many options to fully protect the turbo-compressor against surge occurrences. These systems usually employ compressor inlet and discharge temperature inputs in addition to flow and ΔP for each

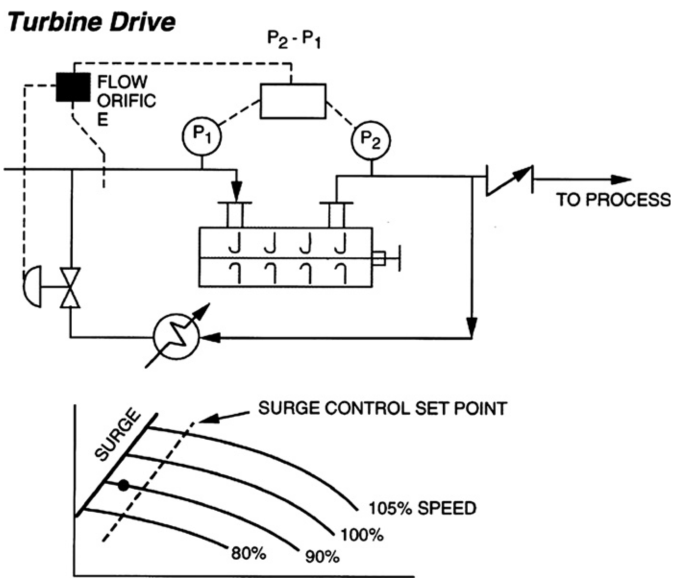


Fig 3.26.6 • The standard two parameter or biased system

compressor section. Examine the relationship for polytropic head and you will see that this approach yields a more accurate approximation of the operating point. This is because input of T_1 and T_2 allow calculation of:

$$\frac{N-1}{N}, T_1, Z_1 \quad \text{and M.W.}$$

The other available options include compensation for gas composition changes, system backups that will protect the compressor in the event of a surge and controller output and options to enable the surge system to be soft in relation to the process system. That is, operation of the surge control system will not adversely affect the process system.

Best Practice 3.27

Trend centrifugal compressor performance (head and efficiency) and integrate performance trends with component mechanical condition trends to achieve the highest possible level of safety and reliability.

Use the concept of component condition monitoring to trend the following components and quickly determine causes of condition change:

- Rotor
- Journal bearings
- Thrust bearings
- Shaft end seals
- Auxiliary systems

Lessons Learned

Approximately 80% of the root causes of component failure are contained in process changes. Failure to integrate performance monitoring (head and efficiency) with traditional mechanical monitoring (vibration, bearing temperatures and seal condition) will significantly reduce compressor reliability and revenue.

The majority of plants that we visit still define condition monitoring of un-spared compressors by mechanical condition monitoring only. As a result, centrifugal compressor disassembly is still done on a time basis (preventive maintenance) and not on a condition basis (predictive maintenance).

Benchmarks

This best practice has been used in all sectors of the industry since the mid-1980s to achieve centrifugal compressor reliabilities in excess of 99.7%, and to minimize turnaround activity. Clean compressor services (refrigeration) were not dissembled for internal inspection until the 4th 4-year turnaround, based on this best practice (after 16 years of operation).

B.P. 3.27. Supporting Material

The major machinery components

Think of all the machinery that you have been associated with and ask; "What are the major components and systems that are common to all types of rotating equipment?"

Figure 3.27.1 presents the major component classifications for any type of machinery:

- Pumps
- Steam turbines
- Compressors
- Motors
- Gas turbines
- Fans
- Others

- Rotor
- Radial bearing
- Thrust bearing
- Seal
- Auxiliary systems

Fig 3.27.1 • Major machinery components and systems

Regardless of the type of machinery, monitor these components, and you will know the total condition of the machine.

Component condition monitoring

As previously stated, component and system functions must first be defined and the normal values for each component listed. These facts are presented in Figure 3.27.2.

- Define the function of each affected component
- Define the system in which each affected component operates
- List the normal parameters for each affected component and system component

Fig 3.27.2 • Component and system functions

Once the function of each component is defined, each major machinery component can be monitored, as shown in Figure 3.27.3.

Baseline

Having defined all condition parameters that must be monitored, the next step in a condition monitoring exercise is to obtain baseline information. It is important to obtain baseline information as soon as physically possible after start-up of the

- **Define** each major component
- **List** condition monitoring parameters
- **Obtain** baseline data
- **Trend** data
- **Establish** threshold limits

Fig 3.27.3 • Component condition monitoring

equipment. However, operations should be consulted to confirm when the unit is operating at rated or lined out conditions. Obtaining baseline information without conferring with operations is not recommended, since faulty information could be obtained which could thus lead to erroneous conclusions in predictive maintenance. Figure 3.27.4 states the basics of a baseline condition.

If you don't know where you started, you do not know where you are going!

Fig 3.27.4 • Base line condition

It is amazing to us how many times baseline conditions are ignored. Please remember Figure 3.27.4, and make it a practice to obtain baseline conditions as soon as possible after start-up.

Trending

Trending is simply the practice of monitoring parameter condition with time. Trending begins with baseline conditions, and will continue until equipment shutdown. In modern day thought, it is often conjectured that trending must be performed by micro-processors and sophisticated control systems. This is not necessary! Effective trending can be performed by periodic manual observation of equipment, or using equipment available to us in the plant, such as DCS systems, etc. The important point is to obtain the baseline and trends of data on a periodic basis. When trending data, threshold points should also be defined for each parameter that is trended. This means that when the pre-established value of the parameter is exceeded, action must be taken regarding problem analysis. Setting threshold values at a standard percentage above the normal value is recommended. Typically values are of the order of 25–30% above baseline values. However, these values must be defined for each component based on experience. Figure 3.27.5 presents trending data for a hydrodynamic journal bearing. All of the parameters noted in Figure 3.27.5 should be monitored to define the condition of this journal bearing.

Specific machinery component and system monitoring parameters and their limits

On the following pages is contained information concerning what parameters should be monitored for each major machinery component to determine its condition. In addition, typical limits are noted for each component.

These limits represent the approximate point at which action should be planned for maintenance. They are not intended to define shutdown values.

The rotor

Rotor condition defines the performance condition (energy and efficiency) of the machine. Table 3.27.1 presents this value for a pump.

Radial bearings

Figures 3.27.6 and 3.27.7 present the facts concerning antifriction and hydrodynamic (sleeve) radial or journal bearing condition monitoring.

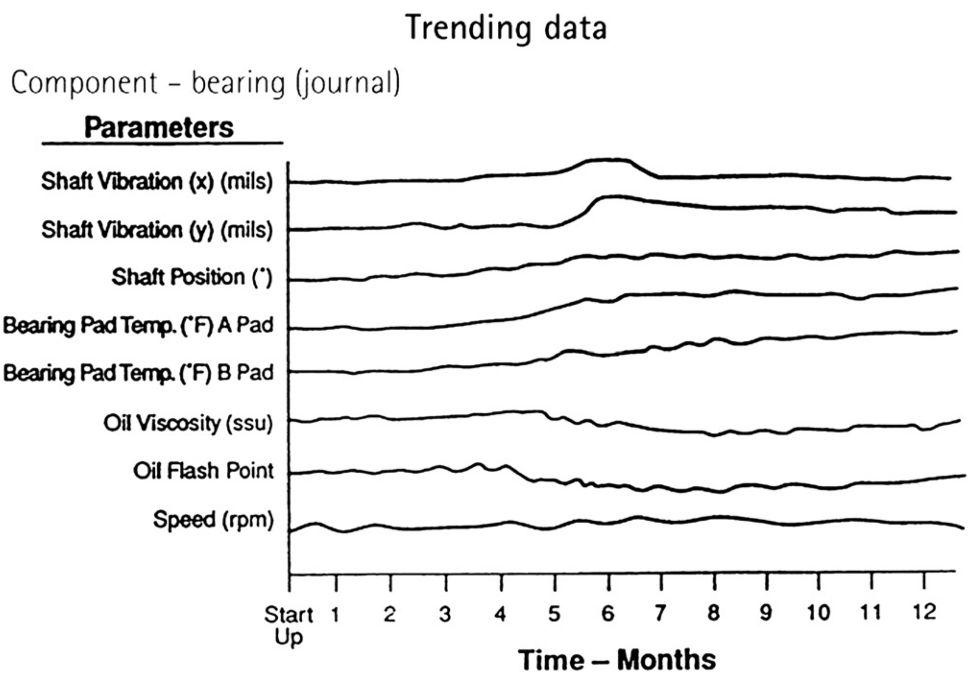


Fig 3.27.5 • Trending data

Table 3.27.1 Pump performance monitoring

1. Take value at minimum flow (shut off discharge valve)	
2. Measure:	
■ P_1	■ Driver bhp
■ P_2	■ Specific gravity
Where: P_1 and P_2 = psig, bhp = brake horsepower.	
1. Calculate:	
A. head produced $\frac{m - Kgp^{-}}{Kgm}\left(\frac{ft - lb_f}{lb_m}\right) = \frac{\Delta P(kPa) \times .102}{S.G.}\left(\frac{\Delta P \times 2.311}{S.G.}\right)$	
B. pump efficiency (%) $= \frac{hd \times \frac{m3}{hr} \times S.G.}{360 \times kW}\left(\frac{hd \times gpm \times SG}{3960 \times bhp}\right)$	
2. Compare to previous value; if $> -10\%$ perform maintenance	

Journal bearing (hydrodynamic)

Parameter	Limits
1. Radial vibration (peak to peak)	2.5 mils (60 microns)
2. Bearing pad temperature	220°F (108°C)
3. Radial shaft position*	> 30° change and/or 30% position change
4. Lube oil supply temperature	140°F (60°C)
5. Lube oil drain temperature	190°F (90°C)
6. Lube oil viscosity	off spec 50%
7. Lube oil particle size	> 25 microns
8. Lube oil water content	below 200 ppm

*Except for gearboxes where greater values are normal from unloaded to loaded.

Fig 3.27.7 • Condition monitoring parameters and their alarm limits – journal bearing (hydrodynamic)

Thrust bearings

Figures 3.27.8 and 3.27.9 show condition parameters and their limits for anti-friction and hydrodynamic thrust bearings.

Seals

Figure 3.27.10 presents condition parameters and their limits for a pump liquid mechanical seal.

Auxiliary systems

Condition monitoring parameters and their alarm limits are defined in Figures 3.27.11 and 3.27.12 for lube and pump flush systems.

Figures 3.27.13, 3.27.14 and 3.27.15 present condition monitoring parameters and limits for dynamic compressor performance, liquid seals and seal oil systems. One final recommendation is presented in Figure 3.27.13.

Predictive maintenance (PDM) techniques

Now that the component condition monitoring parameters and their limits have been presented, predictive maintenance techniques must be used if typical condition limits are exceeded. The following section will address the techniques used for predictive maintenance analysis and root cause analysis techniques.

Now that the principles of turbo-compressor performance have been explained and hopefully understood, they can be implemented to observe internal turbo-compressor condition changes. Always remember that we want to know the internal, not the external condition. Figure 3.27.16 presents the outline of a case history that will show the value of performance condition monitoring.

The first plan

I visited a refinery a few years ago to troubleshoot an existing turbo-compressor problem. While I was on site, another process unit, a reformer, was scheduled for a turnaround. Since I was already on site, I was invited to the pre-turnaround meeting and became involved with turnaround activities.

During this meeting I learned that the recycle compressor was scheduled for a bearing inspection only (radial and thrust). I asked why. The answer was that it was normal practice. I asked if I could see the bearing condition monitoring data (vibration, bearing displacement, bearing temperature, oil flow – valve position and oil sample). I was shown to a room and told "It's in

Journal bearing (anti-friction)

Parameter	limits
1. Bearing housing vibration (peak)	0.4 inch/sec (10 mm/sec)
2. Bearing housing temperature	180°F (85°C)
3. Lube oil viscosity	off spec 50%
4. Lube oil particle size	
• non metallic	25 microns
• metallic	any magnetic particle in the sump
5. Lube oil water content	below 200 ppm

Fig 3.27.6 • Condition monitoring parameters and their alarm limits – journal bearing (anti-friction)

Thrust bearing (anti-friction)

Parameter	Limits
1. Bearing housing vibration (peak)	
• radial	0.4 in/sec (10 mm/sec)
• axial	0.3 in/sec (1 mm/sec)
2. Bearing housing temperature	185°F (85°C)
3. Lube oil viscosity	off spec 50%
4. Lube oil particle size	
• non metallic	> 25 microns
• metallic	any magnetic particles with sump
5. Lube oil water content	below 200 ppm

Fig 3.27.8 • Condition monitoring parameters and their alarm limits – thrust bearing (anti-friction)

Thrust bearing (hydrodynamic)

Parameter	Limits
1. Axial displacement*	> 15–20 mils (0.4–0.5 mm)
2. Thrust pad temperature	220°F (105°C)
3. Lube oil supply temperature	140°F (60°C)
4. Lube oil drain temperature	190°F (90°C)
5. Lube oil viscosity	off spec 50%
6. Lube oil particle size	> 25 microns
7. Lube oil water content	below 200 ppm

**and* thrust pad temperatures > 220°F (105°C)

Fig 3.27.9 • Condition monitoring parameters and their alarm limits – thrust bearing (hydrodynamic)

Pump liquid mechanical seal

Parameter	Limits
1. Stuffing box pressure	< 25 psig (175 kpa)**
2. Stuffing box temperature	Below boiling temperature for process liquid
3. Flush line temperature	+/– 20°F (10°C) from pump case temp
4. *Primary seal vent pressure (before orifice)	> 10 psi (70 kpag)

*On tandem seal arrangements only
**Typical limit – there are exceptions (Sundyne Pumps)

Fig 3.27.10 • Condition monitoring parameters and their alarm limits – pump liquid mechanical seal

Lube oil systems

Parameter	Limits
1. Oil viscosity	off spec 50%
2. Lube oil water content	below 200 ppm
3. Auxiliary oil pump operating yes/no	operating
4. Bypass valve position (P.D. pumps)	change > 20%
5. Temperature control valve position	Closed, supply temperature > 130 55°C)
6. Filter ΔP	> 25 psid (170 kpag)
7. Lube oil supply valve position	change > +/–20%

Fig 3.27.11 • Condition monitoring parameters and their alarm limits – lube oil systems

Pump seal flush (single seal, flush from discharge)

Parameter	Limits
1. Flush line temperature	+/–20°F (+/–10°C) of pump case temperature
2. Seal chamber pressure	< 25 psi (175 kpa) above suction pressure

Fig 3.27.12 • Condition monitoring parameters and their alarm limits – pump seal flush

1. Calibrated: pressure and temperature gauges and flow meter
2. Know gas analysis and calculate k, z, m.w.
3. Perform as close to rated speed and flow as possible
4. Relationships:

$$\text{A. } \frac{N\text{-}1}{N} = \frac{LN\frac{(T_2)}{(T_1)}}{LN\frac{(P_2)}{(P_1)}} \qquad \text{B. } EFFICIENCY_{poly} = \frac{\frac{k\text{-}1}{k}}{\frac{n-1}{n}}$$

$$\text{C. } HEAD_{POLY} = \frac{\text{m kgf}}{\text{kgm}} = \frac{847.4}{\text{mw}}$$

$$\left(\frac{Ft\text{-}lb_f}{Lb_m}\right) = \frac{1545}{MW} \times T_1 \times \frac{n}{n\text{-}1} \times Z_{avg} \times \left(\left(\frac{P_2}{P_1}\right)^{\frac{n\text{-}1}{n}} - 1\right)$$

5. Compare to previous value, if decreasing trend exists greater than 10%, inspect at first opportunity

Fig 3.27.13 • Compressor performance condition monitoring

Compressor liquid seal

Parameter	Limits
1. Gas side seal oil/gas ΔP	
• bushing	< 12 ft. (3.5m)
• mechanical contact	< 20 psi (140 kpa)
2. Atmospheric bushing oil drain temperature	200°F (95°C)
3. Seal oil valve* position	> 25% position change
4. Gas side seal oil leakage	> 20 gpd per seal

*supply valve = + 25%
return valve = – 25%
Note this assumes compressor reference gas pressure stays constant

Fig 3.27.14 • Condition monitoring parameters and their alarm limits – compressor liquid seal

there". In desperation I concluded "Oh, what the hey, we usually inspect bearings anyway".

The second plan

Well, as you might expect, the bearings were removed and they were pretty badly damaged. Wipes on both journal and thrust bearing pads and indications of particle rubs showed that we really needed to both inspect and flush the oil system. It was later learned that an oil system accumulator had a continuous nitrogen purge that was at a higher pressure than the oil system and, oh yes, the bladder had a hole in it! Now back to the bearings. We didn't know the condition or trends of parameters.

Compressor liquid seal oil systems

Parameters	Limits
1. Oil viscosity	off spec 50%
2. Oil flash point	below 200°F (100°C)
3. Auxiliary oil pump operating yes/no	operating
4. Bypass valve position (P.D. pumps)	change > 20%
5. Temperature control valve position	closed, supply temperature 130°F (55°C)
6. Filter ΔP	25 psid (170 kpag)
7. Seal oil valve position	change > 20% open (supply) > 20% closed (return)
8. Seal oil drainer condition	(proper operation)
9. Constant level (yes/no)	level should be observed
10. Observed level (yes/no)	level should not be constant
11. Time between drains	approximately 1 hour (depends on drainer volume)

Fig 3.27.15 • Condition monitoring parameters and their alarm limits – compressor liquid seal oil systems

- The first plan
- The second plan
- The third plan

Fig 3.27.16 • 'The long, long, long, turnaround'

Were there upsets? High vibrations, temperatures, etc.? Were there a lot of surges? Nobody could seem to remember! Does this sound familiar? Well, as you might expect, we prepared a second plan – we had better inspect the oil seals. After all, we didn't have to get into the compressor so we had sufficient time.

The third plan

No doubt about it, the seals were really bad. Bushing seals, both the atmospheric bushings and gas side bushings were wiped and the atmospheric side showed typical evidence of high ΔT, probably due to start-up on low pressure N_2 (to save N_2 costs). Since the gas was sweet, seal leakage was returned to the reservoir via a degassing tank and there was no seal condition monitoring trend data available. (Seal oil leakage – gas side, seal oil leakage – atmospheric side, which could have been obtained by trending seal oil valve position). Am I making my point? Well, it was crunch time (decision time). Were the damaged seals the root cause of bearing failure? Was it the dirty lube and seal system or was there another deeper cause inside of the compressor? I could go on and on but in the interest of time let it suffice to say:

- We decided to open the barrel and compressor – we had time, based on schedule.
- We could not find tools and when we did, they did not work.
- Oh yes, a piece of a suction strainer (supposedly only for start-up) had been lodged in the case/barrel interface for years.
- When we finally got the compressor apart – two days after the turnaround was complete, we got lucky – the internals including the balance drum were perfect!

If only we had established a performance condition monitoring program, as well as seal, bearing, balance drum and lube/seal system condition monitoring.

Incidentally, this was a major refinery that placed high value on reliability, maintainability, root cause analysis, etc. They had all the books and had sent people to the right workshops – even one of mine!

The objectives of turbo-compressor condition monitoring are presented in Figure 3.27.17.

Know turbo-compressor internal condition to determine:
- Loss of daily revenue
- Justification for turnaround activities
- Root cause

Fig 3.27.17 • The objectives

The parameters

What parameters must be measured? Readers should be able to answer this question readily at this point. Fig 3.27.18 presents the reduced parameters – the answers and Figure 3.27.19 contains all the factors necessary for calculation.

- Polytropic (or isentropic) head
- Actual inlet flow rate
- Polytropic (or isentropic) efficiency
- Polytropic exponent
- Horsepower
- Speed (or guide vane position)

Fig 3.27.18 • The turbo-compressor performance condition parameters

- From gas analysis and equation of state
- M.W.
- K average
- Z inlet
- P_1, P_2
- T_1, T_2
- Flow rate
- Speed (if applicable)
- Guide vane position (if applicable)

Fig 3.27.19 • The data (factors) required

Accuracy

Accuracy of data involves both calibration and location of instruments. Before proceeding, we need to present some

important facts concerning condition monitoring. These facts are presented in Figure 3.27.20.

- Specific data (to confirm field guarantees) requires pre-planning to ensure accuracy
- Trends produce relative change in values

Fig 3.27.20 • Turbo-compressor performance condition monitoring facts

Refer back to the 'Life cycle of equipment' in Chapter 1, Figure 1.3.2. If specific data is required to confirm field guarantees, the project team must pre-invest and plan for the proper instruments, number of and location in the design phase of the project!

One (especially the rotating equipment engineer) must remember the words in Figure 3.27.21.

'The field is a place to produce product, not conduct laboratory experiments'

- 1981 A.D. … a production supervisor

Fig 3.27.21 • Remember these words!

Figure 3.27.22 presents important facts concerning instrument calibration.

Inaccurate gas analysis procedures can produce some pretty wild results! Efficiencies that exceed 100%! Figure 3.27.23 presents guidelines for accurate gas analysis.

Location and number of field instruments are just as important as instrument calibration. I cannot overemphasize the importance of this fact. Convince management (plant management first, then project management) to pre-invest in turbo-compressor performance and instrumentation. If you don't know accurately what's happening inside the patient, you can't effectively *prevent* problems! That old medical analogy to a turbo-compressor again! Figure 3.27.24 presents some guidelines from the ASME Turbo-compressor Test Code (PTC-10) regarding the number and location of instruments. The field is not a laboratory!

Field turbo-compressor condition monitoring must be planned such that it does not impact production rates.

Establish a baseline

Before we discuss trending or fully understand turbo-compressor performance condition monitoring, the baseline condition must be defined. Since performance condition monitoring, as well as any type of condition monitoring, is concerned with relative change, the starting point must be established. Figure 3.27.25 defines the baseline condition.

All performance condition monitoring instruments must have known, accurate calibration values!
If:

- ΔT as measured is off 5%, efficiency is affected 20%!
- Inaccurate gas analysis of 5% can affect efficiency 20%
- Pressure gauge inaccuracy has *much less* of an effect on efficiency!

Fig 3.27.22 • Instrument calibration facts

- Take samples from the top of pipe
- Measure gas temperature
- Be sure to analyze gas at temperatures equal or greater than field conditions
- Confirm laboratory experience and methods

Fig 3.27.23 • Gas analysis guidelines

I can't remember how many times in my career I have said, "I sure wish we had established the 'baseline condition'".

The baseline turbo-compressor performance condition should be established once the process unit is on spec, immediately, not one day, week, month or year later. The sooner the baseline is established, the more data there will be in your machine historical file. Many problems occur in the initial period of operation; be sure you have recorded the data. The baseline condition then represents the first point on the trend graph of any measured parameter. Figure 3.27.26 recommends when baseline conditions should be taken.

Trending

Trending requires that a parameter be monitored over a period of time to determine if significant change occurs. These factors are defined in Figure 3.27.27.

It should be noted that significant change is not synonymous with alarm point. If an initial value is small, a small change can be significant and still be far away from the set alarm point!

Details concerning field performance testing will be discussed in the next chapter. We have presented suggested turbo-compressor performance trend parameters in the final figure of this chapter; Figure 3.27.28.

We have included printouts of our excel spreadsheets used for centrifugal compressor CCM. These sheets can be used to completely trend centrifugal compressors using the CCM approach.

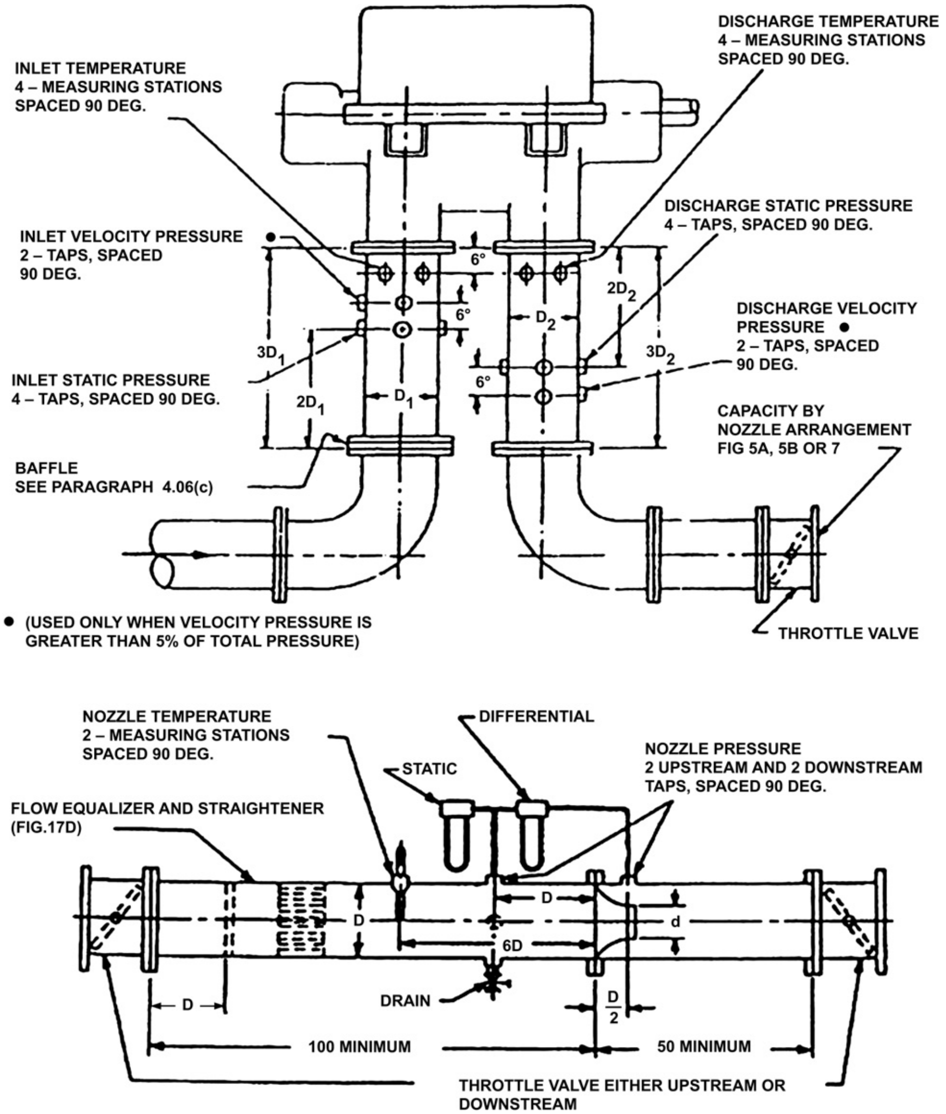


Fig 3.27.24 ● Performance instrumentation location guidelines
(Courtesy of ASME PTC-10) (typical for multi-stage compressor)

- The baseline condition establishes the initial ('new' or 'rebuilt') condition from which all changes (trends) are measured.

Fig 3.27.25 ● The baseline condition

- Establish baseline condition
- Trend parameter (Y axis) vs. time (X axis)
- Compare related trends with same time scale (X axis)
- A significant change in a parameter is a 25% – 50% change.

Fig 3.27.27 ● Trending guidelines

Baseline performance conditions should be established when the process is on spec. immediately after:

- New compressor start-up
- Turnaround (if internals, or thrust bearing was changed)

Fig 3.27.26 ● Baseline performance conditions – when?

- Polytropic head ÷ speed^2 (Hd_{POLY}/N^2)
- Actual flow ÷ speed (Q/N)
- Polytropic efficiency (η_{POLY})
- In certain cases,
- Molecular weight
- Inlet temperature

Fig 3.27.28 ● Useful trend parameters

Table 3.27.2 Performance Monitoring Datasheets: No.1

Compressor Performance Monitoring Exercise					
Item #					
Group					
Given					
M.W.					
P_1 (kg/cm^{2a})					
T_1 (C)					
P_2 (kg/cm^{2a})					
T_2 (C)					
K avg					
Z1					
Flow (m^3/hr)					
N (RPM)					
Calculate					
(K-1)/K	-	-	-	-	-
(n-1)/n	-	-	-	-	-
Gas Density (kg/m^3)	-	-	-	-	-
Mass Flow (kg/hr)	-	-	-	-	-
KW	-	-	-	-	-
Poly Hd (kj/kg)	-	-	-	-	-
Poly Hd (m)	-	-	-	-	-
Poly Eff'y	-	-	-	-	-
Does Compressor Need Maintenance?					

Table 3.27.3 Performance Monitoring Datasheets: No.2

Component Condition Monitoring Worksheet
Item #:
Date/Time:
Journ. Brgs.
DE Horiz. Vibes (micron)
DE Vert. Vibes (micron)
DE Pad Temp. (Deg C)
DE Pad Temp. (Deg C)
NDE Horiz. Vibes (micron)

Table 3.27.3 Performance Monitoring Datasheets: No.2—Cont'd

Component Condition Monitoring Worksheet
NDE Vert. Vibes (micron)
NDE Pad Temp. (Deg C)
NDE Pad Temp. (Deg C)
Thrust Brgs.
Axial displ.
Axial displ.
Active Pad Temp. (Deg C)
Active Pad Temp. (Deg C)
Inactive Pad Temp. (Deg C)
Inactive Pad Temp. (Deg C)
Balance Line Diff. P (kg/cm^2)

Table 3.27.4 Performance Monitoring Datasheets: No.3

			Date:
Equipment # :			Time:
Item	Observations	Comments	
Primary Gas Filter DP (kg/cm^2)			
Primary Gas Supply DP			
Prim. Vent Flow Suct. (Nm^3/hr)			
Prim. Vent Flow Disch. (Nm^3/hr)			
Sec. Gas Filter DP (kg/cm^2)			
Sec. Gas Supply Press. (kg/cm^2)			
Sec. Supply Flow Suct. (Nm^3/hr)			
Sec. Supply Flow Disch. (Nm^3/hr)			
Sec. Vent Suct. Flow (Nm^3/hr)			
Sec. Vent Disch. Flow (Nm^3/hr)			
Oil in Sec. Drain Suct.?			
Oil in Sec. Drain Disch.?			
Separation Gas Filter DP DP			

Table 3.27.5 Performance Monitoring Datasheets: No.4

	Equip #:			Date
	System Name:			Time
Component/ Item	Specified Value	Actual Value	Comments	
Oil Reservoir				
Level				
Oil Temp. (C)				
Air in Oil? (Y/N)				
Gas in Oil?				
Oil Sample?				
Other				
Other				
Pumps				
Aux. Pump Operating?				
P2 (kg/cm^2)				
Suction Noise?				
Suction Filter $\triangle P$ (kg/cm^2)				
Vibration (μm)				
Brg. Bracket Temp. (C)				
Other				
Other				
Couplings				
Noise?				
Strobe Findings				
Other				
Other				
Turbine Driver				
Operating Speed (RPM)				
Trip Speed Setpoint (RPM)				
Vibration (μm)				
Brg. Bracket Temp. (C)				
Gov. Hunting?				

Table 3.27.5 Performance Monitoring Datasheets: No.4—Cont'd
Trip Lever Condition
Gov. Oil Condition
Other
Other
Motor Driver
Operating?
Vibration (μm)
Brg. Bracket Temp. (C)
Axial Shaft Movement (μm)
Fan Noise?
Other
Other
Relief Valves
Passing?
Set Pressure (kg/cm^2)
Pump P2 Press. (kg/cm^2)
Other
Other
Check Valves
Aux. Pump Turning Backwards?
Noise?
Other
Other
Back Pressure Valve
% Open
Stable?
Valve Noise?
Set Pressure (kg/cm^2)
Maintained Pressure (kg/cm^2)
Other
Other
(Continued)

Table 3.27.5 Performance Monitoring Datasheets: No.4—Cont'd
Transfer Valves
One Bank Operating?
Noise?
Other
Other
Coolers
$\triangle$T Oil
CW Valve Pos.
Cooler Operating?
Vent Valves Open?
Other
Other
TCVs
% Open
Set Temp. (C)
Stable?
Actual Temp. (C)
Other
Other
Filters
$\triangle$P (kg/cm^2)
Vent Valves Open?
Last Filter Change
Other
Other
Accumulators
Pre-charged Pressure (kg/cm^2)
Last PM Date
Other
Other

Table 3.27.5 Performance Monitoring Datasheets: No.4—Cont'd
Lube Oil PCV
% Open
Set Pressure (kg/cm^2)
Actual Pressure (kg/cm^2)
Stable?
Other
Other
Control Oil PCV
% Open
Set Pressure (kg/cm^2)
Acual Pressure (kg/cm^2)
Stable?
Other
Other
Lube Oil Rundown Tank (or Emerg. Pump)
Pump or Tank?
Pump Operating?
Tank Overflow
Other
Other
Lube Oil Supply Lines
Leaks?
Noise?
Vibration (μm)
Other
Other

Best Practice 3.28

Re-rates – consider the drive system (transmission and driver power limitations) and suction drum limitations to optimize reliability and production rates.

Confirm sufficient driver, coupling, gear (if applicable) power/torque capability.

Confirm that the suction drum entrainment velocity limit is not exceeded.

Exceeding the entrainment velocity limit can result in compressor fouling and/or suction drum demister deterioration and damage.

Lessons Learned

Failure to consider sufficient drive system capability has resulted in less than desired re-rate capacity, based on power limitations and in some cases, hydraulic coupling slippage requiring rotor and coupling replacement.

Failure to check suction drum for re-rate maximum flow conditions has resulted in:

- Fouling
- Liquid entrainment damage
- Suction drum demister deterioration
- Liquid carryover into the compressor

Benchmarks

This best practice has been used in all re-rate projects performed by FAI since the mid-1980s to ensure trouble free re-rate start-ups, and compressor reliabilities exceeding 99.7%.

Gear and Coupling Best Practices

4

Gears

Gears have a greater potential for low reliability than other types of rotating equipment, because they use gear meshes, and a minimum of four bearings to transmit torque and change the speed of the driven equipment. Gears typically have thousands of gear teeth meshing per second, and require proper alignment and lubrication to achieve optimum reliability. Figure 4.1 below is a plot of the number of gear meshes per second as a function of pitch line velocity and the number of teeth per inch of pitch diameter.

Couplings

Couplings in current use are normally dry, non-lubricated types that use discs or diaphragms to transmit torque and allow misalignment and axial movement.

This chapter will present gear and coupling best practices used since the 1970s.

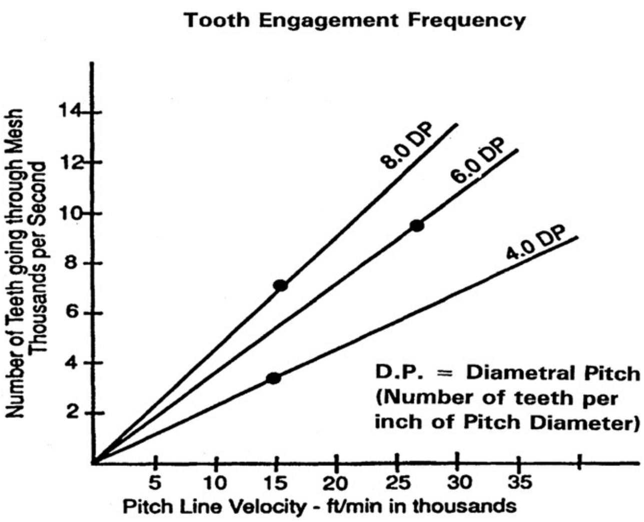


Fig 4.1 • Tooth engagement frequency

Best Practice 4.1

Do not use planetary gears for critical applications; use parallel shaft gears for optimum reliability and lower MTTR.

Planetary or epicyclic gears are maintenance-intensive and have higher mean time to repair (MTTR) than parallel shaft gears.

Planetary gears are more complex in design, having multiple gears and bearings, and consequently are more difficult to monitor than parallel shaft gears.

Always require parallel shaft gears for critical (un-spared) applications.

Lessons Learned

Planetary gears are quoted to minimize baseplate size since they enable the input and output shafts to be in the same axial plane. The design for this requirement requires many additional gears and bearings which can result in lower reliability and MTTR that is at least twice as long.

Benchmarks

This best practice has been used since the early 1980s for all applications, and has resulted in gear reliabilities higher than 99.5%.

B.P. 4.1. Supporting Material

Figure 4.1.1 shows the type of parallel shaft gear preferred for ease of maintenance and minimum components.

Parallel axes are the most common gear arrangement, consisting of a meshing pinion and gear. Parallel arrangements can be simple, or compounded with other parallel gear sets to obtain high gear ratios. The conventional way to describe such a gear is 'double increaser' or 'reduction' or 'triple increaser' or

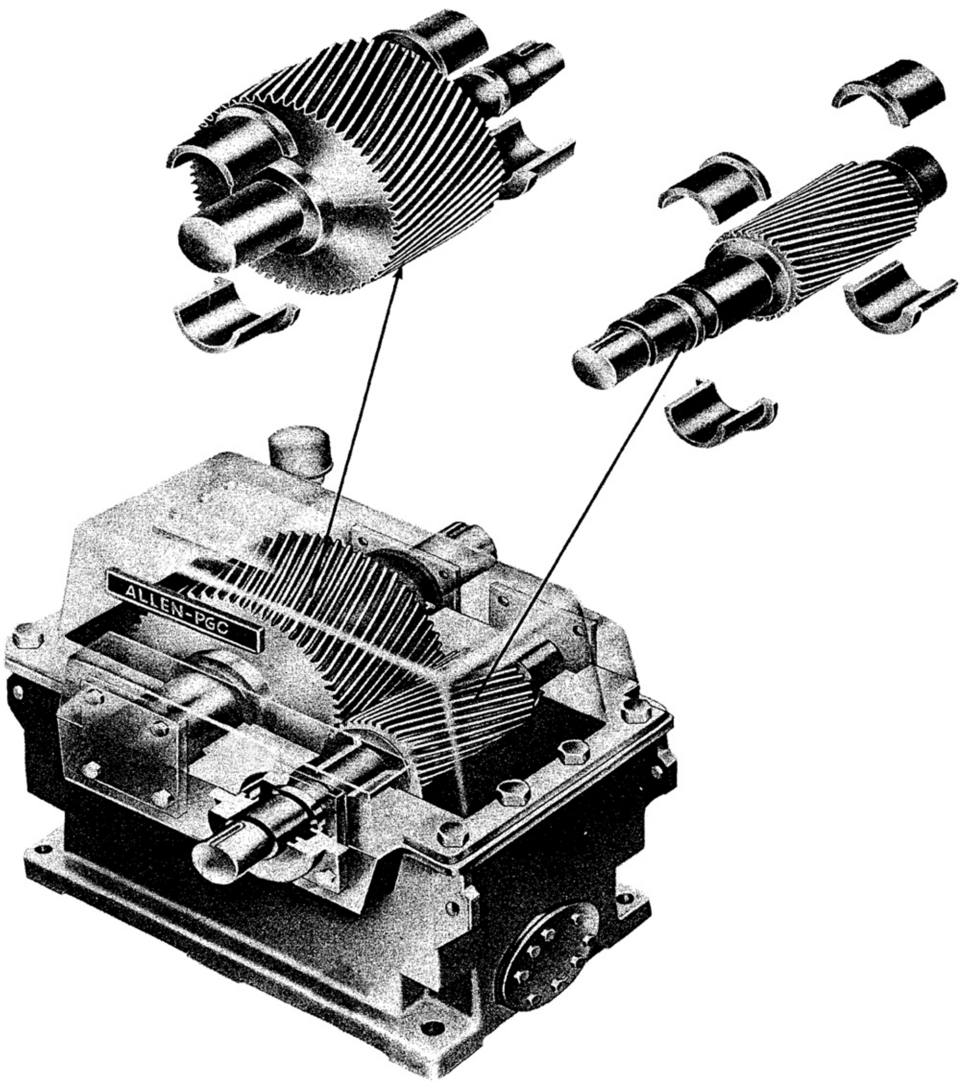


Fig 4.1.1 • Parallel shaft gear (Courtesy of Allen-PGC)

'reduction gear'. Parallel axis gears can use spur, helical, double helical or herringbone elements.

Figure 4.1.2 presents facts concerning single parallel axis arrangements.

Type	Spur	Single helical	Double helical (herringbone)
Thrust Load	Very low	High	Very low
Max Ratio	8:1	10:1	10:1
Max Power	3,725 kW (5,000 H.P.)	22,500 kW (30,000 H.P.)	22,500 kW (30,000 H.P.)
Applications	Low speed increaser or reducer pump drives	High speed pump, compressor or low speed generator	High speed pump, compressor or low speed generator

Fig 4.1.2 • Single parallel axis gear trains

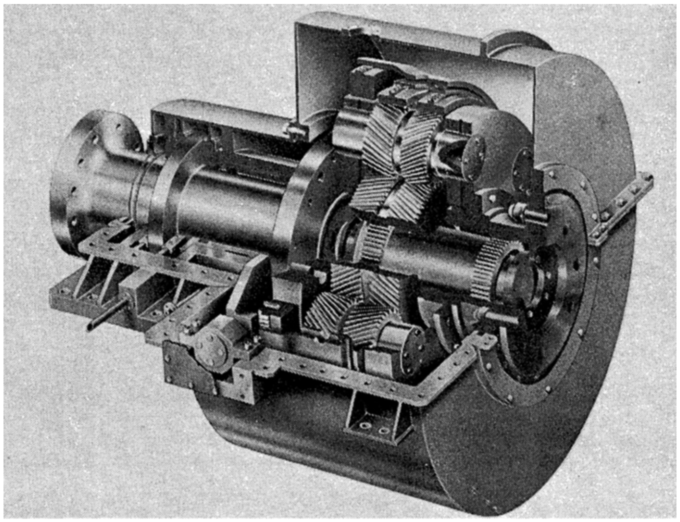


Fig 4.1.4 • Diesel propulsion drives up to 27,000 hp (Courtesy of Allen-PGC)

Parallel axis gear design reliability factors to optimize gear train field reliability are shown in Figure 4.1.3.

1. Ensure uniform face width tooth loading
2. Compensate for torsion (windup) of low stiffness shafts
3. Equalize torque transmission through each shaft (multiple shaft design)
4. Consider ease of assembly/disassembly
5. Minimize axial thrust (helix angle selection)
6. Tooth hardness considerations
7. Limit face width to pitch (L/D) diameters to proven values

Fig 4.1.3 • Parallel axis reliability factors

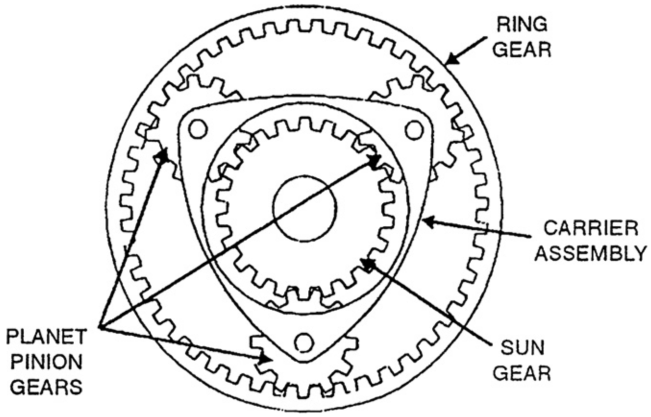


Fig 4.1.5 • Simple planetary arrangement

Figure 4.1.4 shows a planetary or epicyclic gear that has multiple gear meshes and bearings. This type of gear has greater mean time to repair and exposure to lower reliability than parallel shaft gears owing to its larger number of component parts.

Planetary gear units (also known as epicyclic), are widely used. Their principal advantage is that they allow both the input and output axes to be concentric, thus providing a very compact gear arrangement. Their principle uses are for aircraft prop engine drives, automotive and truck transmissions, power generation units and some critical equipment (pump and compressor) drives. Their main disadvantage is that they require more assembly and disassembly time than a conventional parallel shaft gear. Like conventional parallel shaft gears, they can be configured in simple and compound arrangements depending upon the required gear ratio. A simple planetary gear arrangement is shown in Figure 4.1.5.

The figure shows that planetary gears are similar to the solar system. The planet pinion gears each turn on their own axis, while they rotate around the sun gear. The planet pinions mesh with the internal tooth ring gear. All gears in the assembly are in constant mesh. The planet pinion bearings are in the carrier assembly. When power is applied to any one of the members, the entire assembly will rotate. If one of the other members is restrained the other member will provide the output. If none of the members are restrained the assembly will be in a neutral situation. Figures 4.1.6 A and B show side views of simple and compound planetary gear units respectively.

The simple, planetary or epicyclic train consists only of sun, planet and ring gear elements. The number of planets can vary considerably. Both simple and compound planetary gears can be designed with a number of variations. Figures 4.1.7 A and B present a simple planetary configuration with the ring gear restrained and the sun gear driving (4.1.7A) and the carrier assembly driving (4.1.7B).

If power is applied to one member of the planetary system, and a brake mechanism is applied to prevent rotation of a second member, the third member will rotate and become the output.

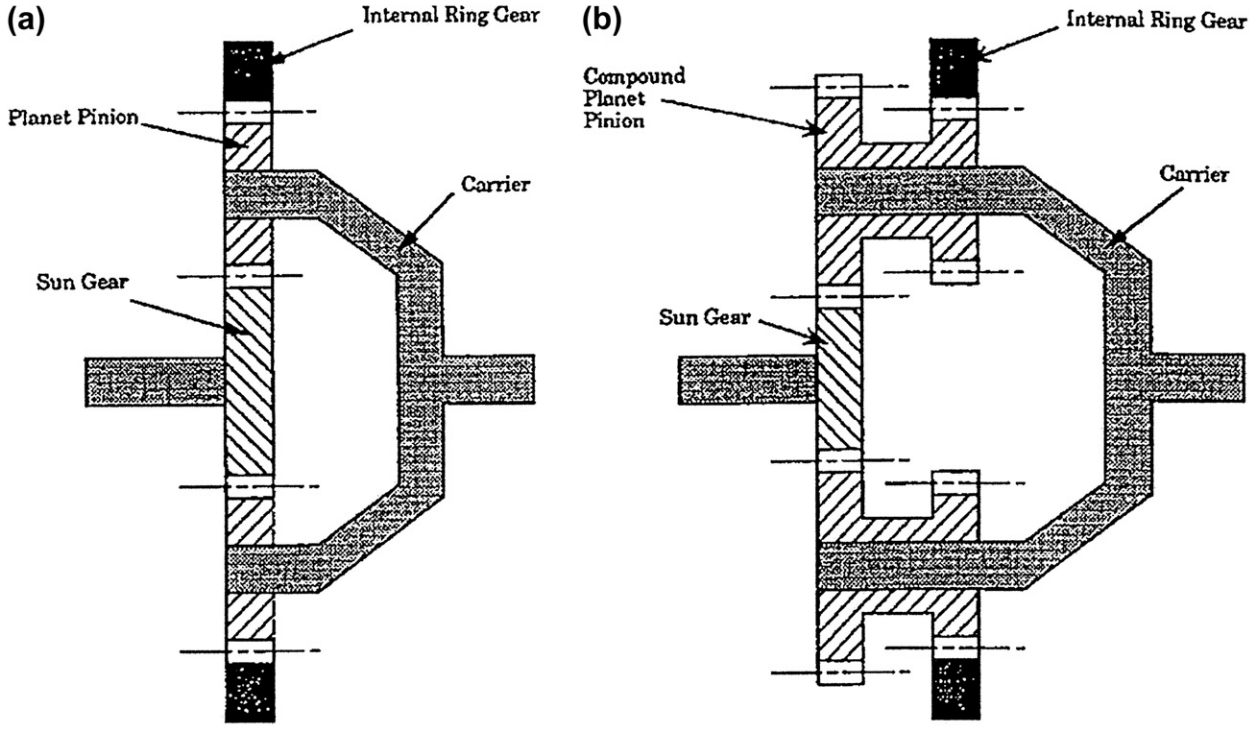


Fig 4.1.6 • Simple and compound planetary gear units

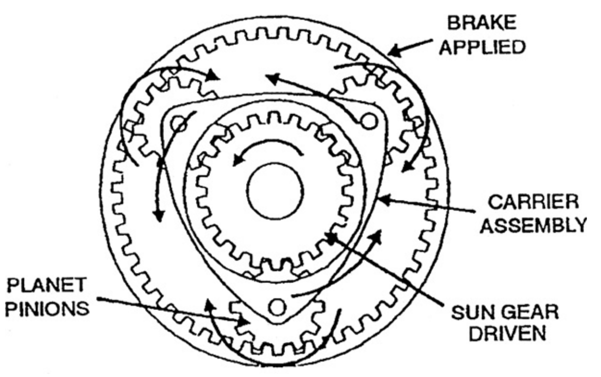


Fig 4.1.7 a) • Ring gear fixed sun gear driving (reduction)

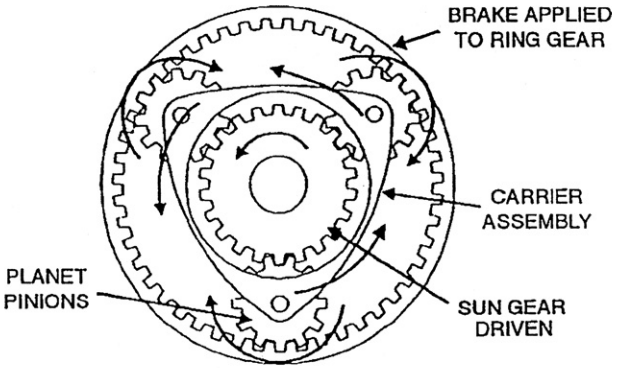


Fig 4.1.7 b) • Ring gear fixed carrier driving (speed increase)

In Figure 4.1.7A, the sun gear is driven, the ring gear is restrained and the carrier gear will rotate in the same direction of the sun but at a lower speed. In Figure 4.1.7B the carrier is driven, the ring gear is again restrained and the sun gear will rotate in the same direction but at a higher speed.

Figure 4.1.8 gives a summary of planetary gear combinations. Table 4.1.1 presents data concerning simple planetary arrangements, and Table 4.1.2 presents data concerning compound planetary arrangements.

	DRIVER	DRIVEN	HELD	DIRECTION OF ROTATION	OUTPUT SPEED
#1	SUN	CARRIER	RING	SAME	REDUCTION
#2	RING	CARRIER	SUN	SAME	REDUCTION
#3	SUN	RING	CARRIER	REVERSE	REDUCTION
#4	CARRIER	RING	SUN	SAME	INCREASE
#5	CARRIER	SUN	RING	SAME	INCREASE
#6	RING	SUN	CARRIER	REVERSE	INCREASE
#7	SUN, RING OR CARRIER	SUN, RING OR CARRIER	TWO MEMBERS ARE LOCKED	SAME	DIRECT DRIVE
#8	NOTHING IS HELD AGAINST ROTATION AND NO UNITS ARE LOCKED TOGETHER				NEUTRAL

Fig 4.1.8 • Summary of planetary gear combinations

Table 4.1.1 Simple planetary data

Kind of arrangement	Fixed member	Input member	Output member	Overall ratio	Range of ratios normally used
Planetary	Ring	Sun	Cage	$\frac{T_3}{T_2}+1$	3:1-12:1
Star	Cage	Sun	Ring	$\frac{T_3}{T_2}$	2:1-11:1
Solar	Sun	Ring	Cage	$\frac{T_2}{T_3}+1$	1.2:1-1.7:1

Key
T_1 = number of planet teeth
T_2 = number of sun teeth
T_3 = number of ring teeth

Table 4.1.2 Compound planetary data

Kind of arrangement	Fixed member	Input member	Output member	Overall ratio	Range of ratios normally used
Compound planetary	Ring	Sun	Cage	$\frac{T_3}{T_2}\frac{T_{11}}{T_{12}}+1$	6:1-25:1
Compound star	Cage	Sun	Ring	$\frac{T_{11}}{T_2}\frac{T_3}{T_{12}}+1$	5:1-24:1
Compound solar	Sun	Ring	Cage	$\frac{T_2}{T_3}\frac{T_{12}}{T_{11}}+1$	1.05:1-2.2:1

Key
T_2 = number of sun teeth
T_3 = number of ring teeth
T_{11} = number of first-reduction planet teeth
T_{12} = number of second-reduction planet teeth

The key reliability factors concerning planetary gears are presented in Figure 4.1.9.

1. Complicated assembly
2. Equal planet load distribution
3. High planet shaft loads
4. Rotating ring gear balance and vibration
5. Gear assembly seizure if foreign objects enter mesh

Fig 4.1.9 • Planetary gear train reliability factors

Best Practice 4.2

Design audit gears when pitch line velocities exceed 6,000 meters/min (20,000 ft/min) to ensure optimum safety and reliability.

Review all vendor pitch line velocity experience in the engineering phase of the project before acceptance.

If a minimum of 2 year field operating experience for gears with a higher pitch line velocity is not shown, design audit the proposed gear box for preventive measures to prevent gearbox flooding.

Gearbox flooding (drain oil contacting the gear teeth) can cause catastrophic gearbox failure.

Lessons Learned

I have been involved with severe gearbox flooding issues during factory acceptance tests, at pitch line velocities greater than 6,000 m/min (20,000 ft/min) that required gear box internal redesign.

Redesign of gearbox internals, if possible, can take a long time, and may require a larger gearbox which will in turn necessitate baseplate redesign.

Benchmarks

This best practice has been used since the mid 1970s to ensure trouble-free operation and maximum gearbox reliability (above 99.5%).

Best Practice 4.3

Ensure proper anti-friction bearing selection (life and DN) for low speed, high torque gear applications.

Confirm the B-10 bearing life calculations with the vendors during the quoting phase of the project.

Confirm vendor D-N (bore diameter times operating speed) for all bearings, and especially for indeterminate (3 bearing) systems.

Carefully evaluate low loaded bearings in three bearing systems and review vendor bearing load experience to eliminate the possibility of bearing skidding.

Lessons Learned

Complete gearbox damage has been caused by the use of high DN, low loaded anti-friction bearings in indeterminate bearing systems in multi-stage, high torque reduction gears.

Benchmarks

This best practice has been used since the mid 1980s for all low speed gear applications using anti-friction bearings.

B.P. 4.3. Supporting Material

Anti-friction bearings

Anti-friction radial bearings support the rotor, by using rolling elements to reduce friction losses. They are used in low horsepower applications (below 375 kW (500 H.P.)), high torque gear boxes and in aero-derivative gas turbines. Examples of roller and ball type anti-friction bearings are shown in Figure 4.3.1.

As previously mentioned, all bearings are designed to have sufficient bearing area to support all the forces acting on the bearing.

That is:

$$P = \frac{F}{A}$$

Where: P = Pressure on the bearing elements
F = The total of all static and dynamic forces acting on the bearing
A = Contact area

For anti-friction bearing applications, the pressure, P is the point contact or 'Hertzian' stress on the bearing elements and rings or 'races'. For an anti-friction bearing to be properly designed, its D-N number and bearing life must be determined. Figure 4.3.2 presents the definition of D-N number and its uses.

Each type of anti-friction bearing has a maximum operating D-N number. If this value is exceeded, rapid bearing failure can occur. In addition, D-N numbers are typically used to determine the type of lubrication required for bearings. A common practice in the turbo-machinery industry has been to use hydrodynamic pressurized bearings when the D-N number exceeds approximately 500,000.

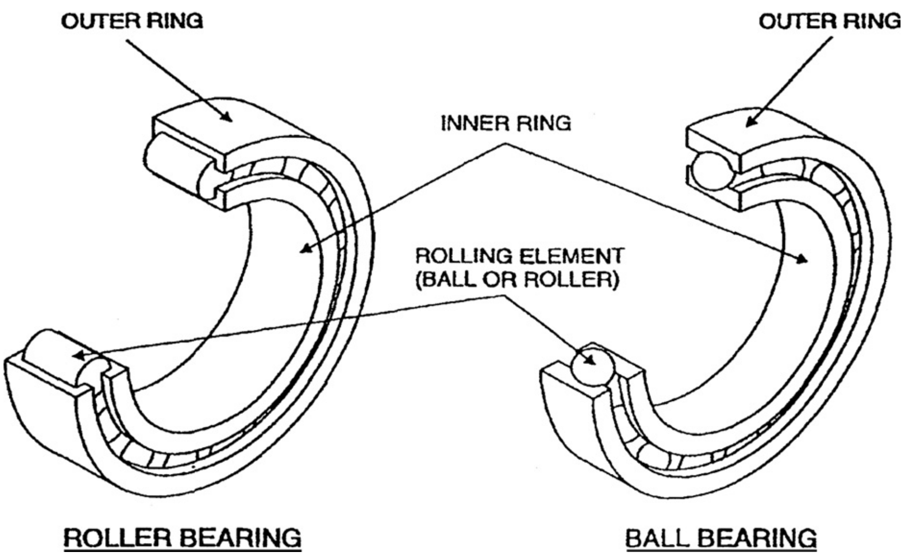


Fig 4.3.1 • Anti-friction bearings

The D-N number is a measure of the rotational speed of the anti-friction bearing elements

D-N number = bearing bore (millimeters) × speed (RPM)

Approximate lubrication ranges

Lubrication type	**D-N Range**
Sealed	below 100,000
Regreaseable	100,000–300,000
Oil lube (unpressurized)	300,000–500,000
Oil lube (pressurized)	above 500,000

Fig 4.3.2 • D-N number

'B-' or 'L-' 10 life is defined as the life in hours that 9 out of 10 randomly selected bearings would exceed in a specific application.

$$\text{'B-' or 'L-' 10 life} = \frac{16700}{N}\left[\frac{C}{F}\right]^3$$

Where: N = RPM

C = Load in LBS that will result in a bearing element life of 1,000,000 revolutions

F = Actual load in LBS

Fig 4.3.3 • 'B-' or 'L-' 10 life

The exception to this rule is aircraft gas turbines, which can have D-N numbers in excess of 2,000,000. In these applications, hydrodynamic bearings are not used, since the size and weight of the required lubrication system would be prohibitive.

Anti-friction bearings have a finite life which is usually specified as 'B-10' or 'L-10' life; as defined in Figure 4.3.3.

An important fact to note is that the life of any anti-friction bearing is inversely proportional to cube of the bearing loads. As a result, a small change in the journal bearing loads can significantly reduce the bearing life. When anti-friction bearings suddenly start failing where they did not in the past, investigate all possible sources of increased bearing loads (piping forces, foundation forces, misalignment, unbalance, etc.). Anti-friction bearings are usually designed for a minimum life of 25,000 hours' continuous operation.

Best Practice 4.4

Correct un-loaded gearbox vibration problems, which typically occur during start-up, by limiting bypass (spill-back) valve* stroke to partially load the gear radial bearings.

90% or more of radial bearing loading in gearboxes is produced by the transmitted torque (power/speed).

Since the radial bearing area is designed for the maximum transmitted torque, start-up loads may not produce sufficient force and corresponding oil wedge to stabilize the gear rotors.

Limiting the opening of the bypass valves will increase the gearbox load and stabilize the gear rotors to reduce vibration during low load (start-up conditions). In forced draft (FD) and induced draft (ID) fan applications, it may be necessary to increase inlet or outlet damper.

Lessons Learned

Many compressor train gears experience high vibration during start-up, and in most cases the cause can be traced to oversized bypass valves that are operated fully open.

Benchmarks

This best practice has been used since 1990 to correct high gearbox vibrations during start-up, and eliminate plant start-up delays.

*Increase damper opening for FD and ID fans.

B.P. 4.4 Supporting Material

Gear reaction (bearing) forces

When considering reaction forces, one must consider the entire gear system, from the gear mesh to the gear foundation. The transmission of torque load through the gear rotors is shown in Figure 4.4.1, assuming a speed increaser.

The amount of torque that is transmitted depends on the operating conditions (start-up, rated load, off-design load, shutdown, etc.).

Figure 4.4.2 shows a typical compressor torque vs. speed curve.

Note that the start-up condition is always at low load, and frequently the shutdown condition will be at low load (if case is vented on shutdown). This is an important fact to consider when gear vibration and/or noise are observed at start-up, shutdown or under off-design conditions. Figure 4.4.3 presents this important consideration.

Since the transmitted torque loads will be considerably less, the gear reaction forces will be considerably less and the component stresses and pressures will be less. This is exactly why gear meshes are noisy at start-up, when vibration increases and bearings can become unstable.

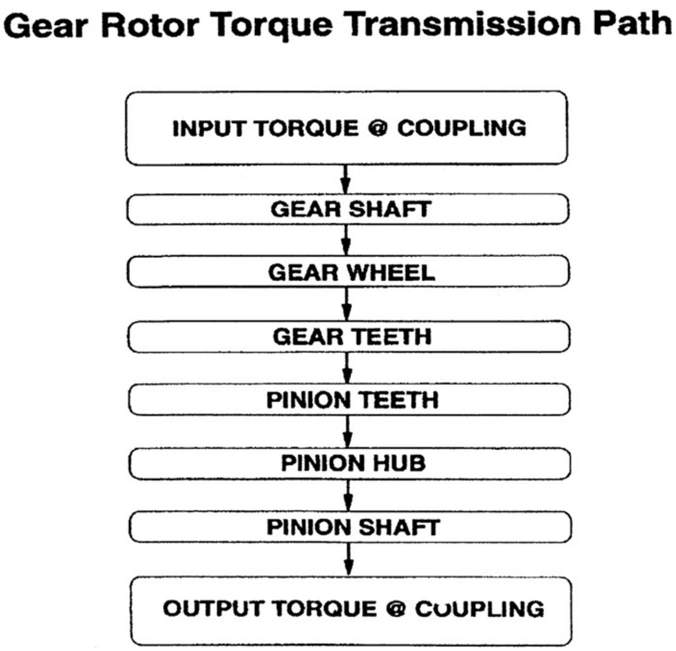


Fig 4.4.1 • Gear rotor torque transmission path

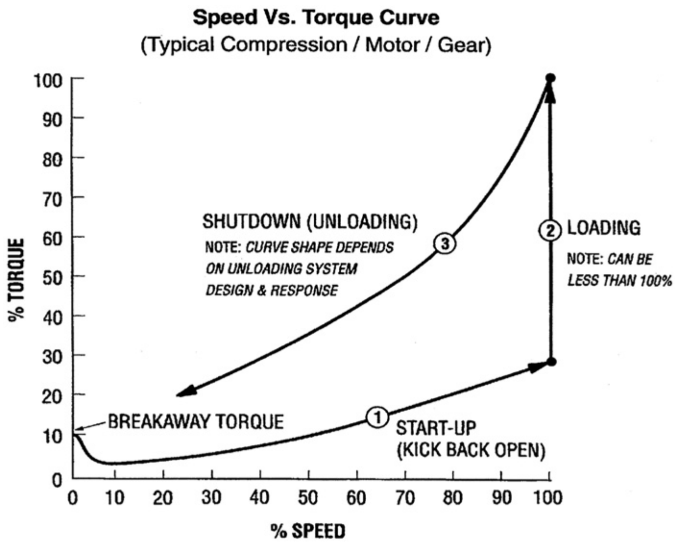


Fig 4.4.2 • Speed vs. torque curve

All gear unit component stresses (pressures)

$$\frac{\text{FORCE}}{\text{AREA}}$$

are designed for rate (maximum) torque loads. Therefore, loads during off-design conditions can be considerably less.

Fig 4.4.3 • Gear unit design basis for reaction forces

Gear Radial Bearing Force Transmission Path

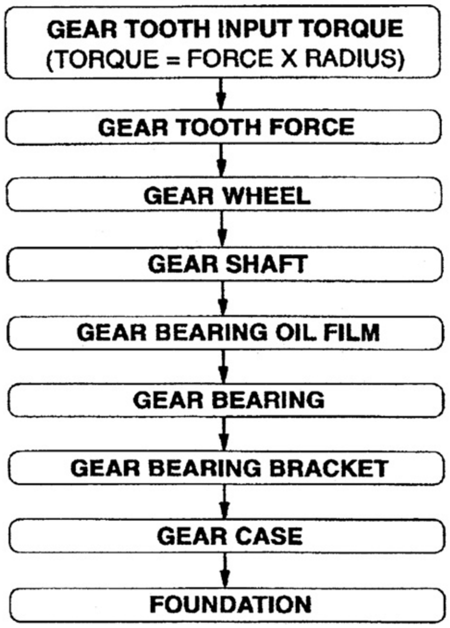
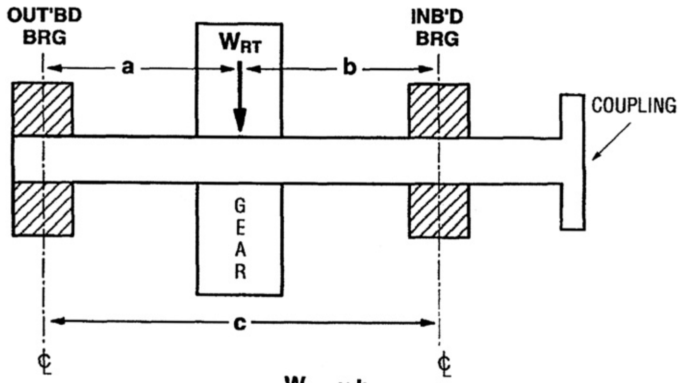


Fig 4.4.4 • Gear radial bearing force transmission path

Determination of Radial Bearing Loads

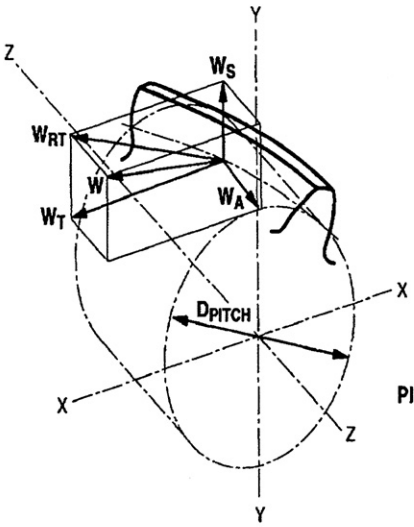


$$W_{RT-INB'D} = \frac{W_{RT} \times b}{c}$$

$$W_{RT-OUT'BD} = \frac{W_{RT} \times a}{c}$$

Fig 4.4.6 • Determination of radial bearing loads

Helical Pinion Tooth Reactions at Pitch Diameter



WHERE:

W_T = Tangential Force (calculated)
W_S = Separating Force
W_A = Axial Force
W_{RT} = Total Radial Force
W = Total Tooth & Bearing Load

$$W_T = \frac{2 \times \text{Pinion Torque}}{\text{Pinion Pitch Diameter}}$$

$$\text{PINION TORQUE} = \frac{\text{Horsepower} \times \frac{63{,}025 \text{ in.-lbs.}}{\text{h.p.}}}{\text{Pinion Speed – rpm}}$$

$$W_S = (W_T) \bullet \left[\frac{\tan(\text{normal Pressure Angle})}{\cos(\text{Helix Angle})}\right]$$

$$WR_T = \sqrt{(W_S)^2 + (W_T)^2}$$

$$W_A = (W_T) \bullet \left[\tan(\text{Helix Angle})\right]$$

$$W = \sqrt{(W_{RT})^2 + (W_A)^2}$$

Fig 4.4.5 • Helical pinion tooth reactions at pitch diameter

Figure 4.4.4 shows the reaction force transmission path of a gear radial bearing. The entire gear unit system contributes to the support of transmitted loads. A change in the load carrying capability of any of the items noted in the figure can result in reduced gear unit reliability.

Gear reaction forces at bearings

Since between-bearing helical gears are the most common type on site, only this type will be covered. However, the relations discussed, with minor modifications, will also apply to internal and external spur gears. Figure 4.4.5 shows the reaction forces that act on a helical pinion tooth.

Once the total radial load, W_{RT}, is known, the individual radial bearing forces can be determined by statics mechanics; as shown in Figure 4.4.6.

The axial load, W_A, is calculated directly as shown in Figure 4.4.5, and can be applied to either the gear shaft, pinion shaft, or divided between the gear and pinion shaft. Most gear designs absorb all thrust on the gear shaft (low speed), since this usually results in the lowest thrust bearing losses.

Best Practice 4.5

Monitor the gear centerline shaft position to condition monitor radial bearing wear and load (attitude) angle.

Gearbox radial bearings are the highest loaded bearings in any type of machinery and approach the limits of oil film pressure (3500kpa or 500psi).

If the area of the radial bearings is increased, bearing instability will occur during low load at start-up (see B.P. 4.4).

Therefore, bearing life is limited, and is lower than radial bearings in other applications (pumps, compressors, turbines, etc.).

Shaft centerline position is monitored by two proximity probes that are mounted at each radial bearing. They will record the position of the shaft and therefore the load angle of the shaft in the bearing. The load angle in gear applications changes with transmitted torque (power/speed).

Trending shaft centerline position will enable condition monitoring of bearing wear (increasing shaft centerline position) and determination of the position of the load angle.

When the load angle lies in the major axis of an elliptical (lemon bore or offset sleeve bearing) vibration, instabilities can occur (oil whirl or oil whip).

Lessons Learned

Critical integral gear compressors have required emergency shutdown due to excessive radial bearing wear that did not cause high levels of vibration prior to shutdown (high vibration occurred when Babbitt material was excessively worn).

Heavily loaded radial bearings do not exhibit high vibration, and can go undetected if shaft position is not monitored, and bearing pad temperature probes are not located at the load point of the bearing.

Benchmarks

This best practice has been used since the 1980s to optimize highly loaded gear radial bearing life, by predicting and recommending machine shutdowns at convenient times, thus eliminating costly emergency shutdowns.

B.P. 4.5. Supporting Material

Please refer to material in B.P. 4.4 in addition to the following:

Hydrodynamic bearing types

Regardless of the type of hydrodynamic bearing, all bearing surfaces are lined with a soft, surface material made from a composition of tin and lead. This material is known as Babbitt. Its melting temperature is above 200°C (400°F), but under load will begin to deform at approximately 160°C (320°F). Typical thickness of Babbitt over steel is 1.5mm (0.060 inches). Bearing embedded temperature probes are a most effective means of measuring bearing load point temperature, and are inserted just below the Babbitt surface. RTDs or thermocouples can be used. There are many modifications available to increase the load effectiveness of hydrodynamic bearings. Among the methods available are:

- Copper backed Babbitt or 'Trimetal' – to aid in heat removal
- Back pad cooling – used on tilt pad bearings to remove heat
- Direct cooling – directing cool oil to maximum load points

A typical straight sleeve hydrodynamic journal bearing is shown in Figure 4.5.1.

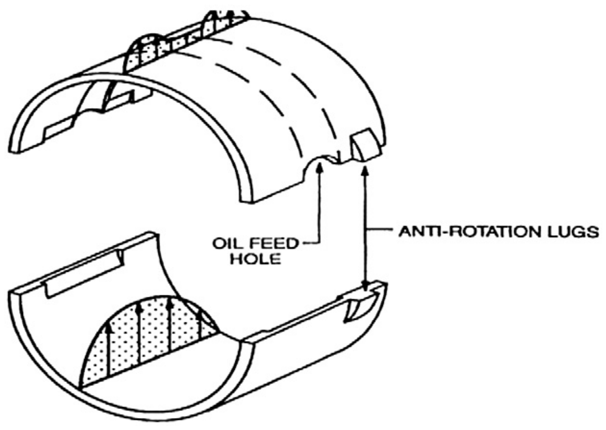


Fig 4.5.1 • Straight sleeve bearing liner (Courtesy of Elliott Co.)

Straight sleeve bearings are used for low shaft speeds (less than 5,000 rpm) or for older turbo-compressor designs. Frequently, they are modified to incorporate a pressure dam, in the direction of rotation. The pressure dam must be positioned in the top half of the bearing to increase the load vector. This action ensures that the tangential force vector will be small relative to the load vector, thus preventing shaft instability. It should be noted that incorrectly assembling the pressure dam in the lower half of the bearing would render this type of bearing unstable. When shaft speed is high, other alternatives to prevent rotor instabilities are noted in Figure 4.5.2.

Shown are examples of anti-whirl bearings. The most common types of these bearings are the three and four lobe design. Elliptical and offset bearing designs do prevent instabilities, but tend to increase shaft vibration if the load vector passes through the major axis of the bearing. These types of bearings may have to be rotated in the bearing brackets to prevent this occurrence.

The most common hydrodynamic bearing for higher speed applications is the tilt pad journal bearing shown in Figure 4.5.3. A tilting pad bearing offers the advantage of increased contact area, since the individual pads conform to the shaft orbit. In addition, this type is also a highly effective anti-whirl bearing, since the spaces between the pads prevent oil whirl. Most end users specify tilt pad radial and thrust bearings for turbo-compressor applications.

Figure 4.5.4 shows the mechanical frictional losses and oil flow requirements for a tilt pad journal bearing as a function of shaft speed.

Note that the basis for horsepower loss and oil flow is an oil temperature rise of 16.7°C (30°F). This is the normal design ΔT for all hydrodynamic bearings. Also given in this figure is the data necessary to calculate bearing pressure at the load point.

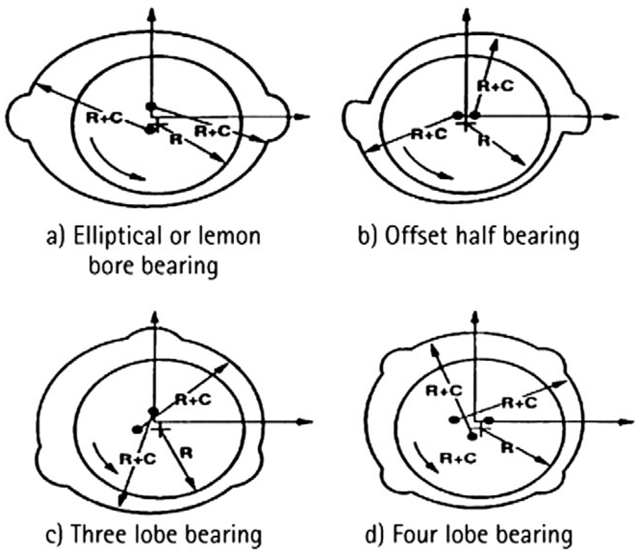


Fig 4.5.2 • Prevention of rotor instabilities

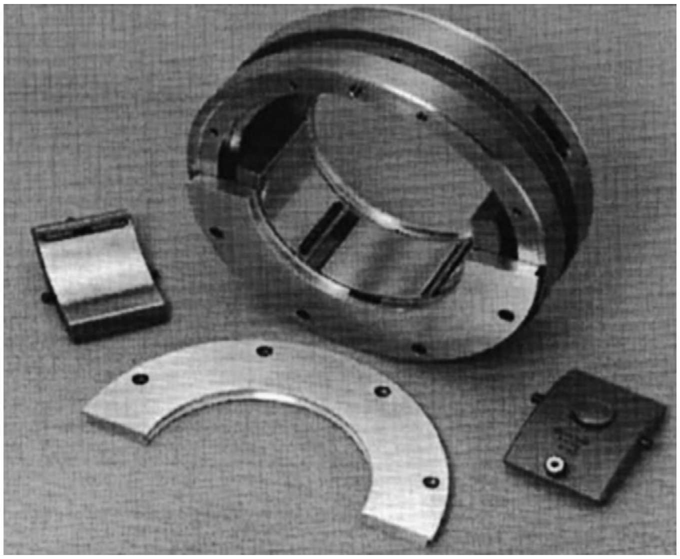


Fig 4.5.3 • Tilting pad journal bearing assembly (Courtesy of Kingsbury, Inc.)

As an exercise, calculate the following for this bearing:

Projected Area

$$A_{PROJECTED} = 5'' \times 2'' = 10 \text{ square inches}$$

Pressure

$$= 3479 \text{ lb force} \div 10 \text{ square inches} = 347.9 \text{ psi on the oil film at load point}$$

Condition monitoring

In order to determine the condition of any journal bearing, all the parameters that determine its condition must be monitored. Figure 4.5.5 presents the eight relevant parameters, along with typical limits. It is also advisable to consult the manufacturer's instruction book for vendor-recommended limits.

One frequently overlooked, but important parameter noted in Figure 4.5.5 is the shaft position. Change in shaft position can only occur if the forces acting on a bearing change, or if the bearing surface wears. Figure 4.5.6 shows how shaft position is determined using standard shaft proximity probes.

Regardless of the parameters that are condition monitored, relative change of condition determines if and when action is required. Therefore, effective condition monitoring requires the following action for each monitored condition:

- Establish baseline condition
- Record condition trend
- Establish condition limit

Figure 4.5.7 presents these facts for a typical hydrodynamic journal bearing.

Based on the information shown in this trend, the bearing should be inspected at the next scheduled shutdown. A change in parameters during month six has resulted in increased shaft position, vibration and bearing pad temperature.

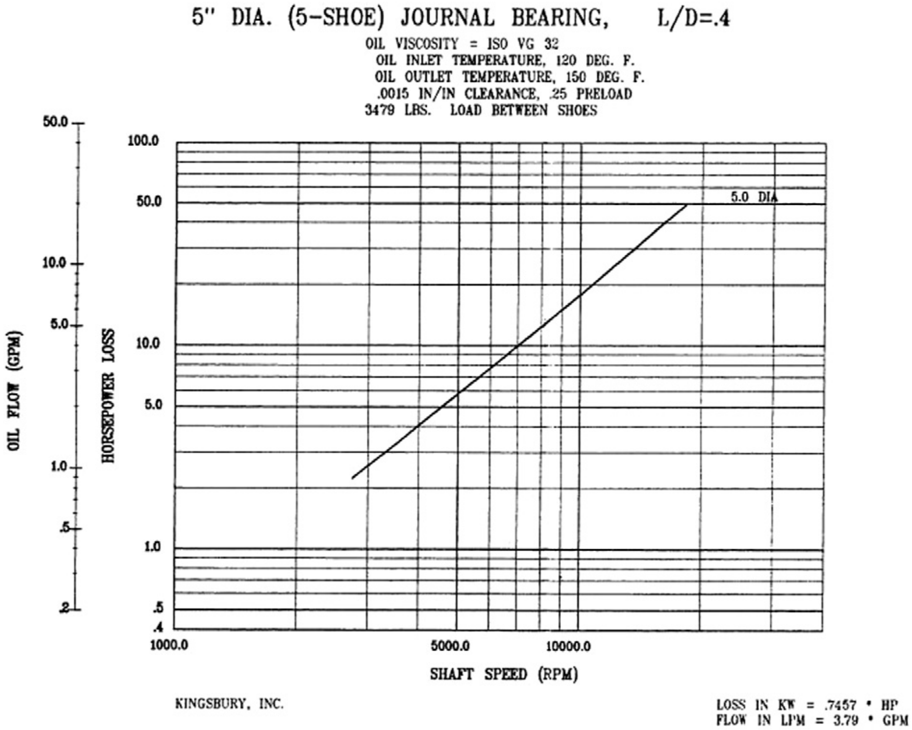


Fig 4.5.4 • Typical journal bearing selection curve (Courtesy of Kingsbury, Inc.)

Parameter	Limits
Radial vibration (peak to peak)	2.5 mils (60 microns)
Bearing pad temperature	220°F (108°C)
Radial shaft position (except for gearboxes where greater values are normal from unloaded to loaded operation)	> 30° change and/or 30% position change
Lube oil supply temperature	140°F (60°C)
Lube oil drain temperature	190°F (90°C)
Lube oil viscosity	Off spec 50%
Lube oil flash point	Below 200°F (100°C)
Lube oil particle size	Greater than 25 microns

Condition monitoring parameters and their alarm limits according to component:
1. Journal bearing (hydrodynamic)

Fig 4.5.5 • The eight parameters that determine the condition of a hydrodynamic journal bearing along with typical limits

Vibration instabilities

Vibration is an important condition associated with journal bearings, because it can provide a wealth of diagnostic information that can be very valuable in determining the root cause of a problem. Figure 4.5.8 presents important information concerning vibration.

Figure 4.5.9 defines excitation forces with examples that can cause rotor (shaft) vibration.

Turbo-compressors generally monitor shaft vibration relative to the bearing bracket using a non-contact or 'proximity probe' system as shown in Figure 4.5.10. The probe generates a D.C. eddy current which continuously measures the change in gap between the probe tip and the shaft. The result is that the peak to peak unfiltered (overall) shaft vibration is read in mils or thousandth of an inch. The D.C. signal is normally calibrated for 200 millivolts per mil. Probe gaps (distance between probe and shaft) are typically 1mm (0.040 mils) or 8 volts D.C. to ensure the calibration curve is in the linear range. It is important to remember that this system measures shaft vibration relative to the bearing bracket, and assumes the bearing bracket is fixed. Some systems incorporate an additional bearing bracket vibration monitor and thus record vibration relative to the earth or 'seismic vibration'.

As previously discussed, vibration limits are usually defined by:

$$\text{Vibration (mils p–p)} = \sqrt{\frac{12000}{RPM}}$$

This value represents the allowable shop acceptance level. API recommends alarm and trip shaft vibration levels be set as follows:

$$V_{ALARM} = \sqrt{\frac{24000}{RPM}}$$

$$V_{TRIP} = \sqrt{\frac{36000}{RPM}}$$

In my opinion, shaft vibration alarm and trip levels should be based on the following parameters as a minimum, and

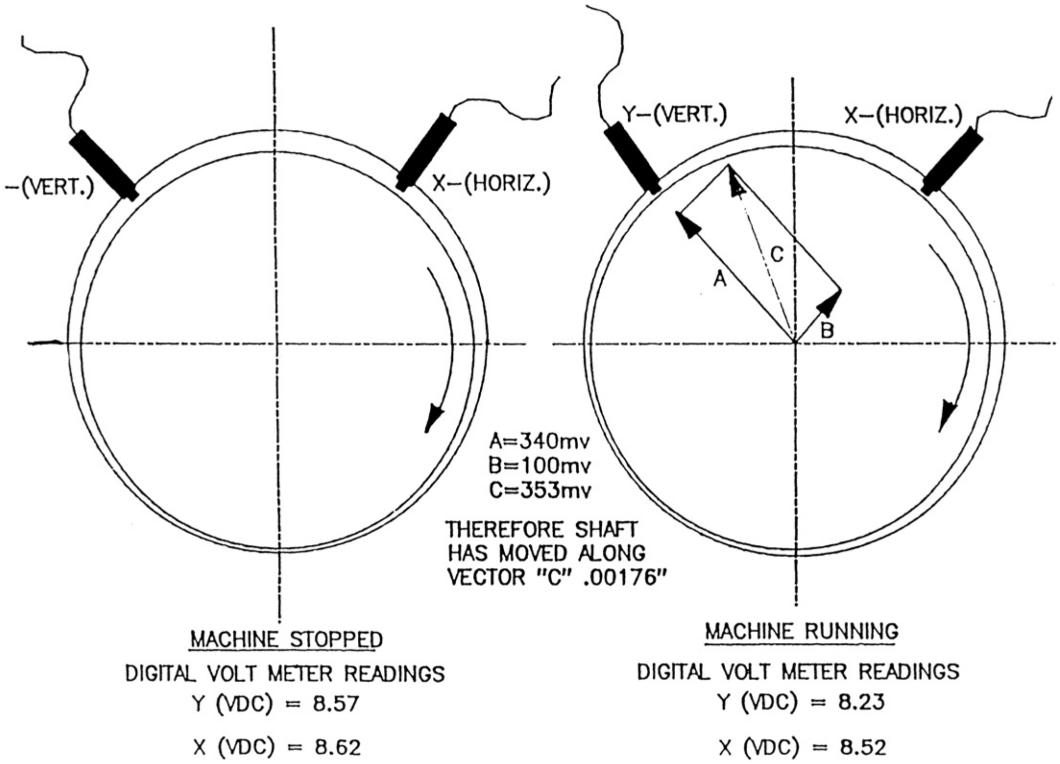


Fig 4.5.6 • Shaft movement analysis (relative to bearing bore) (Courtesy of M.E. Crane, Consultant)

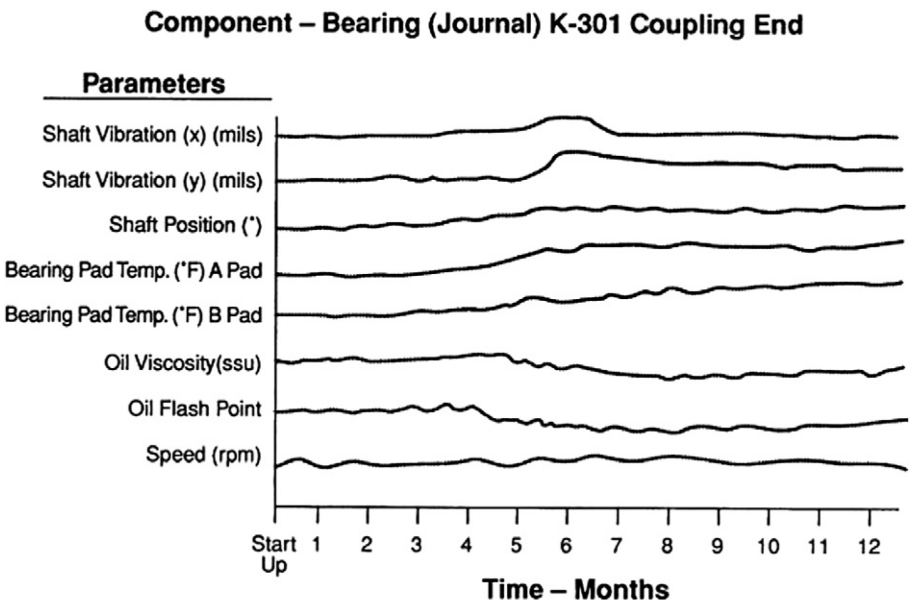


Fig 4.5.7 • Trending data for a typical hydrodynamic journal bearing

Vibration is the result of a system being acted on by an excitation.
This excitation produces a dynamic force by the relationship:

$F_{DYNAMIC} = Ma$

Where: M = Mass (Weight/g)

g = Acceleration due to gravity (386 IN/SEC^2)

a = Acceleration of mass M (IN/SEC^2)

Vibration can be:

Lateral _________ ↕ __________

Axial → ________________ ←

Torsional ____________________

Fig 4.5.8 • Vibration

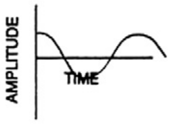
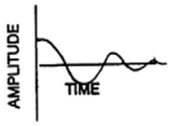
CATEGORY	EXAMPLES	EXCITATION TYPE	VIBRATION TYPE
FORCED VIBRATIONS 	UNBALANCE MISALIGNMENT PULSATION	CONSTANT CONSTANT PERIODIC	LATERAL LATERAL AND AXIAL TORSIONAL
TRANSIENT VIBRATIONS 	SYNCHRONOUS MOTOR START-UP IMPULSE (SHOCK FORCE)	RANDOM RANDOM	TORSIONAL TORSIONAL, AXIAL, RADIAL
SELF EXCITED	INTERNAL RUB OIL WHIRL GAS WHIRL	RANDOM CONSTANT CONSTANT	LATERAL LATERAL LATERAL

Fig 4.5.9 • Excitation forces with examples

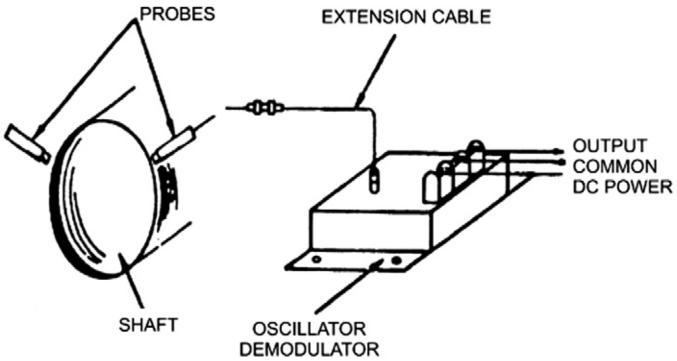


Fig 4.5.10 • Non contact displacement measuring system

should be discussed with the machinery vendor prior to establishing levels:

- Application (critical or general purpose)
- Potential loss of revenue
- Application characteristics (prone to fouling, liquid, unbalance, etc.)
- Bearing clearance
- Speed
- Rotor actual response (Bode Plot)
- Rotor mode shapes (at critical and operating speeds)

Figure 4.5.11 presents a vibration severity chart with recommended action, and a schematic of a shaft vibration and shaft displacement monitor are shown in Figure 4.5.12.

As mentioned above, vibration is measured unfiltered or presents ‘overall vibration’. Figure 4.5.13 shows a vibration signal in the unfiltered and filtered conditions. All vibration diagnostic work (troubleshooting) relies heavily on filtered vibration to supply valuable information to determine the root cause of the vibration.

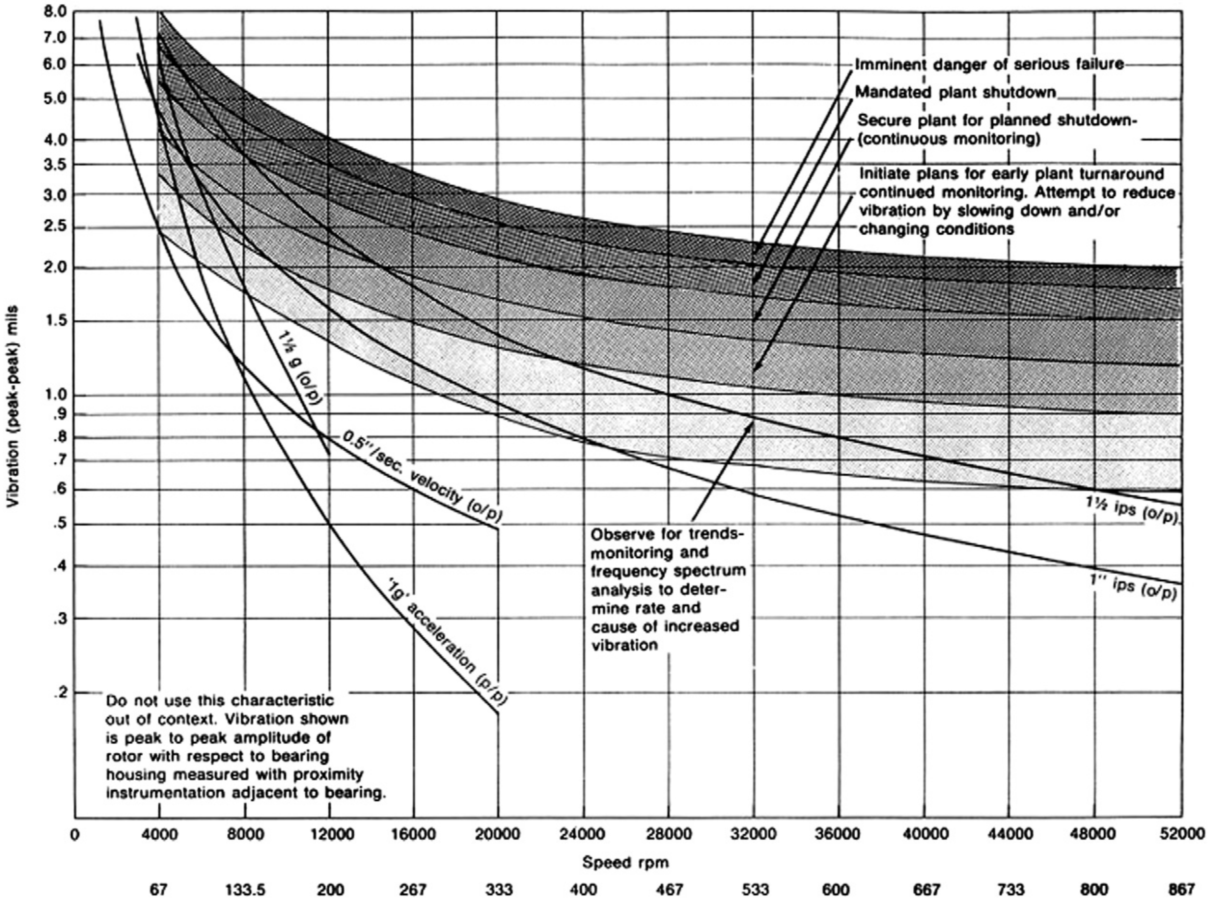


Fig 4.5.11 • Vibration severity chart (Courtesy of Dresser-Rand and C.J. Jackson, P.E.)

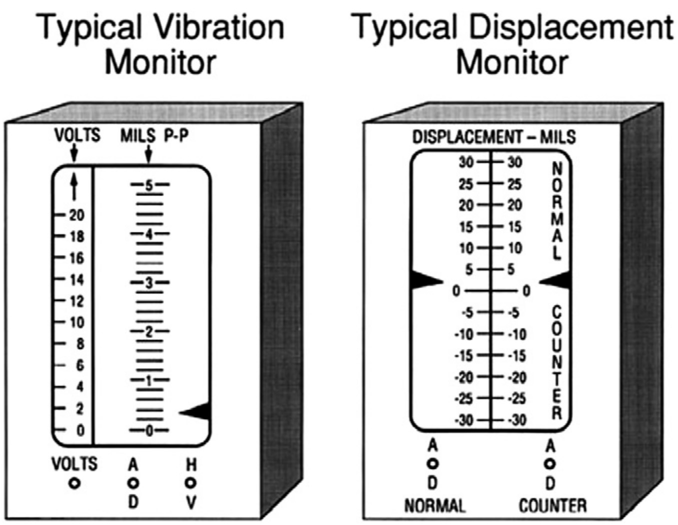
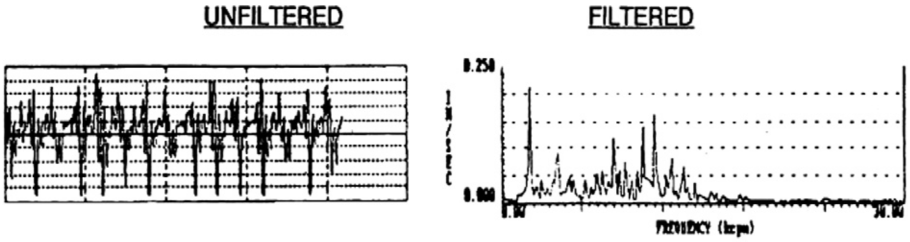


Fig 4.5.12 • Shaft vibration and displacement

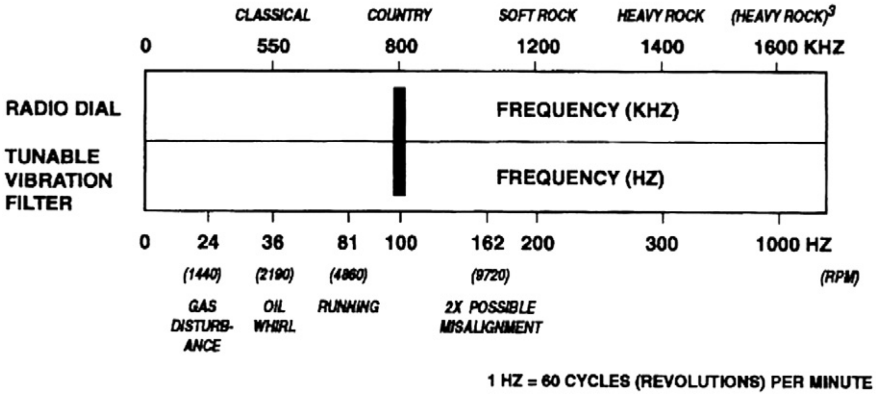
Figure 4.5.14 presents an example of a radio tuner as an analogy to a filtered vibration signal. By observing the predominant filtered frequencies in any overall (unfiltered) vibration signal, valuable information can be gained to add in the troubleshooting procedure and thus define the root cause of the problem.

ANY VIBRATION SIGNAL IS MADE UP OF ONE OR MORE FREQUENCIES. A TYPICAL UNFILTERED (OVERALL) AND FILTERED SIGNAL ARE SHOWN BELOW:



AN ANALOGY TO A FILTERED SIGNAL IS A RADIO. IN A GIVEN LOCALITY, MANY STATIONS ARE TRANSMITTING SIMULTANEOUSLY. ANY GIVEN STATION IS OBTAINED BY ADJUSTING THE TUNER TO THE CORRECT TRANSMISSION FREQUENCY.

Fig 4.5.13 • Vibration frequency



By adjusting the tuner (filter) to a selected station (frequency) the desired program (vibration frequency signal) can be obtained if it is "on the air" (present)

Fig 4.5.14 • Radio tuner/vibration filter analogy

Best Practice 4.6

Replace oil lubricated gear couplings that have experienced lock-up (high vibration causing bearing damage) with flexible disc couplings.

Oil lubricated gear couplings, used prior to the 1970s, can experience lock-up, since they will centrifuge their lubrication oil into the meshing teeth, which can render the coupling rigid.

Regardless of the filter size (even as low as 1 micron), lock-up can occur on large gear couplings and/or at high speeds.

Replacement of gear couplings can be accomplished successfully as long as all of the following items are considered and confirmed to be designed correctly with the new dry coupling:

- Torsional natural frequencies
- Lateral natural frequencies (critical speeds) and rotor response
- Coupling guard internal clearance to coupling hub flange OD

Lessons Learned

Many existing critical machine trains require a yearly shutdown to clean gear coupling teeth after high vibration shutdowns or bearing failures necessitate shutdown. Time for shutdown can exceed three days. Present daily revenues for critical (unspared) compressor trains can exceed $1 MM/day.

Benchmarks

This best practice has been used since the 1990s to remove the necessity of a yearly gear coupling PMs, and to prevent bearing damage resulting from coupling lock-up. The modification cost can be easily justified if yearly shutdowns are required for gear coupling cleaning.

B.P. 4.6. Supporting Material

The coupling function

The function of a flexible coupling is to transmit torque from the driver to the driven machine, while making allowances for minor shaft misalignment and shaft end position changes between the two machines. The design of the coupling should provide for transmission of the required torque at the required speed, with a minimum of extraneous forces and perturbations being exerted on either the driver or driven shaft. Shaft misalignment exists when the centerlines of two shafts that are joined by a coupling do not coincide. Figure 4.6.1 shows the various types of misalignment and shaft end position changes that can occur.

Each coupling type has a maximum tolerance of misalignment and axial position change that is noted on the coupling drawing. Regardless of coupling type, misalignment tolerance is stated in degrees and is usually ¼°. Axial position change tolerance varies with coupling type. Gear type couplings have a large axial position change tolerance compared to flexible element types.

Types

The following is a list of various types of flexible couplings:

- Gear couplings
- Continuous lubrication
- Grease packed
- Flexible membrane or flexible disc couplings
- Single membrane type
- Multiple membrane or multiple disc type
- Couplings with elastomer insert flexible drive members.

Gear couplings

Gear type couplings are shown in Figures 4.6.2 and 4.6.3. They usually include two separate gear mesh units, each of which consists of an external gear that fits closely into an internal gear.

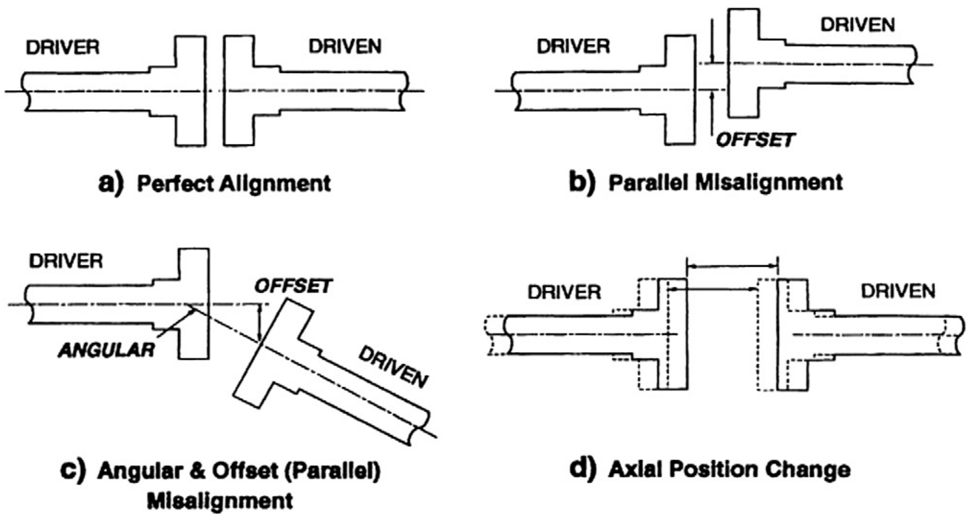


Fig 4.6.1 • Shaft misalignment and axial position

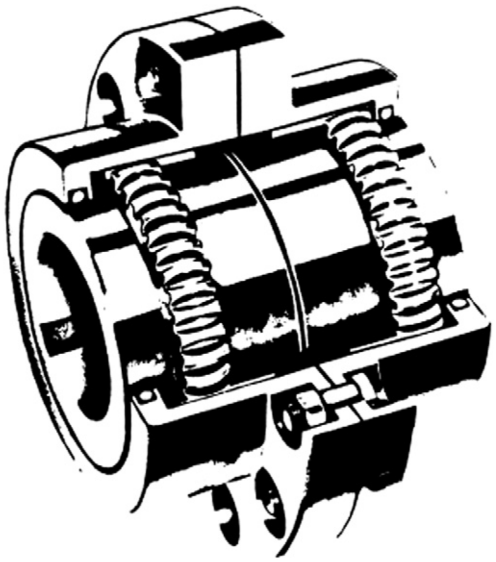


Fig 4.6.2 • Gear tooth coupling (grease packed) (Courtesy of Zurn Industries)

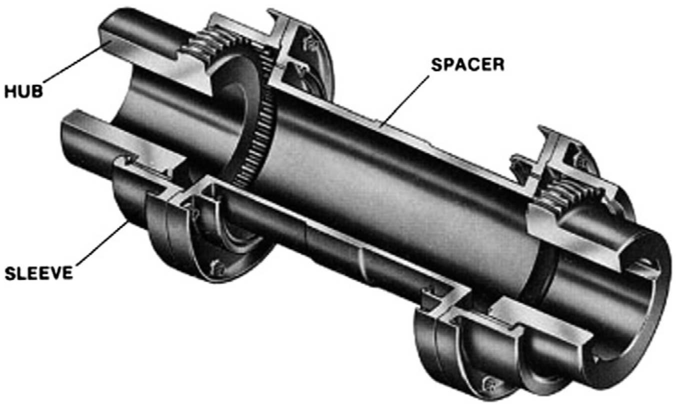


Fig 4.6.3 • Continuously lubricated gear type coupling with spacer (Courtesy of Zurn Industries)

The internal gear can either be part of the coupling hub assemblies, or be mounted on each end of the coupling spacer assembly. If the internal gears are hub mounted, then the external gears are spacer mounted and vice versa.

Grease pack couplings (see Figure 4.6.2) are normally designed with hub mounted external gears, and the internal gears are part of a sleeve type spacer which serves as a retainer for the grease lubrication. The flange joint of the sleeve is either precision ground to avoid lubrication leaks, or has a gasket between the two flange faces. The sleeve ends are fitted with 'O' ring seals to keep dust out and lubrication in.

In recent years, flexible element couplings have been used almost exclusively. They are the most compact of all the coupling designs for any given amount of torque transmission. For this reason, they also have the least overhung weight. In addition, the gear coupling can adapt more readily to requirements for axial growth of the driver and driven shafts. Axial position change tolerances are on the order of $^1/_2$" or greater.

There is a common disadvantage in all gear type flexible couplings. Any gear mesh has a break-away friction factor in the axial direction. This is caused by the high contact force between the two sets of gear teeth. The result is that the forces imposed on the driver and driven shafts are not totally predictable, and are sometimes higher than desired due to the quality of the tooth machine surfaces, and the inevitable build up of sludge or foreign material in the tooth mesh during extended service. These forces are detrimental to the ability of the coupling to make the required corrections for misalignment but, more importantly, can have a disastrous effect on the ability of the coupling to correct for thermal or thrust force changes between the driver and driven machines.

Both coupling manufacturers and users have long been aware of this problem, and have used many methods to minimize the effect. Some of these methods are:

- Reduction of the forces between the gear teeth by increasing the pitch diameter of the gear mesh. This is often self-defeating, in that it results in increased size of the coupling and the coupling weight.
- Reduction of the break-away friction factor by the use of higher quality gear tooth finish and better tooth geometry and fit.
- Reduction of sludge and foreign material build-up in the gear mesh by finer filtration of the coupling lubricant.
- Reduction of sludge and foreign material build-up in the gear mesh by incorporating self-flushing passages and ports in the coupling to allow any contaminants to pass through in the lubricant without being trapped in the gear mesh area.

These steps have been only partially successful and the problem still exists in many applications.

Coupling manufacturers are asked to quote the design break-away friction factor of their coupling as built and shipped from the factory. Machinery train designers then use this figure to calculate the maximum axial force that the coupling would be expected to exert on the connected shafts. From this information, the designers can decide if the thrust bearings adjacent to the coupling are adequate to handle the axial loads within the machine plus the possible load from the coupling resistance to any external forces.

There has been much discussion, and some disagreement, regarding the friction factor to be used when calculating the possible thrust forces that could be transmitted by the coupling. When the coupling is in reasonably good condition, factors from 0.15 to 0.30 have been considered reasonable. Since the factor reflects the total force relationship, the coupling design can have a significant effect on the factor used. The factor is a function of the number of teeth in contact, and the contact areas of each tooth, plus the quality of the tooth contact surface. If we assume that the factor to be used is 0.30, then the axial force which must be exerted in order to allow the coupling to correct for axial spacing changes can be calculated as:

$$F_a = \frac{0.30 \times T}{Dp/2}$$

Where: F_a = Required axial force in kg (pounds)
T = Design torque in Ncm (in/pounds)
Dp = Pitch diameter of gear mesh in cm (inches)

We can assume then, that if we use a coupling with a 15 cm (6 inch) pitch diameter gear mesh, which is transmitting 28,250 Ncm (25,000 inch-pounds) of torque and has a break-away friction factor of 0.30, the axial force required to move the gear mesh to a new axial position would be 11,300 N (2,500 pounds). Adjacent thrust bearings must be capable of handling this force in addition to the machine's normal calculated thrust forces. Machinery train designers and users must be aware of this, and make provisions for it in the built-in safety factors of the thrust bearings and machinery mounting design.

The machinery user must know that the same phenomenon has an effect on vibration when machinery is operated with excessive misalignment. The gear mesh position must change with each revolution of the shaft to correct for the misalignment. This results in counter axial forces on a cyclic basis, since the mesh is moving in opposite directions on each side of the coupling. Vibration detection and monitoring instrumentation will show that the resulting vibration will occur at twice the running frequency of the shafts. Although the primary force generated is axial, the resultant can show up as a radial vibration, due to the lever arm forces required on the coupling spacer to make the gear meshes act as ball and socket connections. Axial or radial vibration in rotating machinery that occurs at twice the frequency of the shaft rotational speed is normally an indication of misalignment between the two machines.

Figure 4.6.4 shows a continuously lubricated, spacer gear type coupling. Spacers are usually required for component removal (seals, etc). They also provide greater tolerance to shaft misalignment. A common spacer size that is used for un-spared (critical) equipment is 46cm (18 inches).

Flexible membrane or flexible disc couplings

Couplings in these categories do not have moving parts and derive their flexibility from controlled flexure of specially designed diaphragms or discs. They do not require lubrication and are commonly known as 'dry couplings'. The diaphragms or discs transmit torque from one shaft to the other just as do the gear meshes in a gear coupling.

The following features are common to all flexible disc or flexible membrane type couplings:

- None require lubrication.
- All provide a predictable thrust force curve for a given axial displacement range.
- Properly applied, operated and maintained, none are subject to wear and have an infinite life span.
- All provide smooth, predictable response to cyclic correction for minor misalignment.

It should be noted that **none** of the above comments can be applied across the board to gear type flexible couplings. For this reason, more and more special purpose machinery trains are being supplied with flexible metallic element couplings in their design. Many users do not allow the use of gear type coupling for critical (un-spared) applications.

The following is a discussion of the various types of 'dry' couplings with comments pertaining to their application ranges and limitations. Figure 4.6.4 shows a typical flexible disc coupling.

This is the most common type and is generally used for general purpose applications (pumps, fans, etc). The major

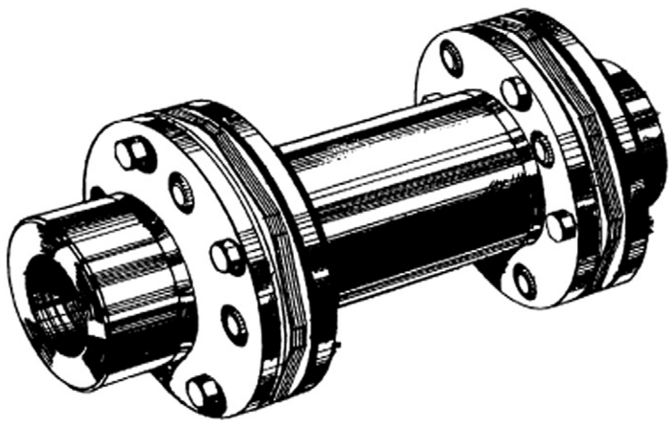


Fig 4.6.4 • Flexible disc spacer coupling (Courtesy of Rexnord)

consideration with this type of coupling is assuring the shaft end separation (BSE) is within the allowable limits of the couplings. This value is typically only 1.5mm (0.060") for shaft sizes in the 1-2" range. At shaft sizes above 4", the maximum end float can be 6mm (0.150") or more. Exceeding the allowable end float will significantly increase the axial load on the thrust bearings of the equipment, and can fail the coupling discs. A single diaphragm, spacer type coupling is shown in Figures 4.6.5 and 4.6.6. Figure 4.6.5 is a cutaway view and Figure 4.6.6 presents a two dimensional assembly drawing.

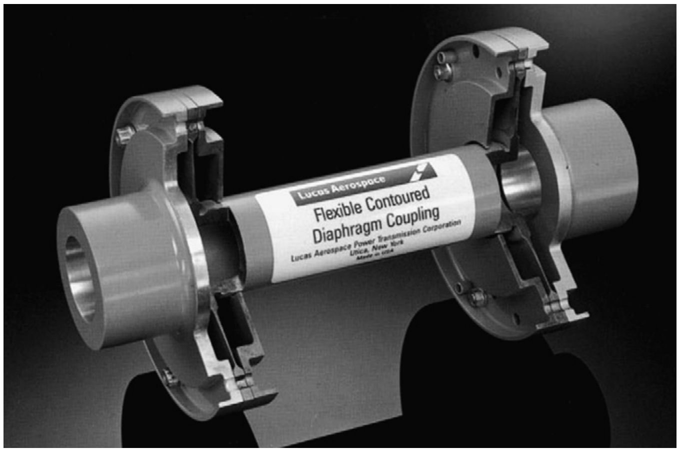


Fig 4.6.5 • Single diaphragm spacer coupling (Courtesy of Lucas Aerospace)

This type of coupling is commonly used for critical (un-spared) applications, where axial end float values are less than 5mm (0.125"). This limit is based on an approximate axial float of $\pm$ 1.5mm (0.062"). If end float is greater than 5mm (0.125"), a convoluted (wavy) diaphragm or multiple type diaphragm must be used. During disassembly, care must be taken when removing the spacer not to scratch or dent the diaphragm element. A dent or even a scratch that penetrates the protective coating can cause a diaphragm failure.

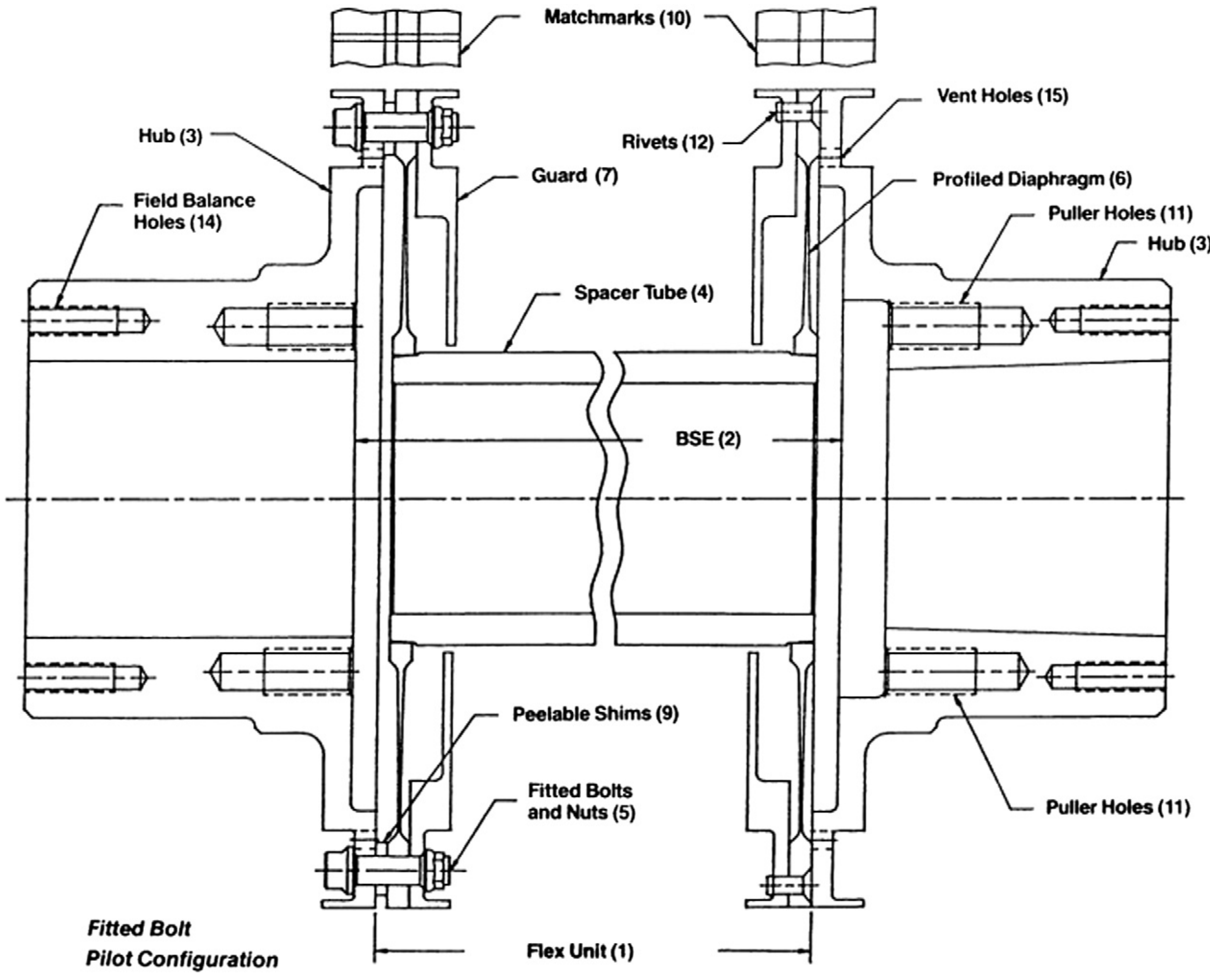


Fig 4.6.6 • Single diaphragm spacer coupling (Courtesy of Lucas Aerospace)

Regardless of the type of diaphragm coupling, it is common practice to 'pre-stretch' them to take full advantage of the maximum available end float. Readers are cautioned to always require that equipment vendors provide axial shaft movement calculations in order to confirm that the coupling maximum end float is not exceeded. Figure 4.6.7 graphically displays the various combinations of end shaft movement and the calculation method, and Figure 4.6.8 is a picture of a multiple, convoluted (wavy) diaphragm spacer coupling. This latter type coupling is used whenever large values of axial end float exist. Axial end float values of ± 22.2mm (0.875") or greater are attainable with this type of coupling.

As previously mentioned, gear type couplings provide the lowest value of overhung weight (coupling moment) on the bearing. However, a dry type coupling will usually have a higher coupling moment, because the flexible assembly is farther from the bearing centerline than the gear teeth in a gear coupling. An excessive coupling moment will reduce the second natural frequency (N_{c2}) of a turbo-compressor, and could move it close to, or within, the operating speed range. A solution in these cases can be to use a reduced moment diaphragm coupling as shown in Figure 4.6.9.

In this design, the diaphragm is moved to the back of the hub, and the flange diameter is reduced, thus significantly reducing the coupling moment. The reduced moment coupling approaches the gear coupling in term of coupling moment value.

Couplings with elastomer insert flexible drive members

This type of coupling is normally used only for low horsepower, general purpose applications. Their limitations are based primarily on the wear factor and the difficulty in maintaining the shape and concentricity of the elastomer insert. These items have a tendency to limit the maximum design speed at which such couplings can be operated. A typical 'jaw and spider' type is shown in Figure 4.6.10.

One exception is a special design used for synchronous motor driven compressor trains. A characteristic of synchronous motors is a variable oscillating torque that decreases linearly in frequency from 2× line frequency (50Hz or 60Hz) at 0 rpm, to zero frequency at rated rpm. Figure 4.6.11 shows a plot of motor rpm vs. transient torsional excitation frequency. The excitation frequency inherent in all synchronous motors will excite all torsional natural frequencies present between 2× line frequency and 0 rpm.

When the motor torsional excitation frequency briefly coincides with a torsional natural frequency, torque values can amplify to as much as five or six times full load torque. The 'Holset'

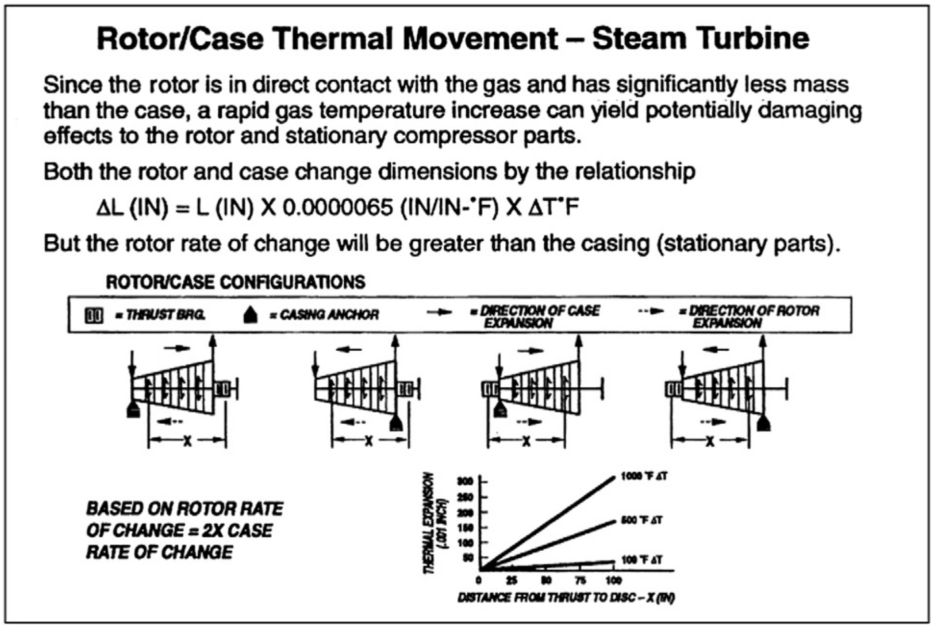


Fig 4.6.7 • Rotor/case thermal movement — steam turbine

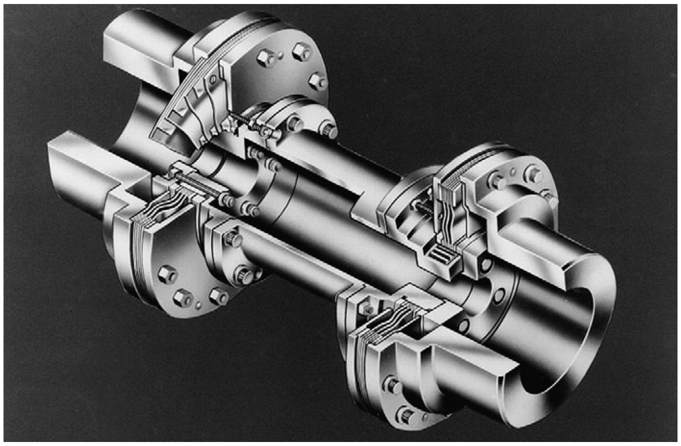


Fig 4.6.8 • Multiple, convoluted diaphragm-spacer coupling (Courtesy of Zurn Industries)

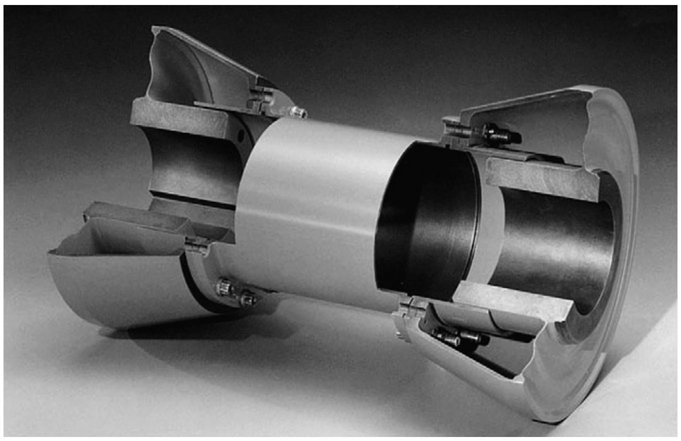


Fig 4.6.9 • Reduced moment convoluted (wavy) diaphragm spacer coupling (Courtesy of Lucas Aerospace)

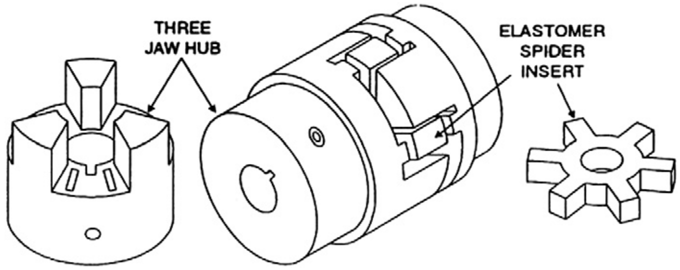


Fig 4.6.10 • Jaw and spider coupling

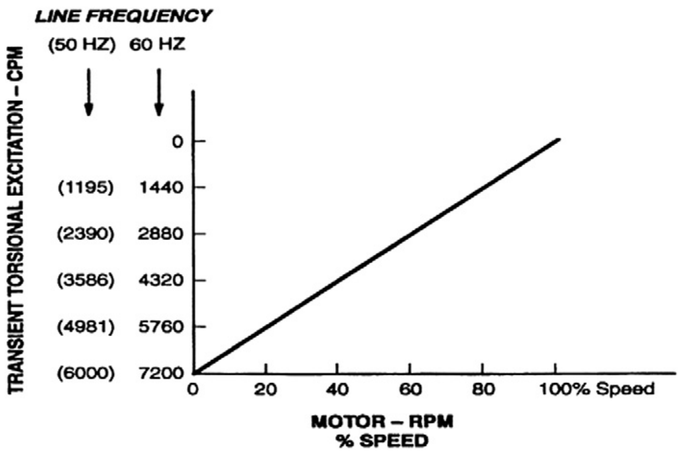


Fig 4.6.11 • Transient torsional excitation — Frequency vs. motor speed

or elastomeric coupling shown in Figure 4.6.12 significantly reduces the torque amplification, by dampening out the response in the elastomeric elements. The hardness of these elements is controlled to limit the maximum amplification factor to an acceptable value (usually 2-3× rated torque).

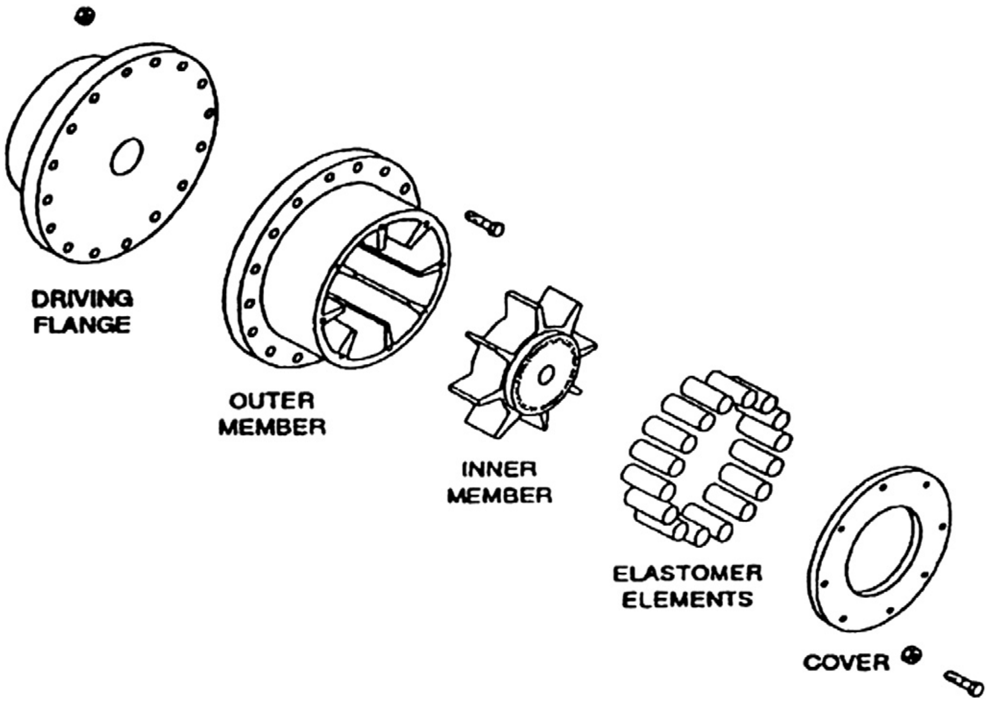


Fig 4.6.12 • Holset coupling (exploded view) non-spacer type

The coupling system

Generally speaking, if couplings are properly selected, the root cause of failure, if it occurs, is in the coupling system.

The 'coupling system' must continuously transmit torque safely between the driver and driven equipment, and must allow for changes in shaft misalignment and axial movement. The components that make up the shaft system are:

- The driver shaft
- Driver shaft/coupling fit
- Coupling
- Driven shaft/coupling fit
- The driven shaft
- The coupling spacer system
- Lubrication (if required)
- Cooling system (if required)

A schematic of a coupling system is shown in Figure 4.6.13.

The reliability of the coupling is a function of the coupling system design and assembly. If any of the items noted above are not properly designed or assembled, then a coupling failure can occur. Coupling assembly/disassembly errors and enclosed coupling guard design are two important areas that are critical to coupling system reliability.

Coupling installation and removal

The most common methods of coupling attachment are:

- Key fit
- Spline fit
- Hydraulic fit

Key fits are used whenever possible. They are the most common method of shaft fit. It is important to ensure keys and keyways are properly manufactured to avoid problems with removal or breakage. Key fits will be used on equipment that does not require coupling removal to remove shaft components (seals, bearings, etc). Keyed fits are usually used on motors, gearboxes and most pumps and small steam turbines. Since heat is usually required to remove keyed couplings, they will not be used where removal in the field is necessary. In these cases, either spline or hydraulic fits are used.

Spline fits consist of a male (on the shaft) and female (in coupling hub) finely-machined mating gear teeth with line to line fit (no backlash). When assembled on the shaft, the fit is rigid and provides no flexibility. Spline fits are commonly used in the gas turbine industry. They do not usually require heat for removal.

Hydraulic fits are used where heat to remove the coupling hub is either not available or not permitted. Usually, turbo-compressors will utilize hydraulic fits for this reason, since hydrocarbon gas, usually present, requires a flame-free environment. Figure 4.6.14 shows a typical coupling hydraulic shrink fit arrangement. Note that the entire torque load is transmitted by the shrink fit and that no keys are used!

The equipment vendor calculates the required shrink fit, based on the shaft and coupling dimensions. Typical values of hydraulic shrink fit are 0.002 inch/inch of shaft diameter.

For ease of hydraulic fit assembly and disassembly, all shafts and coupling hubs are tapered. Different shaft/coupling hub matching tapers are used. The most common are:

- $^1/_2°$ taper
- $^1/_2$" per foot taper
- $^3/_4$" per foot taper (shafts above 4" diameter)

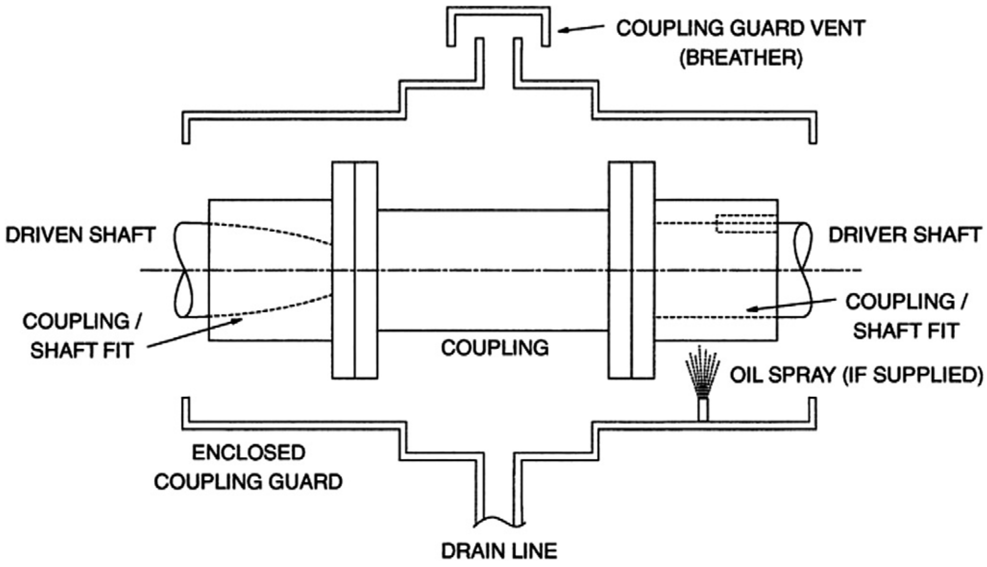


Fig 4.6.13 • The coupling system (Courtesy of M.E. Crane, Consultant)

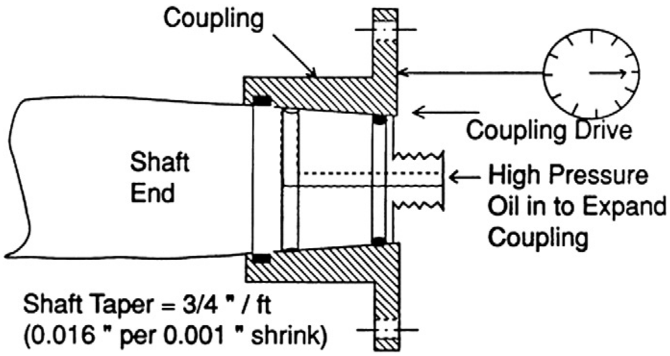


Fig 4.6.14 • Typical coupling hydraulic shrink fit

Once the shrink fit is calculated, the value appears on the coupling drawing and is usually expressed as ‘drive’ or ‘push’ based on the shaft taper. This is the axial distance the coupling must be moved up the shaft. The coupling drive per 0.001" of shrink fit for the most common shaft tapers is noted below:

Shaft taper	Drive per 0.001″ shrink
$^1/_2$°	1.448mm (0.057")
$^1/_2$" per foot	0.610mm (0.024")
$^3/_4$" per foot	0.406mm (0.016")

As an example, a hydraulic fit coupling with a 101.5mm (4") bore requires a 0.008" shrink fit (i.e. the bore diameter is 0.008" less than the shaft). To expand the coupling bore 0.008", what is the drive if the shaft taper is:

Taper	Drive
$^1/_2$°	11.582 mm (0.456")
$^1/_2$" per foot	4.877 mm (0.192")
$^3/_4$" per foot	3.251 (0.128")

Since the load torque is completely transmitted by the shrink fit, one can see the importance of ensuring that the correct shrink fit (or drive) is obtained. The shrink fit amount is directly proportional to torque load capability. If the shrink fit is 50% of the specified value, so is the torque capability! However, industry specifications require that the shrink fit at minimum tolerances be a minimum of 125% greater than the driver maximum torque. Observing the calculated drives in the example above, it can be seen that the smaller the shaft, the more critical the correct drive becomes for a given shaft taper. The coupling drive is measured by positioning a dial indicator on the coupling hub and measuring the axial distance traveled during coupling assembly.

Figure 4.6.15 shows a typical hydraulic coupling mounting arrangement used by Dresser-Rand. All turbo-compressor manufacturers use similar arrangements. There are some slight differences which are:

- Hydraulic oil enters the coupling hub and not the shaft.
- An additional pump is used to move the hydraulic tool axially (Figure 4.39 shows a nut which is manually turned to push the coupling axially).

The basic coupling mounting procedure is as follows (although readers must refer to the specific vendor’s instruction book for the exact procedure):

1. Clean shaft end and coupling bore with light oil.
2. Remove all ‘O’ rings from shaft end and coupling.
3. Lightly blue the coupling hub.
4. Push coupling on shaft **without ‘O’ rings** and tap hub with wood to ensure tight fit.
5. While hub is on shaft, index coupling hub axial position relative to a machined surface on shaft (usually shaft end).
6. Remove hub and confirm contact area of blue is a minimum of 85%. If not, correct as required.
7. When coupling contact of 85% is confirmed, clean shaft and coupling hub and install shaft and coupling ‘O’ rings.
8. Hand push coupling on shaft to **indexed position in step 5. NOTE: It may be necessary to use pump since ‘O’ rings can provide significant resistance to movement.**

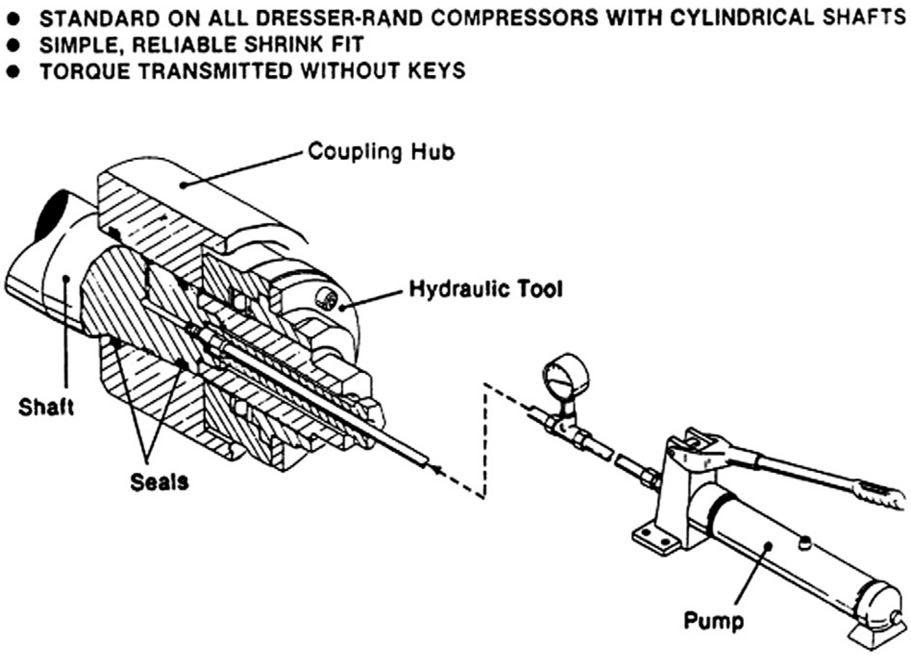


Fig 4.6.15 • Hydraulic fit coupling (Courtesy of Dresser-Rand Corp)

9. **With hub at indexed (zero drive) position**, use hand pump to push coupling axially to value noted on coupling drawing.
10. Coupling drive must be within tolerances noted. NOTE: Pump pressures will be high. Be extremely careful when connecting pump and tubing. Be sure to secure pump so that hand jacking cannot break tubing. Pressures typically required range from 103,000–206,000 kPa (15,000–30,000 PSI) depending on shaft dimensions, coupling dimensions and shrink fit.
11. When coupling is on shaft correct amount, do not remove dial indicator but reduce pump pressure to zero and back off hydraulic tool slightly. Observe that dial indicator does not move before removing tool.
12. Promptly assemble shaft end coupling nut.
13. Measure between shaft end dimension (B.S.E.) to ensure it is as stated on coupling drawing before assembling coupling spacer. If this dimension is not correct, consult instruction book and O.E.M. if necessary before taking corrective action. Under no circumstances should coupling spacers be added unless allowed by the coupling manufacturer or should equipment axial shaft position be changed without O.E.M. consent.
14. When coupling is properly assembled check alignment using 'reverse dial indicator procedure'. NOTE: For coupling removal, consult vendor's instruction book. Under no circumstances should coupling be pulled or heated. Usually, hydraulic pressure required for removal will be higher (5-10%) than that required for assembly. If the value required exceeds 241,000kPa (35,000 PSI), do not proceed until consulting O.E.M. for additional options concerning removal.

Incorrectly mounting a hydraulic coupling can cause catastrophic coupling and/or shaft end failure.

Enclosed coupling guards

Most turbo-compressor couplings are completely enclosed by a spark-proof (usually aluminum) coupling guard. This is because the couplings are the continuously lubricated gear type, or to prevent oil siphoned from the bearing brackets by the windage action of the dry couplings. In either case, proper design of the coupling guard is essential to maintaining coupling reliability. Many coupling failures have resulted from high coupling enclosure temperatures, or enclosures full of oil and debris that has entered the coupling guard from the atmosphere.

As a minimum the following must be checked by the O.E.M. and coupling vendor during equipment design or field coupling retrofit from gear to dry type:

- Proper coupling O.D. to guard and/or bearing bracket I.D. clearance.
- Proper coupling guard baffle design to allow proper drainage. **NOTE: All enclosed coupling guards must be supplied with vent and drain**.
- Proper vent breather sizing and design.

Figure 4.6.16 presents coupling guard dimensional design criteria for dry type couplings operating in enclosed coupling guards. Note that in some designs, D_o may be the ID of the bearing bracket.

It is recommended that coupling guard skin operating temperatures should be below 93°C (200°F) to avoid coupling and coupling guard leakage problems. Under no circumstances should coupling guard skin temperatures approach the flash point of lubricating oil, 200°C (400°F) for new mineral oil.

Field retrofits from lubricated to dry couplings

Many users are retrofitting their older style lubricated couplings to dry couplings because of their advantages. Whenever

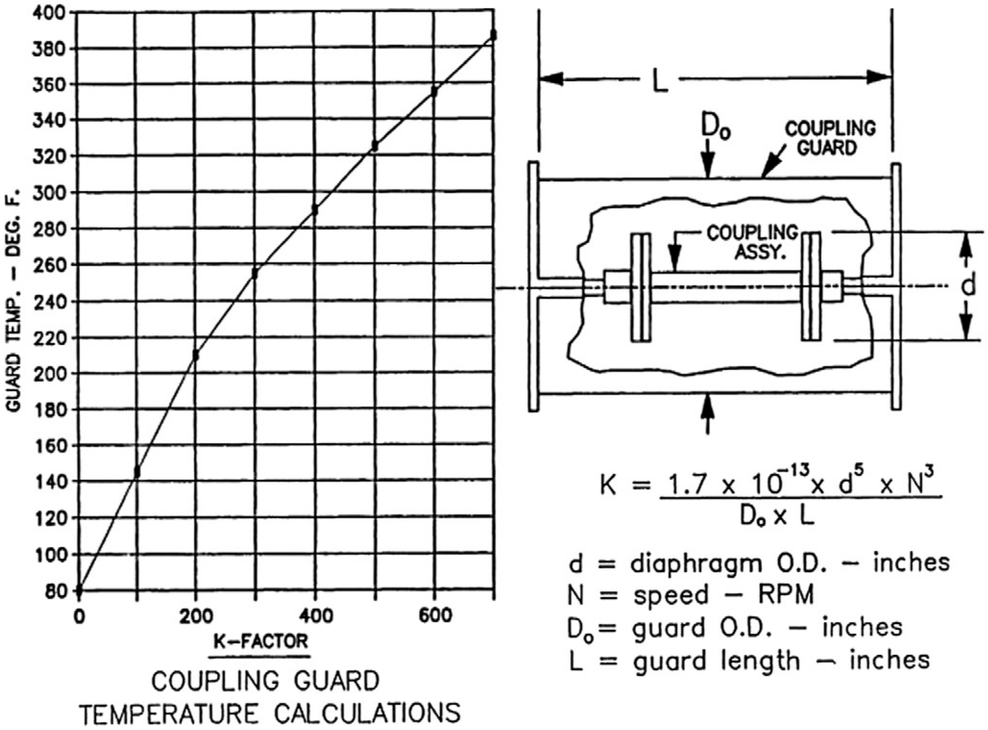


Fig 4.6.16 • Coupling guard dimensional design criteria (Courtesy of M.E. Crane, Consultant)

considering a retrofit, the following action should be taken to maintain or increase coupling system reliability:

- Consult equipment or coupling vendor for proper selection of new coupling.
- Consult equipment O.E.M.(s) (each affected vendor) to confirm:
  Critical speeds will not be affected
  Coupling guard design is acceptable
- Advise coupling vendor or any environmental considerations that may affect dry flexible element life (environmental gases, temperatures, excessive dust, etc.).

Best Practice 4.7

Ensure liquid free start-ups with hydraulically fit couplings, to prevent coupling slip and costly long term shutdowns.

Liquid in the compressor casing with hydraulic fit couplings (no keys) can exceed the slip torque minimum safety factors and slip the coupling hub, causing extensive damage to the coupling, rotor and coupling guard.

Casing drains must be supplied with a means of confirming the compressor case is liquid-free prior to start-up.

Special care is required for motor driven compressors that will accelerate to full load in less than 30 seconds.

Lessons Learned

The majority of hydraulic coupling slip incidents are caused by liquid inside the compressor casing at start-up.

Ensure that the procedure for draining compressor cases is properly written in the operations manual, and is implemented before each start-up.

Benchmarks

This best practice procedure has been used since 1975, when I experienced a catastrophic coupling slip incident. The coupling hub slipped, melted the hub and shaft, broke the coupling assembly and left the coupling guard, whereupon it struck a small diameter gas line and started a refinery fire.

Since that time, each compressor project has required the following plan concerning compressor casing draining:

- Multiple case drains
- A means of assuring that the case was properly drained
- Detailed compressor case draining procedure that is contained in the plant operations manual and must be implemented properly by all shifts.

B.P. 4.7. Supporting Material

The coupling system

It has been the writers' experience that if couplings are properly selected, the root cause of failure, if it occurs, is in the coupling system.

The coupling system must continuously transmit torque safely between the driver and driven equipment and must allow for changes in shaft misalignment and axial movement. The components that make up the shaft system are:

- The driver shaft
- Driver shaft/coupling fit
- Coupling
- Driven shaft/coupling fit
- The driven shaft
- The coupling spacer system
- Lubrication (if required)
- Cooling system (if required)

A schematic of a coupling system is shown in Figure 4.7.1.

The reliability of the coupling is a function of the coupling system design and assembly. If any of the items noted above are not properly designed or assembled, then a coupling failure can occur. Coupling assembly/disassembly errors and enclosed coupling guard design are two important areas that are critical to coupling system reliability.

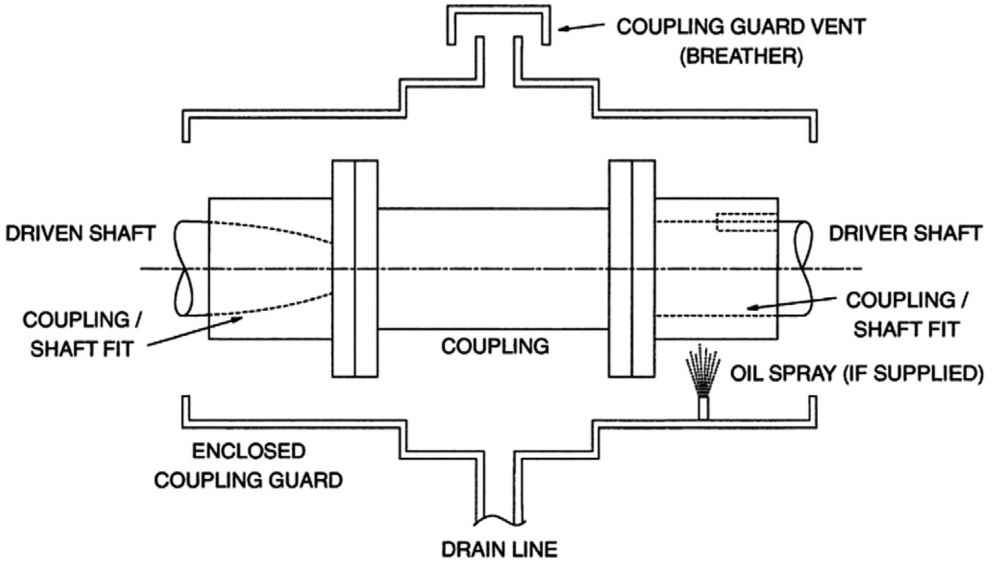


Fig 4.7.1 • The coupling system (Courtesy of M.E. Crane, Consultant)

Best Practice 4.8

Require a review of dry, flexible coupling, shaft end, thermal, axial growth calculations, and never exceed coupling axial installation limits since they can cause coupling failure and possible catastrophic safety issues.

Flexible couplings (disc and diaphragm) have limited axial movement.

The driver and driven equipment suppliers calculate the maximum shaft axial growth of the shafts.

These values, along with the coupling maximum axial movement limits, determine the dimensions of the complete coupling assembly as compared to the between shaft extensions (BSE) dimension.

In many cases, the shaft end axial growth will exceed the dry, flexible coupling, maximum axial movement dimension. In these cases, the coupling is not assembled in a neutral position, but is either pre-compressed or pre-stretched, so that the installed coupling assembly will ensure that the maximum safe amount of movement will not be exceeded at any time during operation.

Coupling breakage under load conditions can cause steam turbine overspeed and blades to exit through the exhaust case, even though the overspeed trip system is set correctly and functions properly. The trapped steam between the trip valve and throttle valves has sufficient energy to accelerate the turbine rotor beyond its design limits.

Lessons Learned

High shaft vibration and/or coupling breakage has occurred as a result of improper shaft end thermal axial growth and/or improper flexible coupling assembly installation.

Benchmarks

This best practice has been used since flexible 'dry' couplings were first used in the late 1960s, and has resulted in trouble free coupling assemblies without any coupling failures.

B.P. 4.8. Supporting Material

Flexible membrane or flexible disc couplings

Couplings in these categories do not have moving parts, and derive their flexibility from controlled flexure of specially designed diaphragms or discs. They do not require lubrication and are commonly known as 'dry couplings'. The diaphragms or discs transmit torque from one shaft to the other just as do the gear meshes in a gear coupling.

The following features are common to all flexible disc or flexible membrane type couplings:

- None require lubrication.
- All provide a predictable thrust force curve for a given axial displacement range.
- Properly applied, operated and maintained, none are subject to wear and have an infinite life span.
- All provide smooth, predictable response to cyclic correction for minor misalignment.

It should be noted that **none** of the above comments can be applied across the board to gear type flexible couplings. For this reason, more and more special purpose machinery trains are being supplied with flexible metallic element couplings in their design. Many users do not allow the use of gear type coupling for critical (un-spared) applications.

The following is a discussion of the various types of 'dry' couplings with comments pertaining to their application ranges and limitations.

Figure 4.8.1 shows a typical flexible disc coupling.

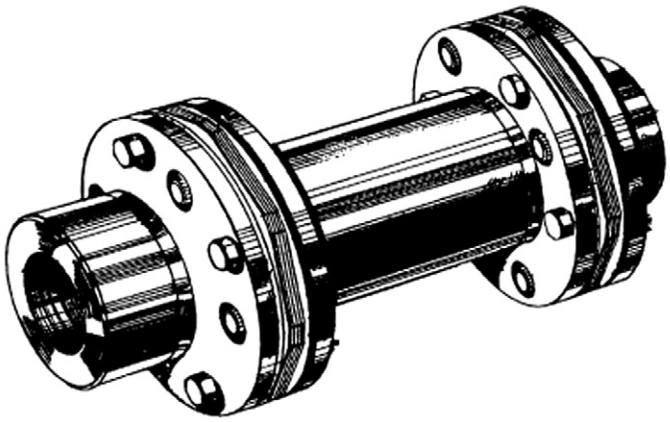


Fig 4.8.1 • Flexible disc spacer coupling (Courtesy of Rexnord)

This is the most common type and is generally used for general purpose applications (pumps, fans, etc). The major consideration with this type of coupling is assuring the shaft end separation (BSE) is within the allowable limits of the couplings. This value is typically only 1.5mm (0.060") for shaft sizes in the 1-2" range. At shaft sizes above 4", the maximum end float can be 6mm (0.150") or more. Exceeding the allowable end float will significantly increase the axial load on the thrust bearings of the equipment, and can fail the coupling discs. A single diaphragm, spacer type coupling is shown in Figures 4.8.1 and 4.8.2. Figure 4.8.1 is a cutaway view and Figure 4.8.2 presents a two dimensional assembly drawing.

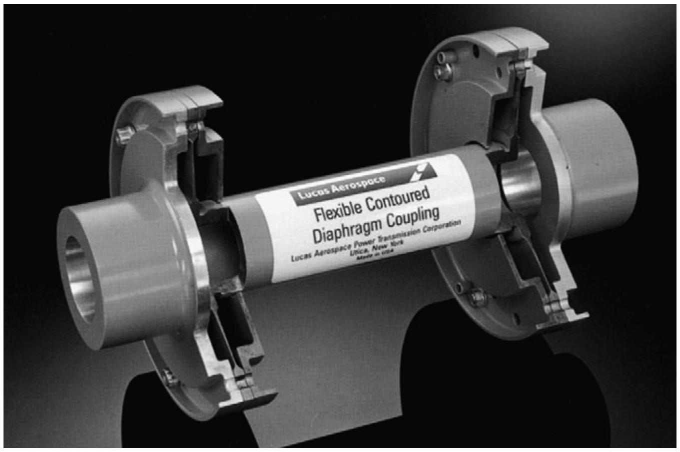


Fig 4.8.2 • Single diaphragm spacer coupling (Courtesy of Lucas Aerospace)

This type of coupling is commonly used for critical (un-spared) applications where axial end float values are less than 0.125". This limit is based on an approximate axial float of ± 0.062". If end float is greater than 0.125", a convoluted (wavy) diaphragm or multiple type diaphragm must be used. During disassembly, care must be taken when removing the spacer to not scratch or dent the diaphragm element. A dent or even a scratch that penetrates the protective coating can cause a diaphragm failure.

Regardless of the type of diaphragm couplings, it is common practice to 'pre-stretch' these couplings to take full advantage of the maximum available end float. Readers are cautioned to always require equipment vendors provide axial shaft movement. Figure 4.8.4 graphically displays the various combinations of end shaft movement and the calculation method.

Figure 4.8.5 is a picture of a multiple, convoluted (wavy) diaphragm spacer coupling. This type of coupling is used whenever large values of axial end float exist. Axial end float values as high as ± 0.875" are attainable with this type of coupling.

As previously mentioned, gear type couplings provide the lowest value of overhung weight (coupling moment) on the bearing. However, a dry type coupling will usually have a higher coupling moment because the flexible assembly is farther from the bearing centerline than the gear teeth in a gear coupling. An excessive coupling moment will reduce the second natural frequency (N_{c2}) of a turbo-compressor and could move it close to or within the operating speed range. A solution in these cases can be to use a reduced moment diaphragm coupling as shown in Figure 4.8.6.

In this design, the diaphragm is moved to the back of the hub and the flange diameter is reduced thus significantly reducing the coupling moment. The reduced moment coupling approaches the gear coupling in term of coupling moment value.

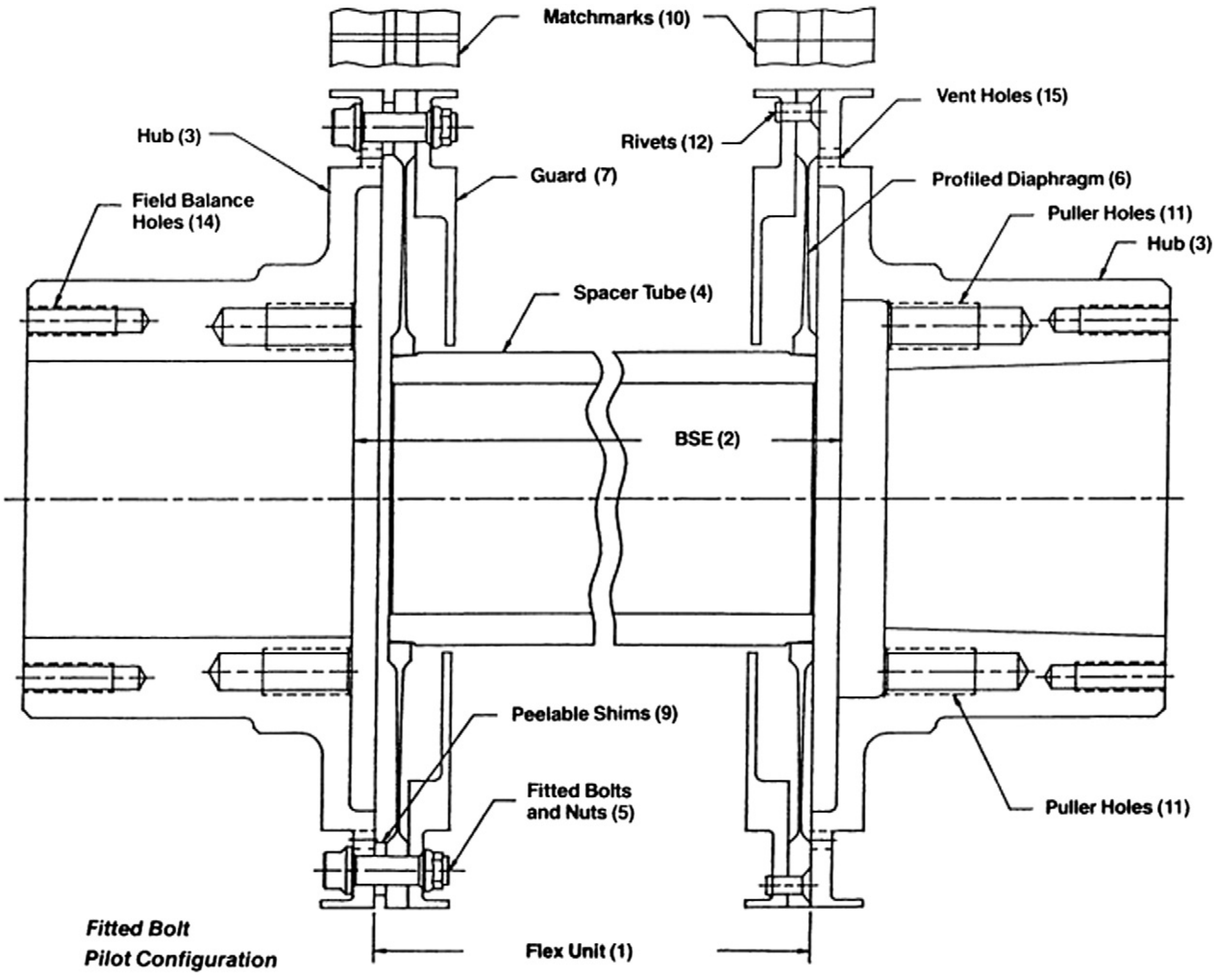


Fig 4.8.3 • Single diaphragm spacer coupling (Courtesy of Lucas Aerospace)

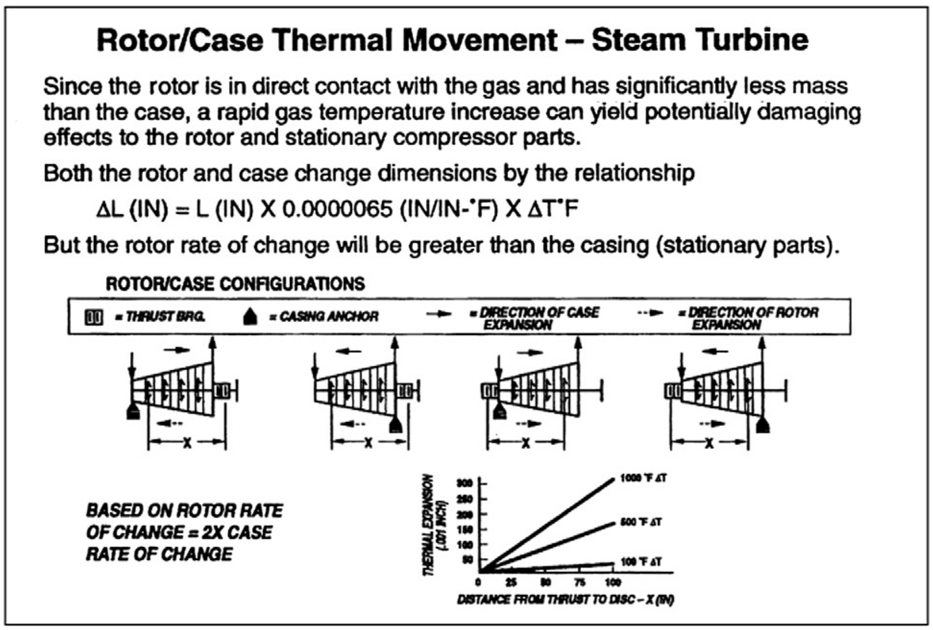


Fig 4.8.4 • Rotor/case thermal movement – steam turbine

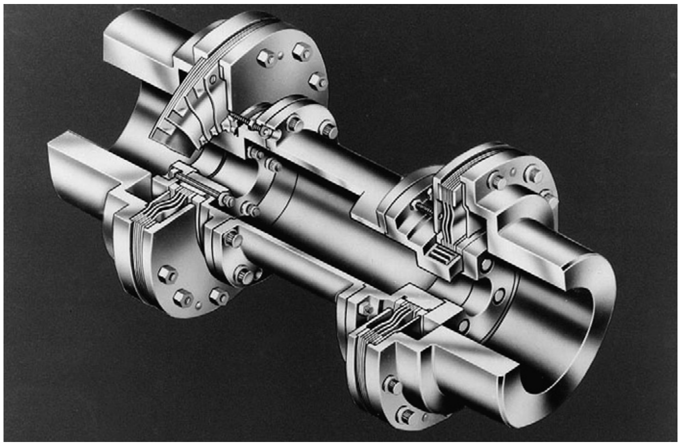


Fig 4.8.5 • Multiple, convoluted diaphragm-spacer coupling (Courtesy of Zurn Industries)

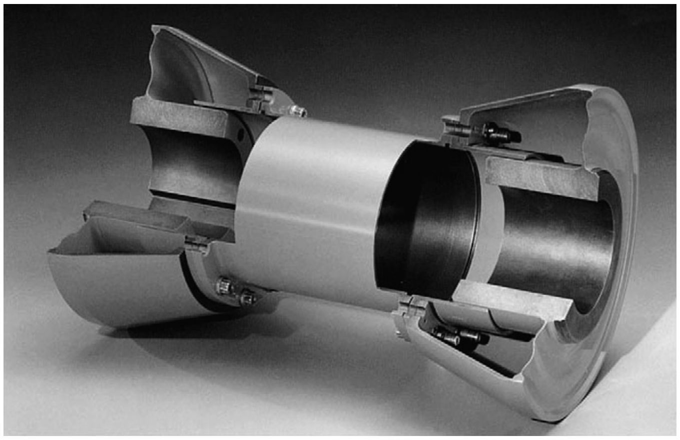


Fig 4.8.6 • Reduced moment convoluted (wavy) diaphragm spacer coupling (Courtesy of Lucas)

Best Practice 4.9

Limit shaft end tapers for hydraulic fit couplings to 1/2" per foot or less to allow sufficient tolerance in coupling axial movement (axial push or drive) to attain shrink fit.

Hydraulic fit couplings use only the shrink fit between the hub and shaft to carry the full transmitted torque load (power/speed).

If the coupling is not assembled with the proper shrink fit it will slip, possibly causing catastrophic damage and exposing the site to a safety incident.

The shaft end taper and the axial coupling hub movement (push) dimension determine the coupling shrink fit.

Coupling shaft end tapers for hydraulic fit vary from vendor to vendor and even from machine to machine for the same vendor. The greater the shaft taper, the less the push required, and the lower the torque transmission capability of the coupling system.

High shaft end tapers (3/4" per foot and above) do not allow sufficient tolerance in the event that the coupling drive (axial push) specified values are not met.

Lessons Learned

3/4" per foot tapers have caused catastrophic shaft, coupling and coupling guard failures on small diameter shafts because they did not allow sufficient tolerance to prevent coupling slippage.

Benchmarks

This best practice has been used since 1975 when I experienced a catastrophic coupling failure due to the use of a 3/4" per foot hydraulic coupling shrink fit. The use of this best practice has resulted in no coupling shrink fit slippage issues since that time in all associated projects.

B.P. 4.9. Supporting Material

Coupling installation and removal

The most common methods of coupling attachment are:

- Key fit
- Spline fit
- Hydraulic fit

Key fits are used whenever possible. They are the most common method of shaft fit. It is important to ensure keys and keyways are properly manufactured to avoid problems with removal or breakage. Key fits will be used on equipment that does not require coupling removal to remove shaft components (seals, bearings, etc). Keyed fits are usually used on motors, gearboxes and most pumps and small steam turbines. Since heat is usually required to remove keyed couplings, they will not be used where removal in the field is necessary. In these applications either spline or hydraulic fits are used.

Spline fits consist of a male (on the shaft) and female (in coupling hub) finely machined mating gear teeth with line to line fit (no backlash). When assembled on the shaft, the fit is rigid and provides no flexibility. Spline fits are commonly used in the gas turbine industry. They do not usually require heat for removal.

Hydraulic fits are used where heat to remove the coupling hub is either not available or not permitted. Usually,

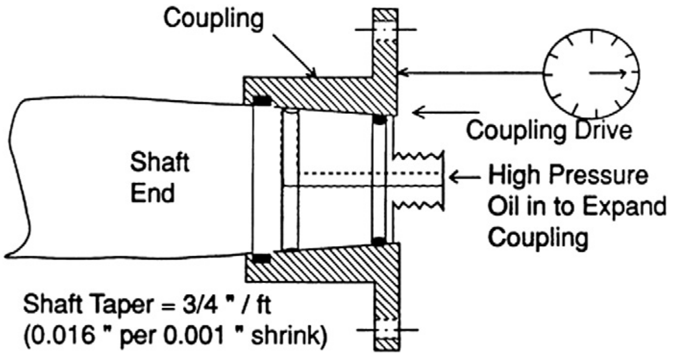


Fig 4.9.1 • Typical coupling hydraulic shrink fit

turbo-compressors will utilize hydraulic fits for this reason, since hydrocarbon gas, usually present, requires a flame free environment. Figure 4.9.1 shows a typical coupling hydraulic shrink fit arrangement. Note that the entire torque load is transmitted by the shrink fit and that no keys are used!

The equipment vendor calculates the required shrink fit based on the shaft and coupling dimensions. Typical values of hydraulic shrink fit are 0.002 inch/inch of shaft diameter.

For ease of hydraulic fit assembly and disassembly, all shafts and coupling hubs are tapered. Different shaft/coupling hub matching tapers are used. The most common are:

- 1/2° taper
- 1/2" per foot taper
- 3/4" per foot taper (shafts above 4" diameter)

Once the shrink fit is calculated, the value appears on the coupling drawing and is usually expressed as 'drive' or 'push' based on the shaft taper. This is the axial distance the coupling must be moved up the shaft. The coupling drive per 0.001" of shrink fit for the most common shaft tapers is noted below:

Shaft taper	Drive per 0.001" shrink
1/2°	1.448mm (0.057")
1/2" per foot	0.610mm (0.024")
3/4" per foot	0.406mm (0.016")

As an example, a hydraulic fit coupling with a 101.5mm (4") bore requires a 0.008" shrink fit (i.e. the bore diameter is 0.008" less than the shaft). To expand the coupling bore 0.008", what is the drive if the shaft taper is:

Taper	Drive
1/2°	11.582mm (0.456")
1/2" per foot	4.877mm (0.192")
3/4" per foot	3.251mm (0.128")

Since the load torque is completely transmitted by the shrink fit, one can see the importance of assuring that the correct shrink fit (or drive) is obtained. The shrink fit amount is directly proportional to torque load capability. If the shrink fit is 50% of the specified value, so is the torque capability! However, industry specifications require that the shrink fit at minimum tolerances be a minimum of 125% greater than the driver maximum torque. Observing the calculated drives in the example above it can be seen that the smaller the shaft, the more critical the correct drive becomes for a given shaft taper. The coupling drive is measured by positioning a dial indicator on the coupling hub and measuring the axial distance traveled during coupling assembly.

Figure 4.9.2 shows a typical hydraulic coupling mounting arrangement used by Dresser-Rand. All turbo-compressor

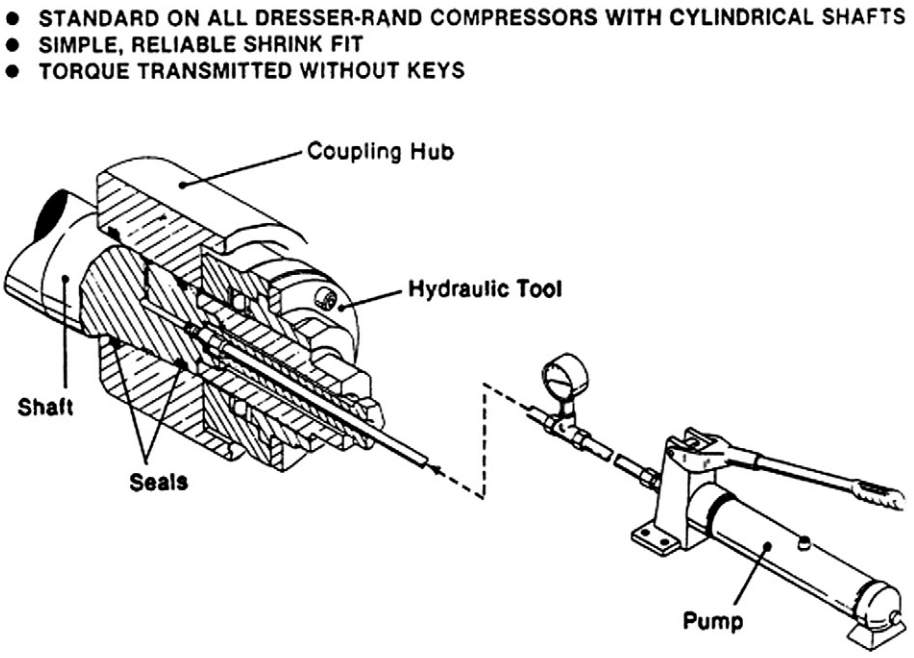


Fig 4.9.2 • Hydraulic fit coupling (Courtesy of Dresser-Rand Corp)

manufacturers use similar arrangements. There are some slight differences which are:

Hydraulic oil enters the coupling hub and not the shaft. Figure 4.9.2 shows a nut which is manually turned to push the coupling axially.

The basic coupling mounting procedure is as follows (although readers must refer to the specific vendors' instruction book for the exact procedure):

1. Clean shaft end and coupling bore with light oil.
2. Remove all 'O' rings from shaft end and coupling.
3. Lightly blue the coupling hub.
4. Push coupling on shaft **without 'O' rings** and tap hub with wood to ensure tight fit.
5. While hub is on shaft, index coupling hub axial position relative to a machined surface on shaft (usually shaft end).
6. Remove hub and confirm contact area of blue is a minimum of 85%. If not, correct as required.
7. When coupling contact of 85% is confirmed, clean shaft and coupling hub and install shaft and coupling 'O' rings.
8. Hand push coupling on shaft to **indexed position in step 5. NOTE: It may be necessary to use pump since 'O' rings can provide significant resistance to movement.**
9. **With hub at indexed (zero drive) position**, use hand pump to push coupling axially to value noted on coupling drawing.
10. Coupling drive must be within tolerances noted. **NOTE: Pump pressures will be high. Be extremely careful when connecting pump and tubing. Be sure to secure pump so that hand jacking cannot break tubing**. Pressures typically required range from 103,000-206,000kPa (15,000-30,000 PSI) depending on shaft dimensions, coupling dimensions and shrink fit.
11. When coupling is on shaft correct amount, do not remove dial indicator but reduce pump pressure to zero and **back off hydraulic tool slightly. Observe that dial indicator does not move before removing tool**.

- Promptly assemble shaft end coupling nut.
- Measure between shaft end dimension (BSE) to ensure it is as stated on coupling drawing before assembling coupling spacer. If this dimension is not correct, consult instruction book and O.E.M. if necessary before taking corrective action. **Under no circumstances should coupling spacers be added unless allowed by the coupling manufacturer or should equipment axial shaft position be changed without O.E.M. consent**.
- When coupling is properly assembled check alignment using 'reverse dial indicator procedure'. **NOTE: For coupling removal, consult vendor's instruction book. Under no circumstances should coupling be pulled or heated. Usually, hydraulic pressure required for removal will be higher (5-10%) than that required for assembly. If the value required exceeds 241,000kPa (35,000 PSI), do not proceed until consulting O.E.M. for additional options concerning removal**.

Incorrectly mounting a hydraulic coupling can cause catastrophic coupling and/or shaft end failure.

Best Practice 4.10

Correct hydraulic coupling hub installation will ensure trouble free operation and optimum unit safety and reliability.

Always install shrink fit couplings first without 'O' rings.
Measure proper (80% contact) and standoff of coupling hub to shaft end.
Assemble 'O' rings to obtain previous standoff.
Drive (push) coupling to specified axial movement.

See procedure contained in B.P. 4.9 Supporting Material for further details.

Lessons Learned

Installing a hydraulic coupling with hub and shaft (if applicable) "O" rings installed will result in insufficient coupling shrink and can cause hydraulic coupling slippage.

Benchmarks

This best practice has been used since 1975, and has resulted in trouble free hydraulically shrunk coupling installations and coupling removals.

B.P. 4.10 SUPPORTING MATERIAL

Please refer to material in B.P. 4.9.

5

Steam Turbine Best Practices

Introduction

In addition to steam turbine component design, steam turbine reliability is dependent on many systems that can and will affect its safety and reliability. These systems are:

- Steam system
- Condensing system
- Governor system
- Protection (trip) system
- Trip valve exerciser system
- Steam seal system
- Cooling system
- Monitoring system

Effective monitoring of these systems will ensure the optimum level of safety, reliability and MTBF. This chapter presents these important best practices.

Best Practice 5.1

Accurately define steam conditions to ensure maximum turbine power output and reliability.

Ensure that inlet and exhaust pressure and temperature ranges (maximum, normal and minimum) at the turbine flange are confirmed prior to data sheet preparation.

Lower than specified pressure and temperature differential conditions during field operation will limit produced power and affect plant revenue.

For condensing turbines, be sure that condenser conditions are maintained properly to prevent power reduction or exhaust end blade erosion (high vacuum conditions).

Lessons Learned

Failure to properly specify steam conditions has led to low turbine power output and erosion of the last stages of blading in condensing turbines.

Low steam energy conditions in the field (low pressure and temperature differentials) will reduce power output.

High vacuum conditions (closer to a perfect vacuum) increase the moisture in the exhaust steam. Moisture content in excess of 12% will result in reduced exhaust end blading life in rotor and diaphragm blading assemblies.

Benchmarks

This best practice has been used since the mid 1970s, and has resulted in optimum steam turbine performance and reliability (in excess of 99.7%).

B.P. 5.1. Supporting Material

Steam conditions

Steam conditions determine the energy available per pound of steam. Figure 5.1.1 explains where they are measured, and how they determine the energy produced.

- The steam conditions are the pressure and temperature conditions at the turbine inlet and exhaust flanges.
- They define the energy per unit weight of vapor that is converted from potential energy to kinetic energy (work).

Fig 5.1.1 • Steam conditions

Frequently, proper attention is not paid to maintaining correct steam conditions at the flanges of a steam turbine. Failure to do this will affect the power produced, and can cause mechanical damage to turbine internals resulting from blade erosion and/or corrosion. Figure 5.1.2 presents these facts.

Inlet steam conditions should be as close as possible (+/− 5%) to specified conditions because:

- Power output will decrease
- Exhaust end steam moisture content will increase, causing blade, nozzle and diaphragm erosion.

Fig 5.1.2 • Steam condition limits

Mollier diagrams or steam tables allow determination of the energy available in a pound of steam for a specific pressure and temperature. Figure 5.1.3 describes the Mollier diagram and the parameters involved.

Refer to Figure 5.1.4, an enlarged Mollier diagram.

Describes the energy per unit mass of fluid when pressure and temperature are known.

- Enthalpy (energy/unit mass) is plotted on Y axis
- Entropy (energy/unit mass degree) is plotted on X axis
- Locating P_1, T_1 gives a value of enthalpy (H) horizontal and entropy (S) vertical
- Isentropic expansion occurs at constant entropy ($\Delta S = 0$) and represents an ideal (reversible) expansion

Fig 5.1.3 • The Mollier Diagram

As an exercise, plot the following values on the Mollier diagram in this section and determine the corresponding available energy in BTUs per pound.

1. $P_1 = 600$ PSIG, $T_1 = 800°F$ $\quad h_1 = \dfrac{BTU}{LB_M}$
2. $P_2 = 150$ PSIG, $T_2 = 580°F$ $\quad h_2 = \dfrac{BTU}{LB_M}$
3. $P_1 = 1500$ PSIG, $T_1 = 900°F$ $\quad h_1 = \dfrac{BTU}{LB_M}$
4. $P_2 = 2$ PSIG, % moisture = 9% $\quad h_2 = \dfrac{BTU}{LB_M}$

Having plotted various inlet and exhaust conditions on the Mollier diagram to become familiar with its use, please refer to Figure 5.1.5, which presents the definitions and uses of steam rate.

Theoretical steam rate

The theoretical steam rate is the amount of steam, in kg or lb per hour, required to produce one (1) horsepower, if the isentropic efficiency of the turbine is 100%. As shown in Figure 5.1.5, it is determined by dividing the theoretical enthalpy, $\Delta h_{isentropic}$, into the amount of kJ/hr (btu/hr in one (1) unit of power (kW or hp)).

Actual steam rate

The actual steam rate is the amount of steam, in kg or lb per hour, required to produce one (1) unit of power based on the actual turbine efficiency. As shown in Figure 5.1.5, it is

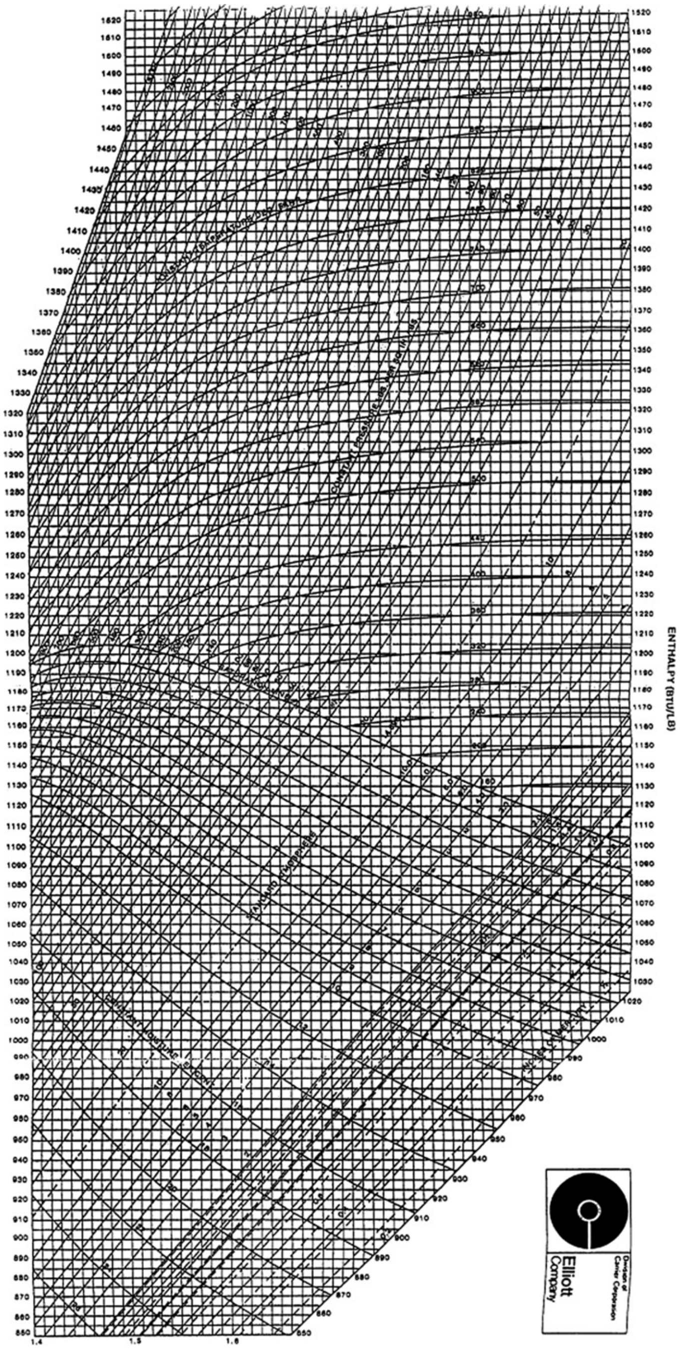


Fig 5.1.4 • Mollier steam diagram (Courtesy of Elliott Company)

determined by dividing the theoretical steam rate (TSR) by the turbine efficiency. Alternately, if the turbine efficiency is not known, and the turbine inlet and exhaust conditions are given (P_2, T_2 or % moisture), the actual steam rate can be obtained in the same manner as theoretical steam rate, but substituting ΔH_{actual} for $\Delta H_{isentropic}$.

Turbine efficiency

As shown in Figure 5.1.5, turbine efficiency can be determined either by the ratio of TSR to ASR or Δh_{actual} to $\Delta H_{isentropic}$.

It is relatively easy to determine the efficiency of any operating turbine in the field if the exhaust conditions are superheated. All that is required are calibrated pressure and temperature gauges on the inlet and discharge, and a Mollier diagram or steam tables. The procedure is as follows:

1. For inlet conditions, determine h_1
2. For inlet condition with $\Delta S = 0$, determine h_{2ideal}
3. For outlet conditions, determine $h_{2actual}$
4. Determine $\Delta h_{ideal} = h_1 - h_{2ideal}$
5. Determine $\Delta h_{actual} = h_1 - h_{2actual}$
6. Determine efficiency

$$\text{Efficiency} = \frac{\Delta H_{actual}}{\Delta H_{ideal}}$$

However, for turbines with saturated exhaust conditions, the above procedure cannot be used because the actual exhaust condition cannot be easily determined. This is because the % moisture must be known. Instruments (calorimeters) are available, but results are not always accurate. Therefore the suggested procedure for turbines with saturated exhaust conditions is as follows:

1. Determine the power required by the driven equipment or record turbine power if a torquemeter is installed. This is equal to the power produced by the turbine.
2. Measure the following turbine parameters using calibrated gauges:

   - P_{in}
   - T_{in}
   - $P_{exhaust}$
   - Steam flow (in lbs/hr)

3. Determine the theoretical steam rate by plotting P_{in}, T_{in}, $P_{exhaust}$ @ $\Delta S = 0$, and dividing $\Delta h_{isentropic}$ into the constant.
4. Determine the actual steam rate of the turbine as follows:

$$\text{Actual Steam Rate (A.S.R.)} = \frac{\text{Steam Flow (lb/hr)}}{\text{BHP required by driven equipment}}$$

5. Determine efficiency

$$\text{Efficiency} = \frac{\text{T.S.R.}}{\text{A.S.R.}}$$

Figure 5.1.6 presents the advice and values concerning steam turbine efficiencies. The efficiencies presented can be used for estimating purposes.

Refer to Figures 5.1.7 and 5.1.8 for typical efficiency values for multistage and single stage steam turbines as a function of steam conditions, power and speed.

Uses:

- Determine the amount of steam required per hour
- Determine the amount of potential kW (horsepower)

Required:

- Steam conditions
- Theoretical steam rate table or Mollier diagram
- Thermal efficiency of turbine

Formula:

<u>Metric Units</u>

- Theoretical steam rate

$$\text{TSR (kg/kWhr)} = \frac{3600\ \text{kJ/kW-hr}}{\Delta H_{ISENTROPIC}}$$

- Actual steam rate

$$\text{A.S.R. (kg/kWhr)} = \frac{\text{T.S.R.}}{\text{Efficiency}} = \frac{3600\ \text{kJ/kW-hr}}{\Delta H_{ACTUAL}}$$

- Turbine efficiency

$$\text{Efficiency} = \frac{\text{T.S.R.}}{\text{A.S.R}} = \frac{\Delta H_{ACTUAL}}{\Delta H_{ISENTROPIC}}$$

<u>U.S. Units</u>

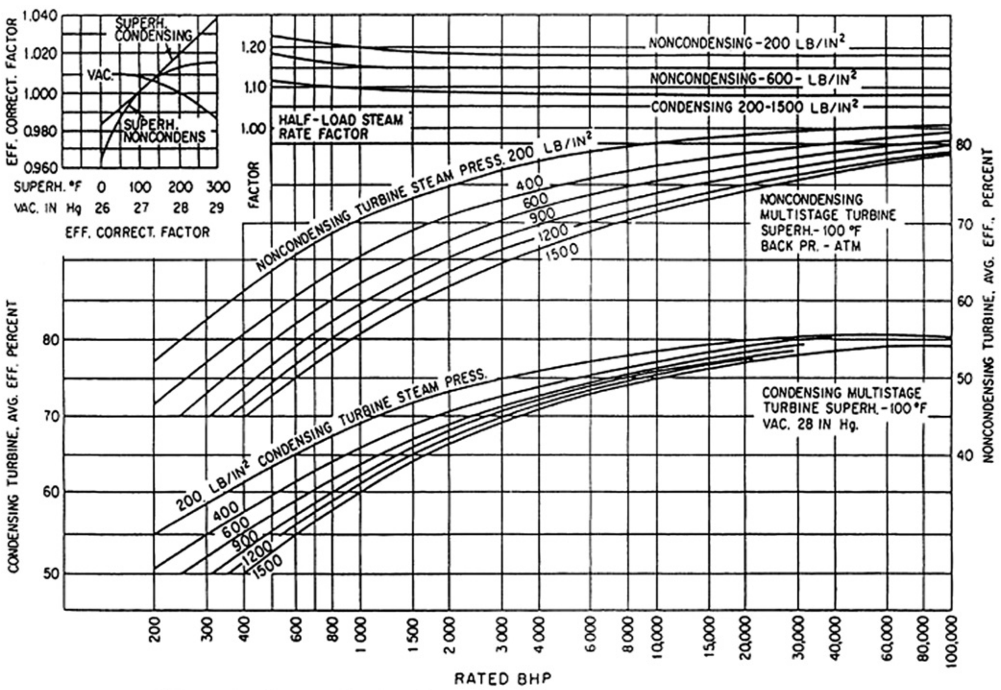
$$\text{TSR (lb/HP-hr)} = \frac{2545\ \text{BTU'S/HP-hr}}{\Delta H_{ISENTROPIC}}$$

$$\text{A.S.R. (lb/HP-hr)} = \frac{\text{T.S.R}}{\text{Efficiency}} = \frac{2545\ \text{BTU/HP-hr}}{\Delta H_{ACTUAL}\ \text{HP/hr}}$$

Fig 5.1.5 • Determining steam rate

- Quoted turbine efficiencies are external efficiencies; they include mechanical (bearing, etc.) and leakage losses
- Turbine efficiency at off load conditions will usually be lower than rated efficiency
- Typical efficiencies are presented for impulse turbine:
  - Condensing multi-stage
  - Non-condensing multi-stage
  - Non-condensing single state

Fig 5.1.6 • Typical steam turbine efficiencies



Average efficiency of multistate turbines (gear loss not included).

Fig 5.1.7 • Efficiency of multistage turbines (Courtesy of IMO Industries)

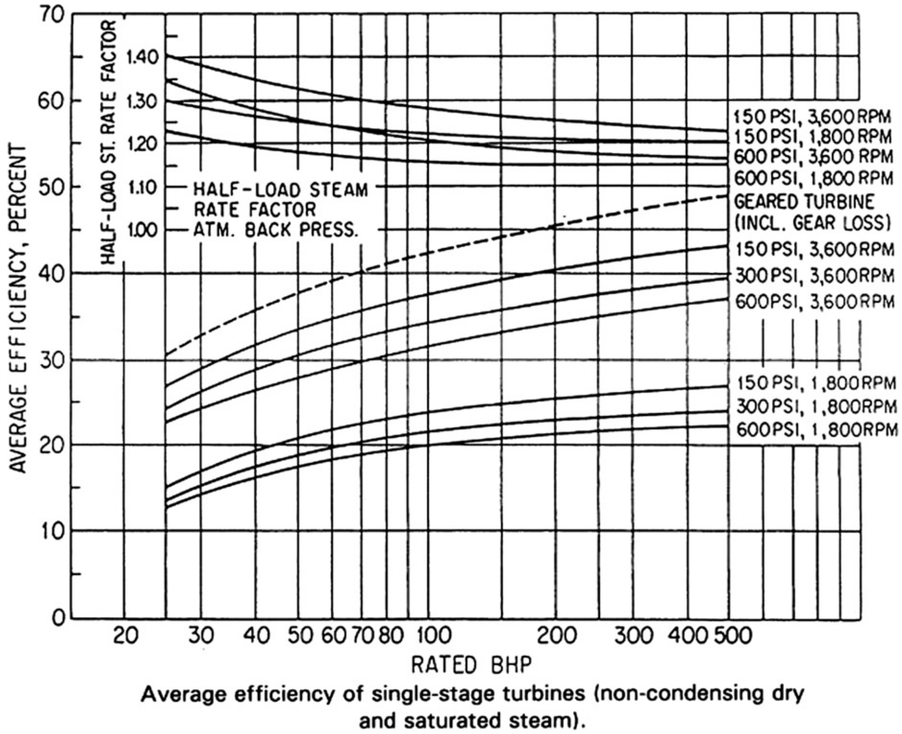


Fig 5.1.8 • Efficiency of single stage turbines (Courtesy of IMO Industries)

Best Practice 5.2

Size critical (un-spared) steam turbines for the maximum possible driven equipment load (full curve power) for maximum driven equipment flow rate, revenue and cost savings over costly driver re-rates after plant start-up.

Present industry practice for steam turbine power is to size the driver for 10% greater power than the rated point of the driven equipment.

Turbine fouling, lower than anticipated steam conditions and/or driven equipment fouling will limit driven equipment flow rate and result in reduced plant revenue.

The potential revenue increase from sizing the steam turbine for full curve power should be evaluated in the pre-FEED phase of the project against the added cost and reduced turbine efficiency, and compared to the cost of re-rating the turbine after plant start-up.

Lessons Learned

Potential increased production rates have been negated by insufficient steam turbine power caused by steam condition inadequacy and/or fouling, resulting in lost opportunity revenue until a costly re-rate project is approved, resulting in millions of dollars of lost revenue.

Benchmarks

This best practice has been used since 2000, when mega projects have become the norm, and have allowed pre-investment in additional driver power based on additional revenue potential.

B.P. 5.2. Supporting Material

See B.P. 5.1 for supporting material.

Best Practice 5.3

Screen for proven blade row experience, energy per blade row (blade loading) and tip speed to ensure critical (un-spared) steam turbine optimum safety and reliability.

State in the invitation to bid document (ITB) that proven blade profile, energy per blade row (blade loading) and tip speed experience will be reviewed with each quoting turbine vendor prior to submitting a priced proposal.

Lack of proven experience should disqualify a quoting turbine vendor for the project if they cannot change their design to incorporate proven blading for each stage of the turbine.

Be sure to discuss disqualification with the vendor, noting that they will be considered for any future projects where they have proven blade experience. Disqualification during the early stages of bidding saves the vendor proposal costs, which are significant.

Lessons Learned

The use of prototype blading has resulted in costly blade failures for critical (un-spared) turbine applications.

Benchmarks

This best practice has been used since the 1980s, and resulted in smooth start-ups and trouble free steam turbine operation without any blade failures occurring.

B.P. 5.3. Supporting Material

Typical compressor train pre-bid meeting agenda

The following agenda for a steam turbine drive compressor train is included for guidance purposes. Note: 1-2 days will be required for the meeting to review all details based on unit risk classification.

1. Compressor experience review (vendor to include necessary reference charts, tables, etc.)
   - Casing experience and review of compressor layout drawing
   - Impeller experience (flow and head coefficient)
   - Individual impeller curve (location of rated point to impeller best efficiency point)
   - Impeller stress
   - Rotor response
   - Stability analysis (if applicable)
   - Bearings – surface speed, load and experience
   - Thrust balance
   - Seals – surface speed, balance forces and experience
   - Surge control and process control system
2. Steam turbine or motor experience review (vendor to include necessary reference charts, tables, etc.)
   - Turbine casing experience and review of layout drawing
   - Stage nozzle and blade experience (profile, velocity ratio, BTU/stage)
   - Blade attachment method and blade stresses
   - Campbell and Goodman diagram review
   - Rotor response
   - Bearings – surface speed, load and experience
   - Thrust balance (reaction and hybrid types)
   - Shaft seals
   - Transient torsional response experience review (synchronous motors)
   - Control and protection system
3. Gear experience (if applicable) (vendor to include necessary reference charts, tables, etc.)
   - Gear box experience review and review of layout drawing
   - Review of gear data sheet
   - Gear calculation review (in accordance with API 613)
   - Bearings – surface speed, load and experience
   - Thrust loading – single helical gears
   - Pitch line velocity review
4. Auxiliary system experience (lube, dry gas seal and control oil system)
   - Review of P&IDs
   - Review of API 614 data sheets
   - Review of typical arrangement drawings
   - List of experienced system sub-suppliers
   - Review of proposed dry gas seal supplier information
5. Scope of supply for compressor train (all components and auxiliaries) review
6. Compressor train (all components) exceptions to specification
7. Meeting summary and action required.
   Note: Based on machinery risk, the following 'design checks' may be required:
   - Aerodynamic
   - Thermodynamic
   - Rotor response
   - Stability analysis
   - Seal balance
   - Thrust balance
   - Bearing loading
   - Control system simulations
   - System layout maintenance accessibility

Best Practice 5.4

Trend "after first stage" pressures vs. steam flow, as well as phase angle to detect the onset of steam path fouling, in order to determine the necessity for water washing.

Ensure that after first stage, pressures are trended and plotted on vendor-provided "after first stage" pressure vs. steam flow curves.

If after the first stage, the pressure falls above the curve for a measured steam flow, the turbine is beginning to foul.

Further confirmation is any change in vibration phase angles once abnormal first stage pressures are observed.

Consult management, and plan for a timely on-line turbine water wash, closely following vendor recommendations.

The above procedure also applies to the post-first stage pressure in the low pressure section of extraction turbines.

If transmitters are installed, this information can be brought into the control room DCS and plotted on the vendor curves to provide operations and reliability personnel with immediate indication of fouling.

Lessons Learned

Failure to observe, trend and use the vendor's "after first stage" pressure vs. steam flow curve will result in unexpected turbine vibration that can trip the unit on high vibration.

Fouling will initially accumulate evenly on a rotor, but will break off unevenly, so causing unbalanced vibration that can reach levels that are high enough to trip the turbine.

Benchmarks

This best practice has been used since the 1990s, achieving steam turbine reliabilities above 99.5% and to direct timely on-line water washes.

B.P. 5.4. Supporting Material

The mechanism of fouling

As mentioned earlier, one can reduce any blade row or impeller to a series of equivalent orifices. Flow is a function of area and velocity.

Whenever any blade row or impeller is designed, the designer sets the inlet and discharge blade areas such that optimum velocities relative to the blade will be achieved at each location. By a combination of tests and experience, designers have defined optimum relative velocity rather well. Therefore the resulting inlet and discharge areas will produce optimum velocities and corresponding optimum impeller efficiencies. If, however, the areas were to change, and flow passages were to become rough and non-continuous, the performance of the impeller would change as a result. Figure 5.4.1 shows the effect of fouling on a closed centrifugal impeller.

Fouling is defined as the accumulation of debris in the impeller or blade passage, which reduces the flow area and roughens the surface finish. The distribution of the foulant on the impeller or blade row is non-uniform and usually changes with time. Flow patterns within the impeller or blade cause unequal distribution. In addition, the forces exerted on the foulant cause it to chip off with time as it becomes dry and brittle. This results in a change in rotor balance and a change in performance (head and efficiency).

The effect of fouling on the operating point

If we refer back to the previous example of a backward leaning centrifugal compressor impeller, the effect of fouling can be understood. Figure 5.4.2 shows the effect of fouling on the relative velocity.

Since the area of the flow passage is reduced when the impeller is fouled, V_{REL} will increase, the flow angle, $\propto$, will increase, and therefore result in an absolute velocity (increased R) as shown in Figure 5.4.3.

The increase in $\propto$ and R due to fouling will reduce the tangential velocity of the gas as shown in Figure 5.4.4.

Since the head (energy) produced by the impeller is the product of the impeller tip speed 'U', which does not change in the fouled condition, and the tangential velocity which is reduced, the head produced will be reduced in the fouled condition. In addition, the non-uniform distribution of the foulant will reduce the efficiency of the impeller stage.

Figure 5.4.5 shows the effect of fouling on the impeller stage curve. Impeller fouling is the accumulation of material in the

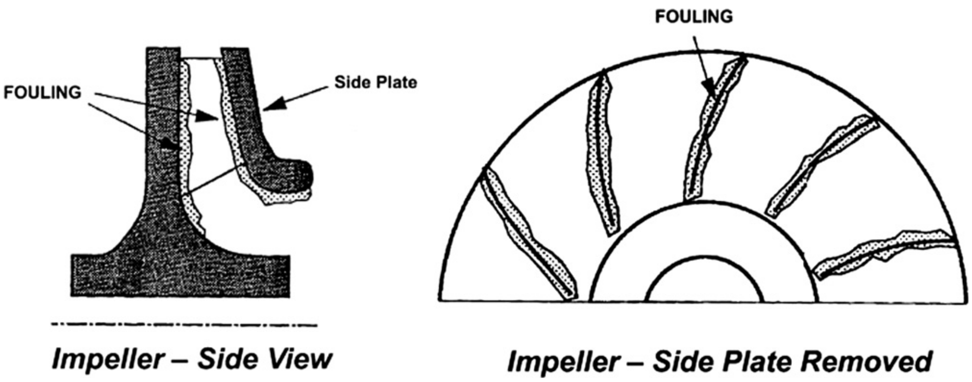


Fig 5.4.1 • Fouling – the effect on the operating point. Left: impeller – side view. Right: impeller – side plate removed

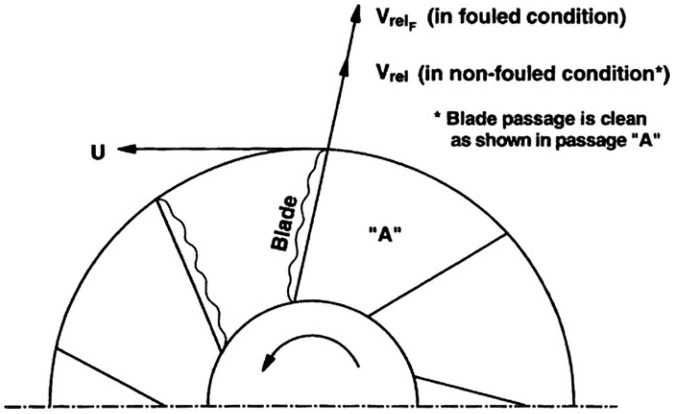


Fig 5.4.2 • Impeller with side plate removed

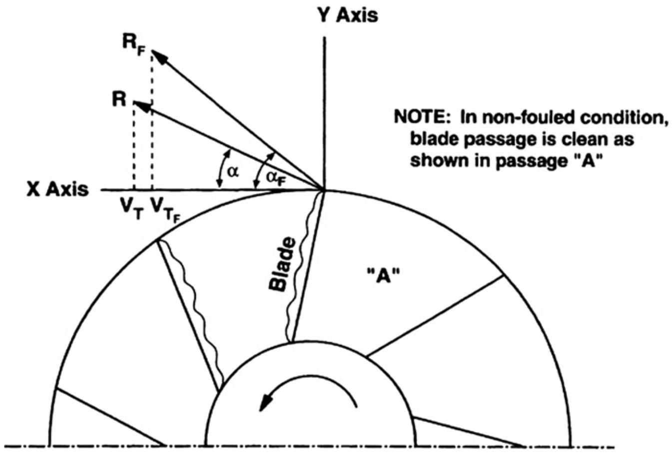


Fig 5.4.4 • Impeller with side plate removed

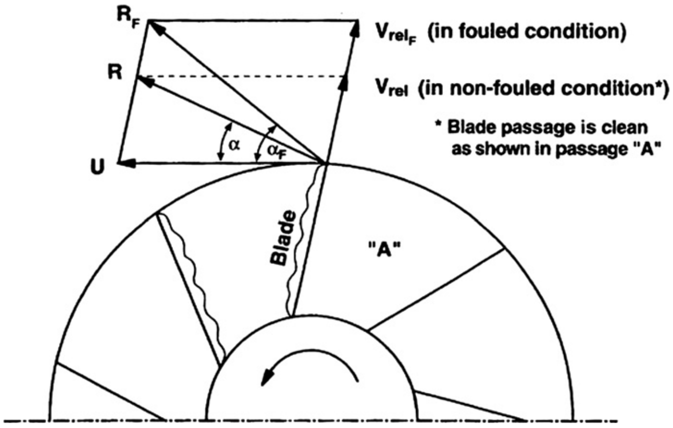


Fig 5.4.3 • Impeller with side plate removed

impeller passages that reduces flow area and roughens surface finish. It reduces impeller head capacity and efficiency.

Note that the surge margin actually increases slightly in the fouled condition. This is because the cause of surge is low gas velocity. Since the area of the flow passage is reduced, the gas velocity increases; thus increasing the surge margin. The surge margin is defined as the flow at surge divided by the impeller design flow. However, the stage head produced by the impeller at any flow rate is reduced. Therefore, for the same process system head required, the impeller flow rate will be reduced, thus forcing the operating point closer to the surge line.

While this example considers a centrifugal compressor stage, fouling has the same effect on a steam turbine stage. Steam velocity in the blades is increased, efficiency is reduced and less power is produced.

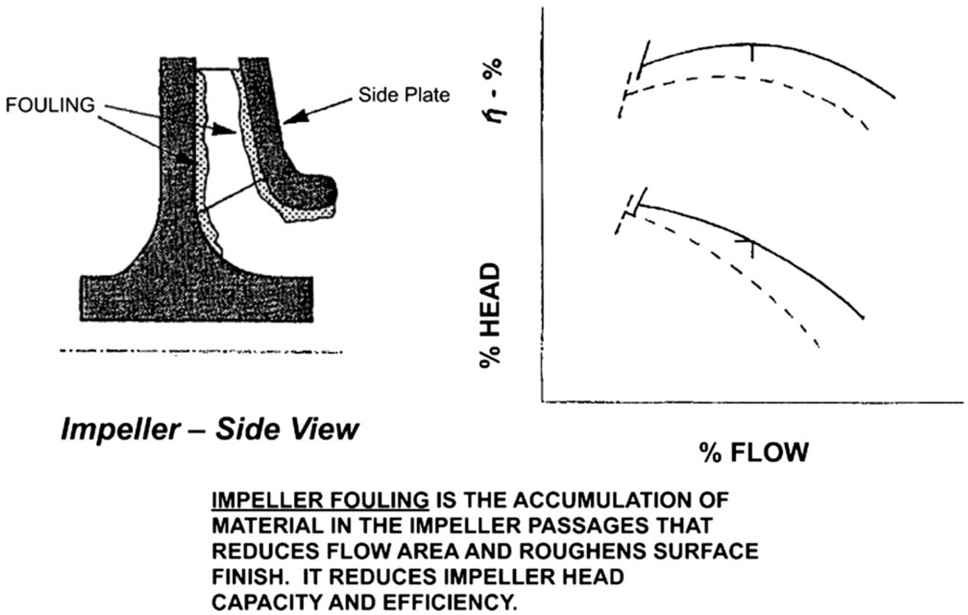


Fig 5.4.5 • Impeller fouling

The causes of fouling in steam turbines

The causes of fouling are described in Figure 5.4.6.

- Boiler upsets resulting in deposits of calcium and/or silica on turbine blades
- Improper boiler feed water treatment resulting in calcium and/or silica deposits on turbine blades

Fig 5.4.6 • The causes of fouling in steam turbines

Figure 5.4.7 shows a typical “after first stage” pressure curve for the high pressure section of an extraction/condensing steam turbine.

Measurement of steam flow and post-first stage pressure in the high pressure section will enable plotting of the operating point on this curve. If this is above the curve, which is based on clean turbine steam path conditions, fouling is present assuming flow and pressure measurement are accurate.

For this reason, trending is advised and confirmation of pressure/flow instrumentation is advised prior to taking corrective action (on-line or off-line water washing).

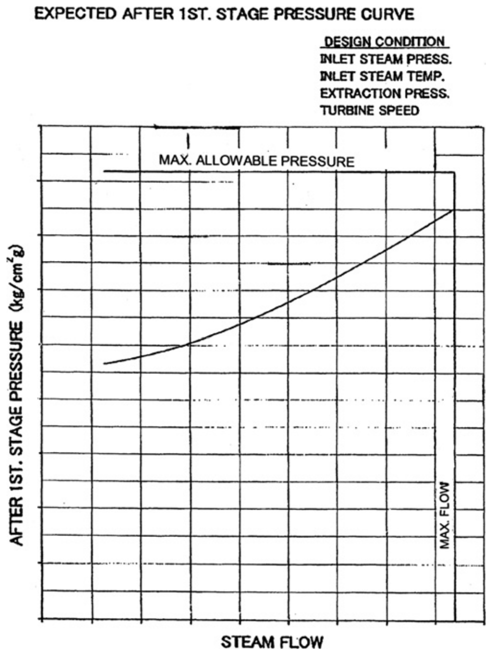


Fig 5.4.7 • Typical “after 1st stage” turbine curve (Courtesy of MHI)

Best Practice 5.5

Use shaft stiffness ratio to detect possible rotor instabilities and the need for a turning gear.

Shaft stiffness ratios are defined by bearing span divided by the diameter of the shaft in the blade disc area. Values of this parameter above 10 indicate the need to review vendor experience for similar turbines to ensure acceptable field rotor vibration. The following approach is recommended if the ratio exceeds 10:

- Request vendor experience list for similar ratios and operating speeds showing stiffness ratios, predicted first and second critical speeds and measured critical speeds on test
- Require that the rotor response analysis be run with the parameters used for the examples that had the closest predicted critical speeds to the recorded critical speeds on test
- Contact the references of the these units to confirm satisfactory field operation

Shaft stiffness ratio can also help in determining if a turning gear will be required on smaller turbines. If the ratio exceeds 10, a turning gear is recommended.

Lessons Learned

Failure to screen shaft stiffness has resulted in unexpected factory acceptance tests (FAT), critical speed margins that do not meet the project specifications, and discovery in the field that a turning gear should have been used.

Benchmarks

This best practice has been used since 1990 to ensure trouble-free rotor vibration characteristics and to require turning gears when vendors did not recommend them. The result has been trouble-free field operation and steam turbine reliabilities exceeding 99.5%.

B.P. 5.5. Supporting Material

The term ‘critical speed’ is often misunderstood. In nature, all things exhibit a natural frequency. This is defined as that frequency at which a body will vibrate if excited by an external force. The natural frequency of any body is a function of its stiffness and mass. As mentioned, for a body to vibrate, it must be excited. A classical example of natural frequency excitation is the famous bridge ‘Galloping Gerty’ in the state of Washington, USA. That bridge vibrated to destruction when its natural frequency was excited by prevailing winds.

In the case of turbo-compressor rotors, their natural frequency must be excited by some external force to produce a response that will result in increased amplitude of vibration. One excitation force that could produce this result is the speed of the rotor itself, which gives rise to the term ‘critical speeds’. The term ‘critical speed’ defines the operating speed at which a natural frequency of a rotor system will be excited. All rotor systems have both lateral (horizontal and vertical) and torsional (twist about the central shaft axis) natural frequencies. Only lateral critical speeds will be discussed in this section.

In the early days of rotor design, it was thought that the rotor system consisted primarily of the rotor supported by the bearings. This led to the assumption that only the stiffness of the rotor supported by rigid bearings needed to be considered in the analysis of the natural frequency. Countless machinery problems have proven this assumption to be false over the years. The concept of the 'rotor system' must be thoroughly understood. The rotor system consists of the rotor itself, the characteristics of the oil film that support the rotor, the bearing, the bearing housing, the compressor case that supports the bearing, compressor support (base plate), and the foundation. The stiffness and damping characteristics of all of these components together result in the total rotor system that produces the rotor response to excitation forces.

We will examine a typical rotor response case in this section and note the various assumptions, the procedure modeling, the placement of unbalance, and the response calculation output, and discuss the correlation of these calculations to actual test results.

Critical speeds

The natural frequency of any object is defined by the relationship:

$$F_{NATURAL} = \sqrt{\frac{K}{M}}$$

Where: K = Stiffness
M = Mass

When excited by an external force, any object will vibrate at its natural frequency. If the frequency of the exciting force is equal to the natural frequency of the object, and no damping is present, the object can vibrate to destruction. Therefore, if the frequency of an exciting force equals the natural frequency of an object, the exciting force is operating at the 'critical frequency'.

Rotor speed is one of the most common external forces in turbo-machinery. When the rotor operates at any rotor system natural frequency, it is said that the rotor is operating at its critical speed. The critical speed of a rotor is commonly designated as NC and the corresponding natural frequencies or critical speeds are: NC_1, NC_2, NC_3, etc.

Every turbo-compressor must have its rotor system critical speeds determined prior to manufacture. In this section, we will follow the procedure for the determination of the necessary parameters to define a rotor system's critical speed. The procedure is commonly known as determination of rotor response. Figure 5.5.1 is a representation of a critical speed map for a rotor system.

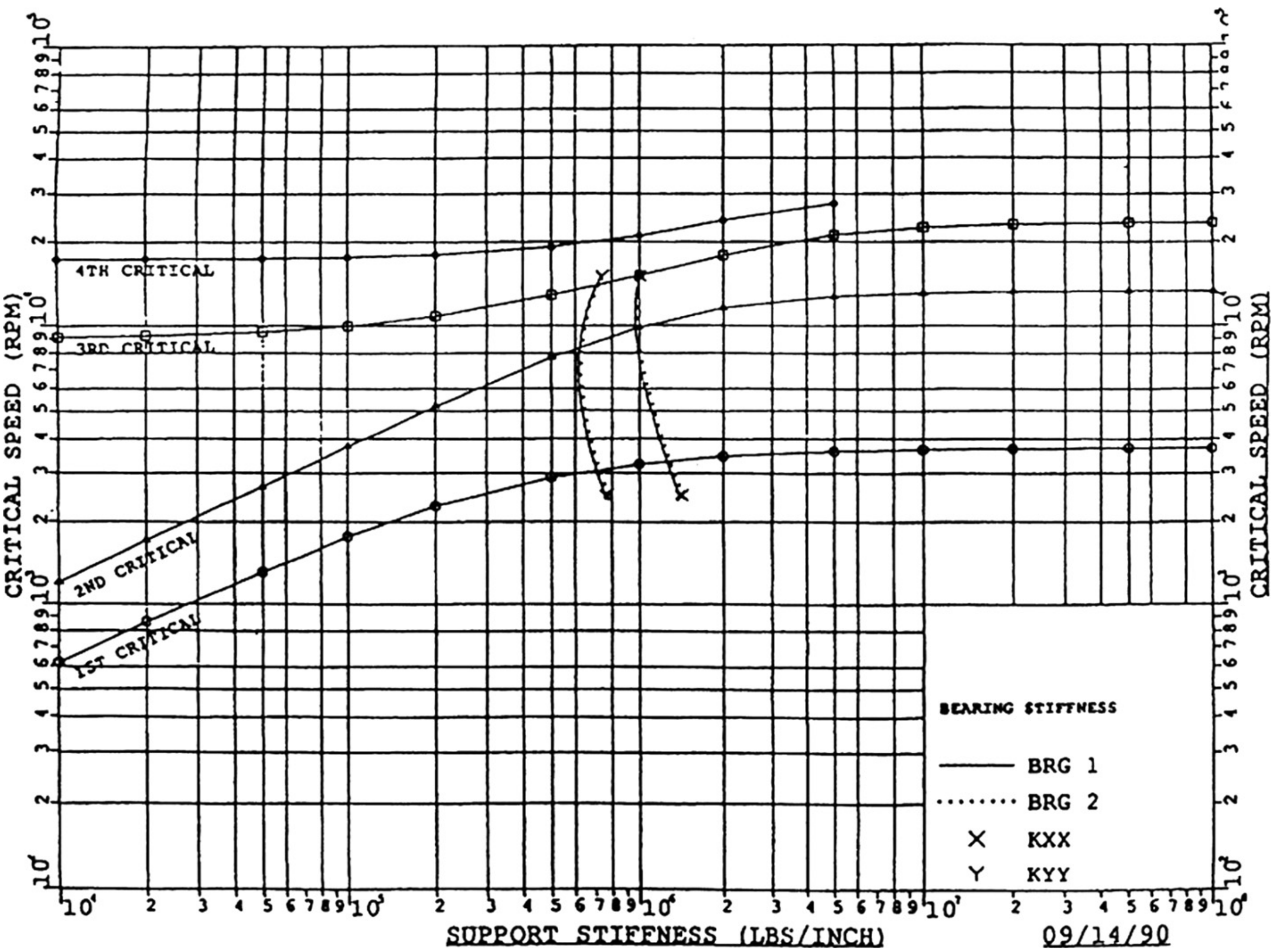


Fig 5.5.1 • Compressor rotor critical speed map – no damping (Courtesy of Elliott Company)

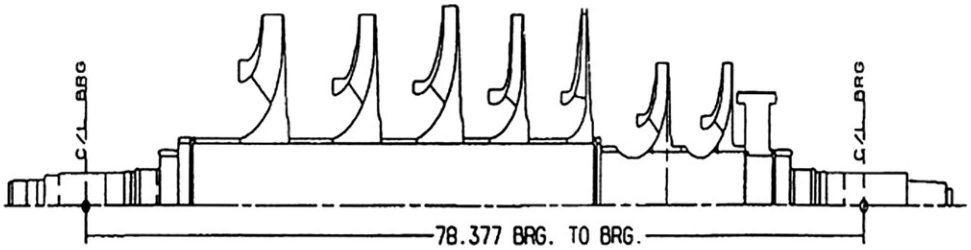


Fig 5.5.2 • Rotor response modeling – rotor (Courtesy of Elliott Co.)

It should be understood that all stiffness values are 'calculated' and will vary under actual conditions. As an exercise, determine NC_1, NC_2 and NC_3 for the horizontal and vertical directions for each bearing in Figure 5.5.1 (assume bearing 1 and 2 stiffness are the same).

Critical speed	Horizontal (X)	Vertical (Y)
NC_1	3,300 rpm	3,000 rpm
NC_2	9,700 rpm	8,000 rpm
NC_3	16,000 rpm	15,000 rpm

Based on a separation margin of ±20% from a critical speed, what would be the maximum allowable speed range between NC_1 and NC_2 in Figure 5.5.1?

- Maximum speed 6,600 rpm
- Minimum speed 4,000 rpm

Remember, changing any value of support stiffness will change the critical speed. Plotted on the x axis is support stiffness in lb/inch. The primary components of support stiffness in order of decreasing increasing influence are:

- Oil support stiffness
- Bearing pad or shell
- Bearing housing
- Bearing bracket
- Casing support foot
- Baseplate
- Foundation

Note that this analysis of the critical speed does not include oil film damping. It is common practice to first determine the 'undamped critical speeds' to allow for necessary modifications to the rotor or support system. This is because the effects of stiffness on the location of critical speed are significantly greater than damping. Figure 5.5.2 shows four distinct critical speeds. Operation within ± 20% of actual critical speeds is to be avoided. Also plotted are the horizontal (x) and vertical (y) bearing stiffness for each bearing. Note that these values vary with speed and are the result of changes in the oil stiffness. Therefore, a change in any of the support stiffness components noted above can change the rotor critical speed. Experience has shown that critical speed values seldom change from ± 5% of their original installed values.

If a turbo-compressor with oil seals experiences a significant change in critical speeds, it is usually an indication of seal lock-up. That is, the seal does not have the required degrees of freedom and supports the shaft acting like a bearing. Since the seal span is less than the bearing span, the rotor stiffness 'K' increases and the critical speeds will increase in this case.

The rotor system (input)

Figure 5.5.2 shows a typical turbo-compressor rotor before modeling for critical speed or rotor response analysis.

Since the natural frequency or critical speed is a function of shaft stiffness and mass, Figure 5.5.4 presents the rotor in Figure 5.5.2 modeled for input to the computer rotor response program.

Figure 5.5.4 is an example of a modeled rotor and only includes the rotor stiffness (K) and mass (M).

In order to accurately calculate the rotor critical speeds, the entire rotor system stiffness, masses and damping must be considered. Figure 5.5.3 models the oil film stiffness and damping of the journal bearings at different shaft speeds.

Note that it is essential that the type of oil to be used in the field (viscosity characteristics) must be known. End users are cautioned to confirm with the OEM before changing oil type as this will affect the rotor response. In addition to modeling of the rotor and bearings, most rotor response calculations also include the following additional inputs:

- Bearing support stiffness
- Oil film seal damping effects

Of all the input parameters, the effects of bearing and seal oil film parameters are the most difficult to calculate and measure. Therefore, a correlation difference will always exist between the predicted and actual values of critical speed. Historically, predicted values of NC_1 (first critical speed) generally agree within ± 5%. However, wide variations between predicted and

4X1.6" tilt 20.5" TB 3.0" shaftend 7.5–6.5" shaft Bendix coupling

Static bearing load (lbs)	897	diameter (inches)	4.00
Bearing station	12	length (inches)	1.60
Bearing location	thrust	diam assembly clearance (inches)	5.7487E–03
Bearing type	tilt pad	diam machined clearance (inches)	8.7500E–03
Location of load	between pads	inlet oil temperature (deg F)	120.0
Preload	0.343	type of oil	DTE–light (150SSU @100°F)

Fig 5.5.3A • Typical compressor oil film bearing parameters (Courtesy of Elliott Co.)

		Fluid film stiffness		Damping	
Speed (rpm)	50mm No	KXX (lb/in)	KYY (lb/in)	WCXX (lb/in)	WCYY (lb/in)
2500	0.114	1.3871E 06	7.5446E 05	7.7995E 05	4.6249E 05
3000	0.137	1.2984E 06	7.1330E 05	7.8487E 05	4.7587E 05
4000	0.183	1.1769E 06	6.6147E 05	8.0311E 05	5.0825E 05
4500	0.206	1.1341E 06	6.4543E 05	8.1400E 05	5.2564E 05
5500	0.252	1.0703E 06	6.2556E 05	8.3686E 05	5.6116E 05
6613	0.303	1.0230E 06	6.1679E 05	8.6656E 05	6.0354E 05
7000	0.321	1.0109E 06	6.1616E 05	6.7775E 05	6.1885E 05
8000	0.366	9.8751E 05	6.1898E 05	9.0798E 05	6.5935E 05
9000	0.412	9.7305E 05	6.2684E 05	9.4015E 05	7.0111E 05
10000	0.458	9.6556E 05	6.3864E 05	9.7461E 05	7.4430E 05
11000	0.504	9.6360E 05	6.5354E 05	1.0110E 06	7.8878E 05
12000	0.549	9.6610E 05	6.7094E 05	1.0490E 06	8.3434E 05
13000	0.595	9.7225E 05	6.9037E 05	1.0881E 06	8.8080E 05
14000	0.641	9.8144E 05	7.1149E 05	1.1283E 06	9.2801E 05
15000	0.687	9.9317E 05	7.3403E 05	1.1696E 06	9.7586E 05

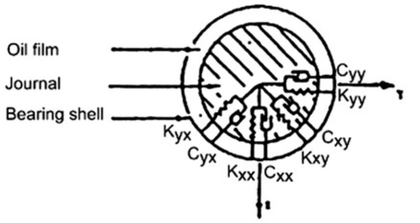


Fig 5.5.3B • Continued – Typical compressor oil film bearing parameters (Courtesy of Elliott Co.)

actual values above the first critical speed (NC_1) exist for NC_2, NC_3, etc.

When selecting machinery, the best practice is to request specific vendor experience references for installed equipment with similar design parameters as follows:

- Bear span ÷ major shaft diameter
- Speeds
- Bearing design
- Seal design
- Operating conditions (if possible)

Once the rotor system is adequately modeled, the remaining input parameter is the amount and location of unbalance. Since the objective of the rotor response study is to accurately predict the critical speed values and responses, an assumed value and location of unbalances must be defined. Other than bearing and seal parameters, unbalance amount and location is the other parameter with a 'correlation factor'. There is no way to accurately predict the amount and location of residual unbalance on the rotor. Presently, the accepted method is to input a value of 8 x A.P.I. acceptable unbalance limit $\frac{(4W)}{N}$.

This results in a rotor response input unbalance of $\frac{32W}{N}$.

The location of the unbalance is placed to excite the various critical speeds. Typically the unbalances are placed as noted below:

Location	To excite
Mid span	NC_1
Quarter span (2 identical unbalances)	NC_2
At coupling	NC_2, NC_3

Failure to accurately determine the value and location of residual rotor unbalance is one of the major causes of correlation differences between predicted and actual critical speeds.

Rotor response (output)

The output from the rotor response study yields the following:

- Relative rotor mode shapes
- Rotor response for a given unbalance

Figure 5.5.5 shows the relative rotor mode shapes for NC_1, NC_2, NC_3 and NC_4. Usually, the rotor will operate between NC_1 and NC_2.

Rotor mode shape data is important to the designer because it allows determination of modifications to change critical speed values.

For the end user, this data provides an approximation of the vibration at any point along the shaft as a ratio of the measured vibration data. As an example in Figure 5.5.5, determine the vibration at the shaft mid span if the vibration measured by

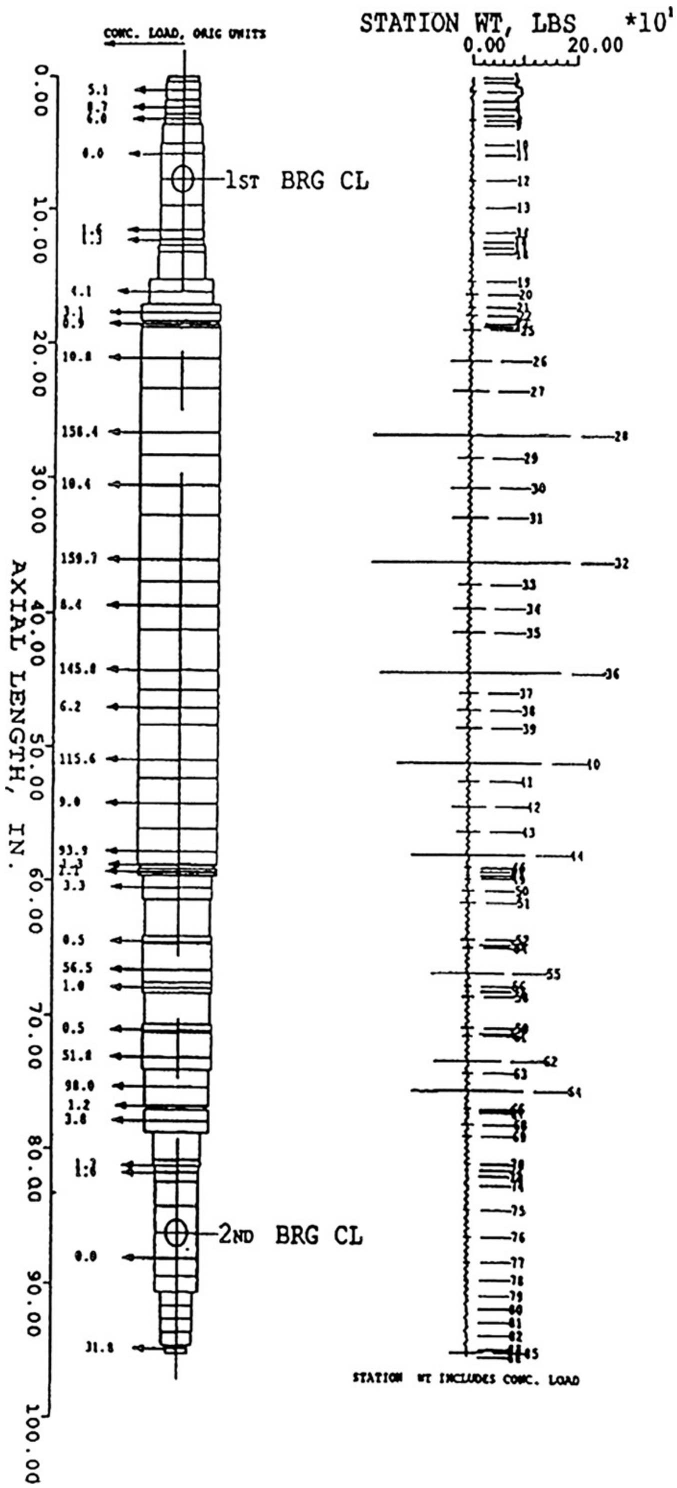


Fig 5.5.4 • Rotor response input data – dimensions, masses and unbalances (Courtesy of Elliott Co.)

the probe C_2 when operating at NC_1 is 2.00 mils. From Figure 5.5.5, the vibration at the shaft mid span when operating at the first critical speed of 3327 rpm (50 in location) is:

$$\frac{1.00}{.1} \text{ or } 10 \times \text{the bearing vibration}$$

Ten (10) times the value at C_2 or 20.0 mils!

Mode shape data should always be referred to when vibration at operating speed starts to increase and your supervisor asks

'When do we have to shutdown the unit?'

or

'Can we raise the radial vibration trip setting?'

In this example, the bearing clearance may be 0.006 or 6 mils, and an honest request would be 'We'll replace the bearing at the turnaround, please run to 7.0 mils vibration'.

Refer to Figure 5.5.5 and remember:

- The compressor must go through NC_1
- The shaft vibration **increases** at NC_1 (usually 2×, 3× or more)
- The vibration at center span is approximately **10×** the probe vibration

Therefore:

Vibration at the mid span during the first critical speed will be:

$$= (7.0 \text{ mils}) \times (2.0) \times (10)$$
$$\text{Probe value} \quad NC_1 \text{ amplification} \quad \text{Mode shape difference}$$
$$= 140 \text{ mils!!}$$

Normal clearance between the rotor and interstage labyrinths is typically 40 mil! This vibration exposes the diaphragms, which are usually cast iron, to breakage. One final comment; during shutdown, the rate of rotor speed decrease **CANNOT** be controlled as in the case of start-up. It depends on rotor inertia, load in the compressor, the process system characteristics and the control and protection system. If the vibration at the probe locations is high, the best advice is to stop the compressor while fully loaded, which will reduce the time in the critical speed range as much as possible. Yes, the compressor will surge, but the short duration will not normally damage it. Figures 5.5.6 and 5.5.7 present the primary output of a rotor response study.

Rotor response plots display vibration amplitude, measured at the probes, vs. shaft speed for the horizontal and vertical probes. Note that a response curve must be plotted for each set of unbalance locations and unbalance amount.

Figure 5.5.6 shows the rotor response for the non-drive end (NDE) set of probes with the first set of unbalance. Figure 5.5.7 shows the rotor response for the drive end set of probes (DE). The operating speed range of this example is 6,000–8,000 rpm.

Measured rotor response

During shop test, the rotor response of every turbo-compressor rotor is measured during acceleration to maximum speed and deceleration to minimum speed. Values are plotted on the same coordinates as for the rotor response analysis. The plot of shaft vibration and phase angle of unbalance vs. shaft speed is known as a **bode plot**.

Bode plots represent the actual signature (rotor response) of a rotor for a given condition of unbalance, support stiffness and unbalance. They indicate the location of critical speeds, the

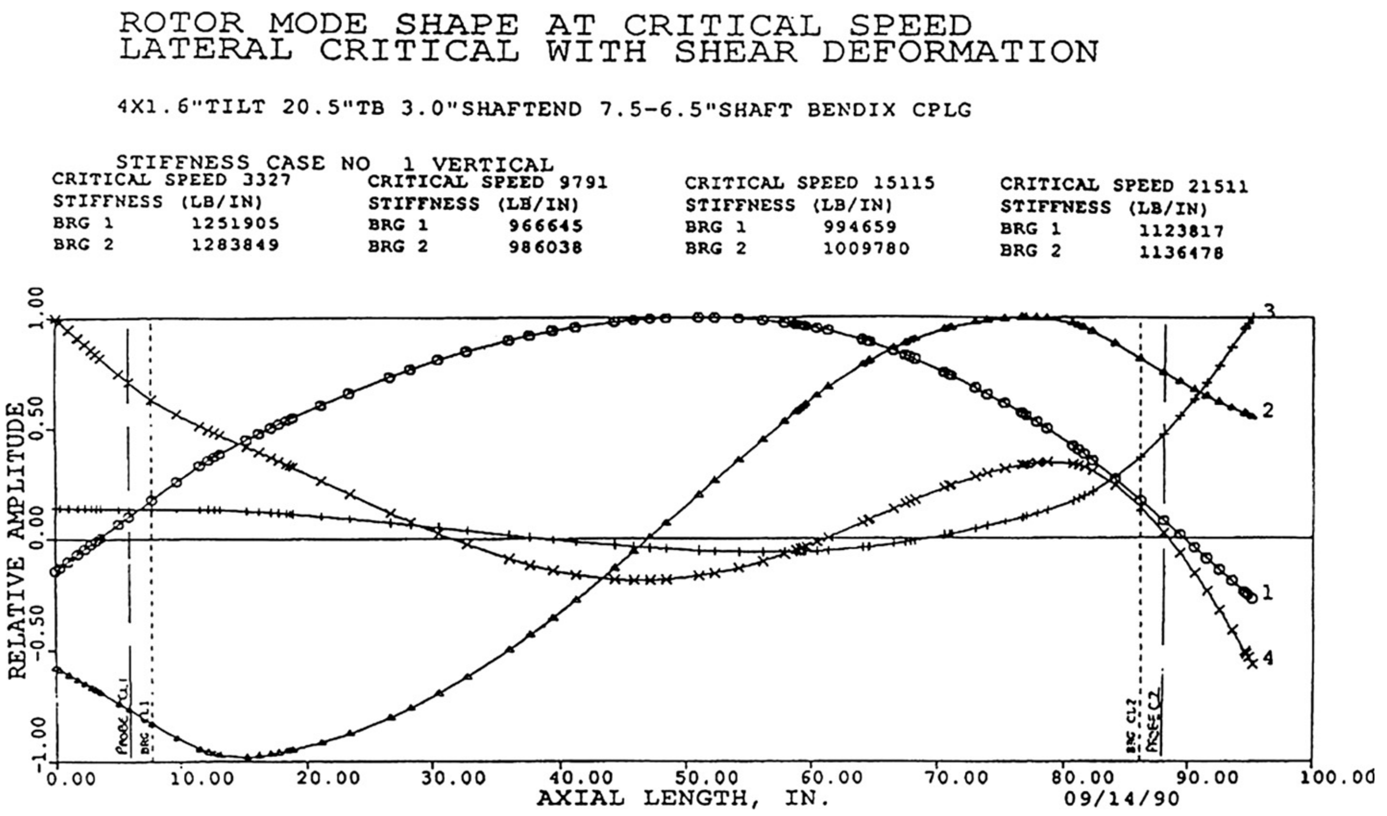


Fig 5.5.5 • Rotor natural frequency mode shapes (Courtesy of Elliott Co.)

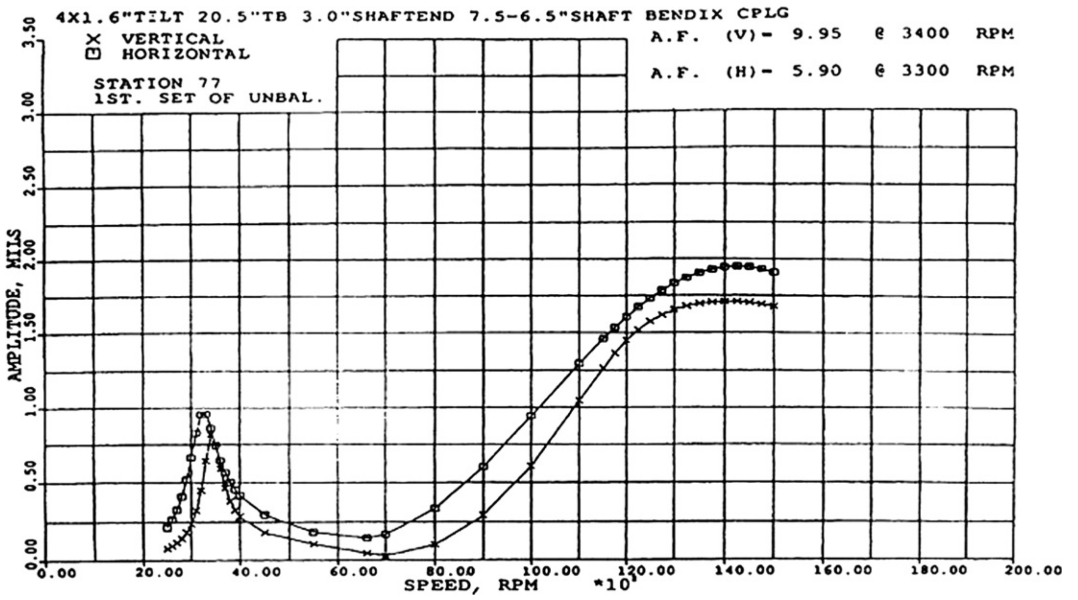


Fig 5.5.6 • Rotor response output at non-drive end bearing (NDE) (Courtesy of Elliott Co.)

change of shaft vibration with speed and the phase angle of unbalance at any speed. A bode plot is a dynamic or transient signature of vibration for a rotor system and is unique to that system for the recorded time frame. Bode plots should be recorded during every planned start-up and shutdown of every turbocompressor. As discussed in this section, the bode plot will provide valuable information concerning shaft vibration and phase angle at any shaft speed.

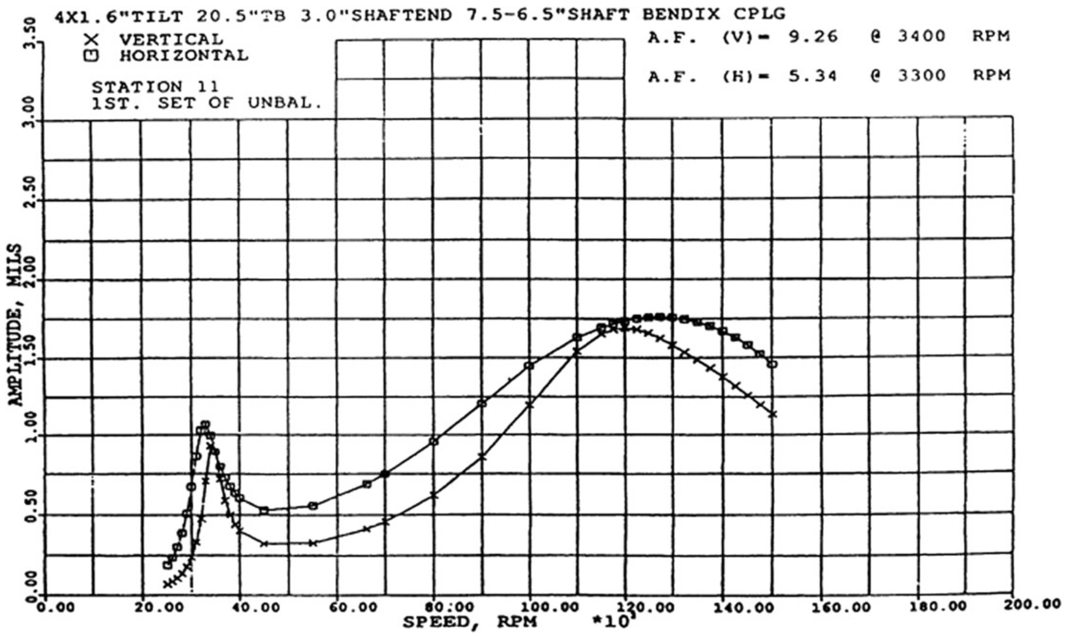


Fig 5.5.7 • Rotor response output drive end bearing (D.E.) (Courtesy of Elliott Co.)

Best Practice 5.6

Use tilting pad radial bearings, and not lemon or offset sleeve anti-whirl type bearings to positively eliminate vibration instabilities.

Tilting pad radial bearings provide vibration stability at any load angle.

Lemon bore (elliptical) or offset sleeve (to achieve an elliptical arrangement) bearings do not eliminate vibration instabilities if the load angle lies in the major axis of the ellipse, since the oil film stiffness in this region may not be sufficient to prevent vibration instabilities.

Lessons Learned

Lemon bore or offset sleeve bearings have caused extended FAT periods necessary to modify bearing split line orientation or changes to three or four lobe or tilt pad bearings.

If bearing instabilities are experienced, a possible modification is to rotate the major axis of the ellipse to provide sufficient oil film stiffness at the load angle. This procedure takes time and must be repeated in the field each time the machine is disassembled.

If the above procedure is not successful, installation of multi lobe or tilting pad bearings will be required. This procedure will be time consuming and can delay tests for months.

Benchmarks

This best practice has been used since the early 1980s, when offset bearings were required to be changed to multi-lobe bearings during the FAT. Delivery delay was 2 months. It should be noted that the vendor made the modified multi lobe bearing standard on all subsequent turbines. Use of this best practice has resulted in trouble free turbine operation and reliabilities exceeding 99.5%.

B.P. 5.6. Supporting Material

Hydrodynamic bearings

Hydrodynamic bearings support the rotor using a liquid wedge formed by the motion of the shaft (see Figure 5.6.1).

Oil enters the bearing at supply pressure values of typically 103-138 kPa (15-20 psig). The shaft acts like a pump which increases the support pressure to form a wedge. The pressure of the support liquid (usually mineral oil) is determined by the area of the bearing by the relationship:

$$P = \frac{F}{A}$$

Where: P = Wedge support pressure (P.S.I.)
F = Total bearing loads (static and dynamic)
A = Projected bearing area ($A_{PROJECTED}$)
$A_{PROJECTED} = L \times d$
Where: L = Bearing axial length
d = Bearing diameter

As an example, a 4" diameter bearing with an axial length of 2" (L/d = 0.5) would have $A_{PROJECTED} = 8\ in^2$.

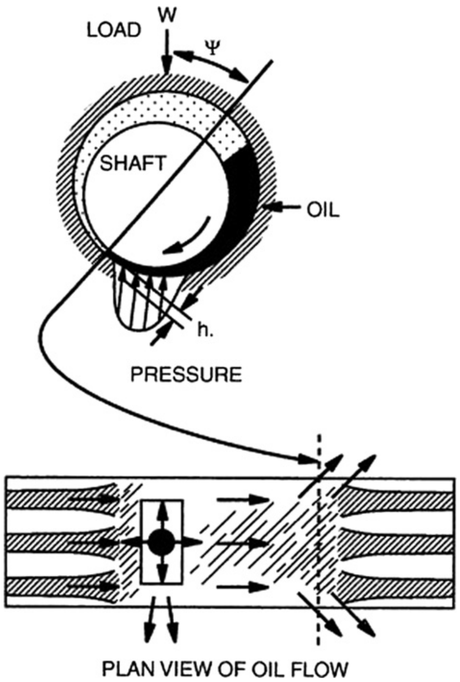


Fig 5.6.1 • Hydrodynamic Lubrication (Courtesy of Bently Nevada Corp.)

If the total of the static and dynamic forces acting on the bearing is 1600 lbs force, the pressure of the support wedge is:

$$P = \frac{1600\ \text{Lb}_{\text{FORCE}}}{8\ \text{in}^2} = 200\ \text{psi}$$

The maximum desired design wedge pressure for oil is approximately 3,450 kPa (500 psi). However, it has been common practice to limit hydrodynamic bearing loads to approximately 1,725 kPa (250 psi) in compressor applications. Figure 5.6.2 is a side view of a simple hydrodynamic bearing showing the dynamic load forces.

The primary force is the load which acts in the vertical direction for horizontal bearings. However, the fluid tangential force can become large at high shaft speeds. The bearing load vector is then the resultant of the load force and fluid tangential force. The fluid radial force opposes the load vector and thus supports the shaft. It has been demonstrated that the average velocity of the oil flow is approximately 47-52% of the shaft velocity. The fluid tangential force is proportional to the journal oil flow velocity. If the fluid tangential force exceeds the load force, the shaft will become unstable and will be moved around the bearing shell. This phenomenon is known as oil whirl.

Hydrodynamic bearing types

Regardless of the type of hydrodynamic bearing, all its surfaces are lined with a soft, surface material made of a composition of tin and lead. This material is known as Babbitt. Its melting temperature is above 204°C (400°F), but under load will begin to deform at approximately 160°C (320°F). Typical thickness of Babbitt over steel is 1.5mm (0.060"). Bearing embedded temperature probes are a most effective means of measuring bearing load point temperature and are inserted just below the Babbitt surface. RTDs or thermocouples can be used. There are many modifications available to increase the load

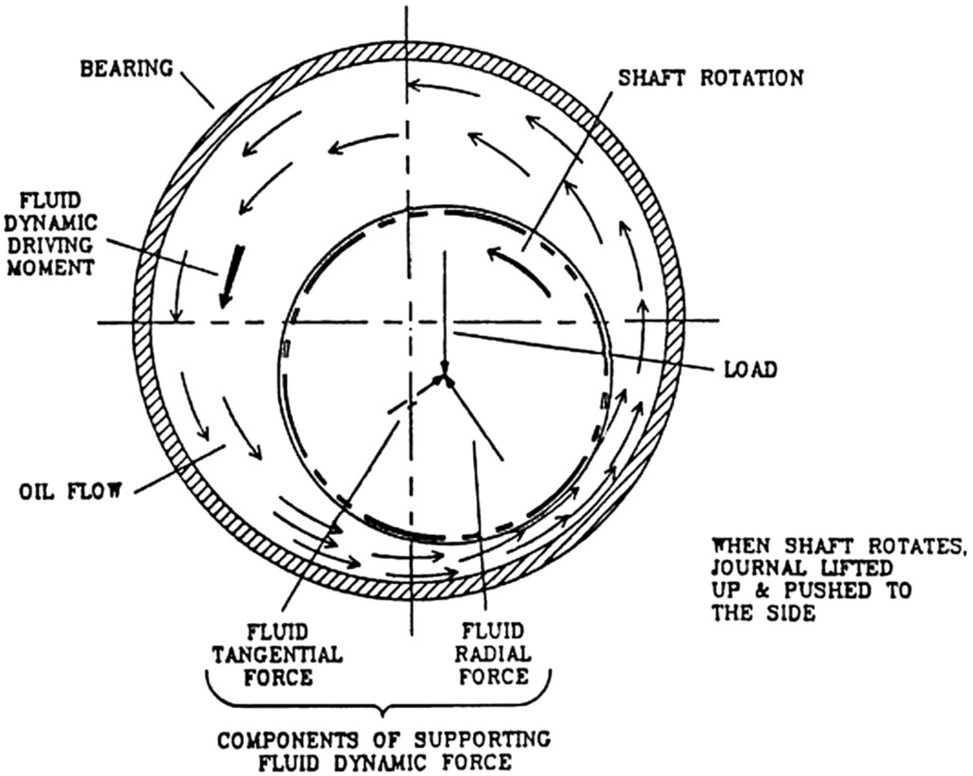


Fig 5.6.2 • Shaft/bearing dynamics (Courtesy of Bently Nevada Corp.)

effectiveness of hydrodynamic bearings. Among the methods available are:

- Copper backed Babbitt or 'Trimetal' – to aid in heat removal
- Back pad cooling – used on tilt pad bearings to remove heat
- Direct cooling – directing cool oil to maximum load points

A typical straight sleeve hydrodynamic journal bearing is shown in Figure 5.6.3.

Straight sleeve bearings are used for low shaft speeds (less than 5,000 rpm) or for older turbo-compressor designs. Frequently, they are modified to incorporate a pressure dam, in the direction of rotation. This pressure dam must be positioned in the top half of the bearing to increase the load vector (see Figure 5.6.2). This ensures that the tangential force vector will be small relative to the load vector, thus preventing shaft instability. It should be noted that incorrect assembly of the pressure dam in the lower half of the bearing will render this type of bearing unstable. When shaft speed is high, other alternatives to prevent rotor instabilities are noted in Figure 5.6.4.

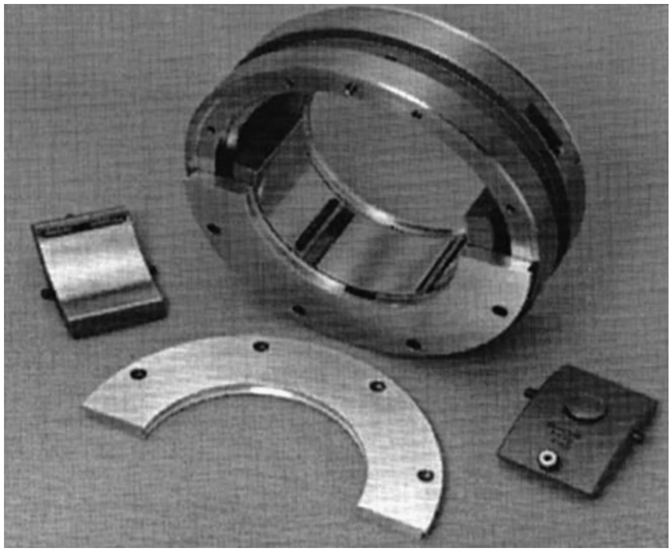


Fig 5.6.5 • Tilting pad journal bearing assembly (Courtesy of Kingsbury, Inc.)

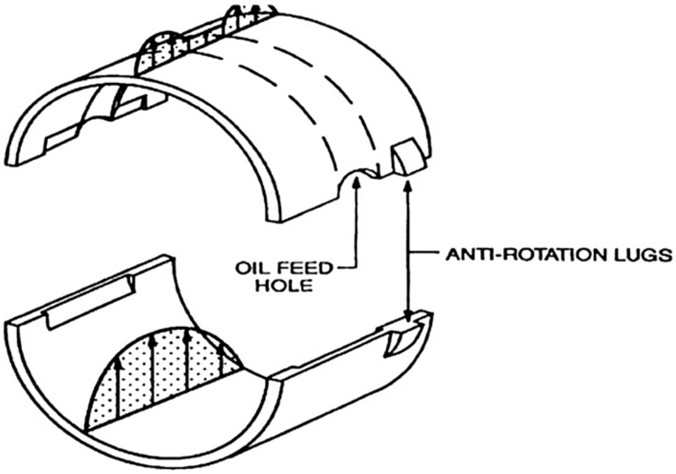


Fig 5.6.3 • Straight sleeve bearing liner (Courtesy of Elliott Co.)

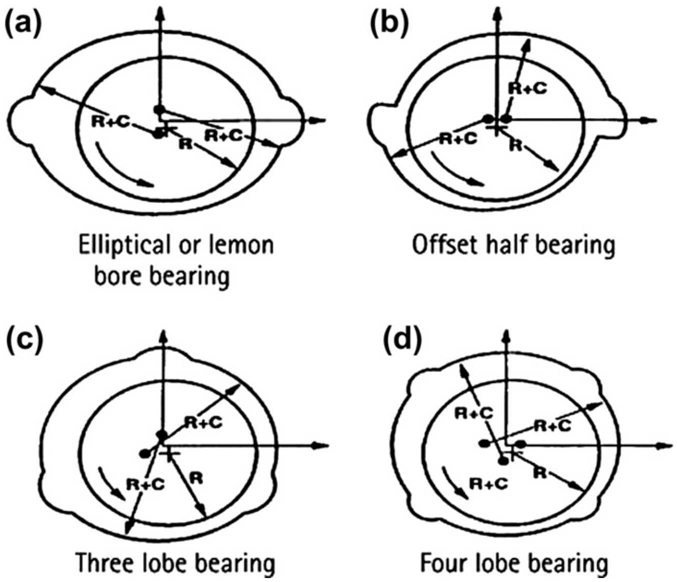


Fig 5.6.4 • Prevention of rotor instabilities

Shown are examples of anti-whirl bearings. The most common types of these bearings are the three and four lobe design. Elliptical and offset bearing designs do prevent instabilities but tend to increase shaft vibration if the load vector passes through the major axis of the bearing. These types of bearings may have to be rotated in the bearing brackets to prevent this occurrence.

The most common hydrodynamic bearing for higher speed applications is the tilt pad journal bearing shown in Figure 5.6.5.

A tilting pad bearing offers the advantage of increased contact area, since the individual pads conform to the shaft orbit. In addition, this type is also a highly effective anti-whirl bearing since the spaces between the pads prevent oil whirl. Most end users specify tilt pad radial and thrust bearings for turbo-compressor applications.

Figure 5.6.6 shows the mechanical frictional losses and oil flow requirements for a tilt pad journal bearing as a function of shaft speed.

Note that the basis for horsepower loss and oil flow is an oil temperature rise of 30°F. This is the normal design ΔT for all hydrodynamic bearings. Also given in this figure is the data necessary to calculate bearing pressure at the load point. As an exercise calculate the following for this bearing:

- Projected Area

$$A_{PROJECTED} = 5'' \times 2'' = 10 \text{ square inches}$$

- Pressure

$$= 3479 \text{ lb force} \div 10 \text{ square inches}$$
$$= 347.9 \text{ psi on the oil film at load point}$$

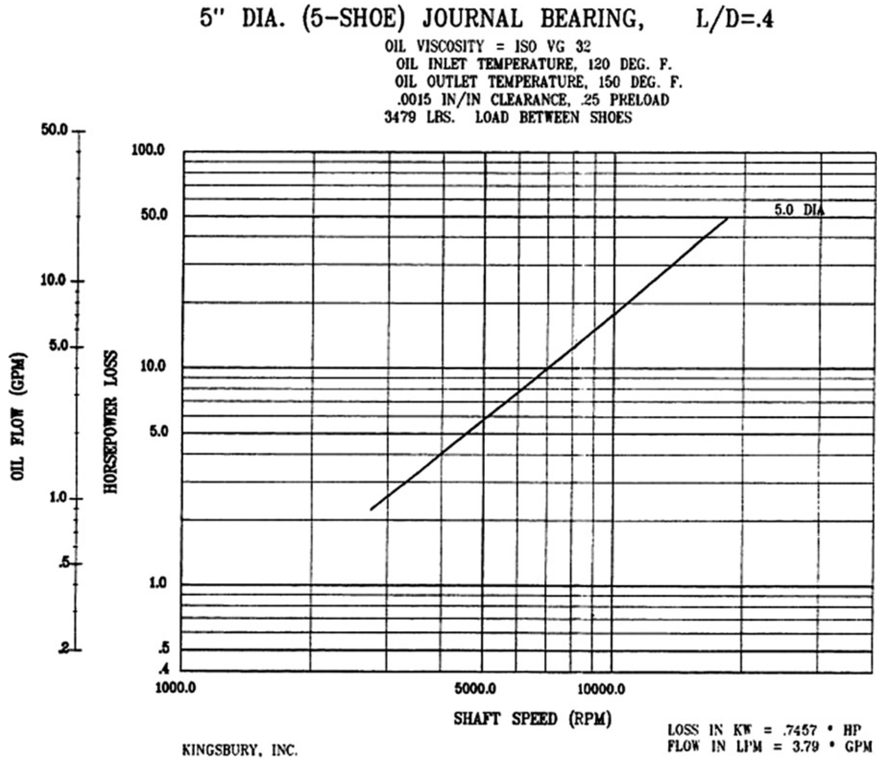


Fig 5.6.6 • Typical journal bearing selection curve (Courtesy of Kingsbury, Inc.)

Best Practice 5.7

Design audit thrust bearing loading for impulse and reaction turbines, to ensure that maximum encountered field thrust pad temperatures do not exceed 110°C and achieve maximum MTBF.

Impulse bladed turbines, which historically did not produce significant thrust loads, have used reaction blades in the last three stages since the 1980s for increased turbine efficiency.

The use of reaction blading in the last stages has significantly increased thrust loads – to the surprise of both end users and designers.

To avoid higher than expected thrust pad temperatures in impulse turbines, consider undercutting the turbine shaft to act as a balance drum, and reduce the load on the thrust bearing during the project pre-bid phase (before vendor final priced proposals are submitted).

In reaction turbine designs, where each blade row produces high thrust, the inlet end steam seal diameter is raised to function as a balance drum. Vendor balance drum calculations should be audited during the pre-bid project phase to ensure proper thrust balance under all load and steam conditions.

Lessons Learned

Many impulse steam turbines designed in the 1980s and 1990s have encountered high thrust bearing pad temperatures. This restricted turbine power and required multiple modifications to reduce thrust pad temperatures to an acceptable level.

While vendor modifications helped reduce thrust loads in impulse turbines, it was not until rotor modifications were made that values were reduced to acceptable levels (below 110°C [230°F] maximum).

Benchmarks

This best practice has been used since the 1990s to finally reduce impulse turbine thrust pad temperatures to acceptable levels without the use of 'band aid' modifications (questionable fixes that may not last). This best practice has produced steam turbines with trouble-free thrust bearing operation and thrust bearing MTBFs exceeding 100 months.

B.P. 5.7. Supporting Material

Rotor thrust balance

Figure 5.7.1 shows how a balance drum or opposed impeller design reduces thrust force. The total impeller force is the sum of the forces from the individual impellers. If the suction side of the impellers is opposed, as noted in Figure 5.7.1, the thrust force will be significantly reduced and can approach zero. If the suction side of all impellers are the same (in series), the total impeller thrust force can be very high and may exceed the thrust bearing rating. If this is the case, a balance drum must be mounted on the rotor as shown in Figure 5.7.1. The balance drum face area is varied such that the opposing force generated by the balance drum reduces the thrust bearing load to an acceptable value. The opposing thrust force results from the differential between compressor discharge pressure (P_F) and compressor suction pressure (P_1) since the area behind the balance drum is usually referenced to the suction of the compressor.

This is accomplished by a pipe that connects this chamber to the compressor suction. This line is typically called the 'balance line'.

It is very important to note that a balance drum is used only where the thrust bearing does not have sufficient capacity to absorb the total compressor axial load. And the effectiveness of the balance drum depends directly on the balance drum seal. Fail the seal (open clearance significantly), and thrust bearing failure can result.

A common misunderstanding associated with balance drum systems is that a balance drum always reduces the rotor thrust to zero. Refer to Figure 5.7.2 and observe that this statement may or may not be true, depending on the thrust balance system design. And even if it is, the thrust is zero only at one set of operating conditions.

Figure 5.7.2 shows a rotor system designed four different ways. Note how the thrust **always** changes with the flow rate regardless of the design. Another misconception regarding thrust balance systems is the normal or 'active' direction of thrust. In many cases, the active thrust is assumed to always be towards the suction of the compressor.

Observing Figure 5.7.2, it is obvious that the 'active' direction can change when the turbo-compressor has a balance drum or is an opposed design. It is recommended that the use of active thrust be avoided where possible and that axial displacement monitors be labeled to allow determination of the thrust direction at all times.

Please refer to Figure 5.7.3 which shows a typical thrust displacement monitor.

These monitors detect thrust position by targeting the shaft end, thrust collar or other collar on the rotor. Usually two or three probes (multiple voting arrangement) are provided to eliminate unnecessary compressor trips. The output of the probes is noted on the monitor as either + (normal) or – (counter). However, this information gives no direct indication of the axial direction of the thrust collar. The following procedure is recommended:

1. With compressor shutdown, push rotor towards the suction and note direction of displacement indicator.
2. Label indicator to show direction towards suction of compressor.

Knowing the actual direction of the thrust can be very useful during troubleshooting exercises in determining the root cause of thrust position changes.

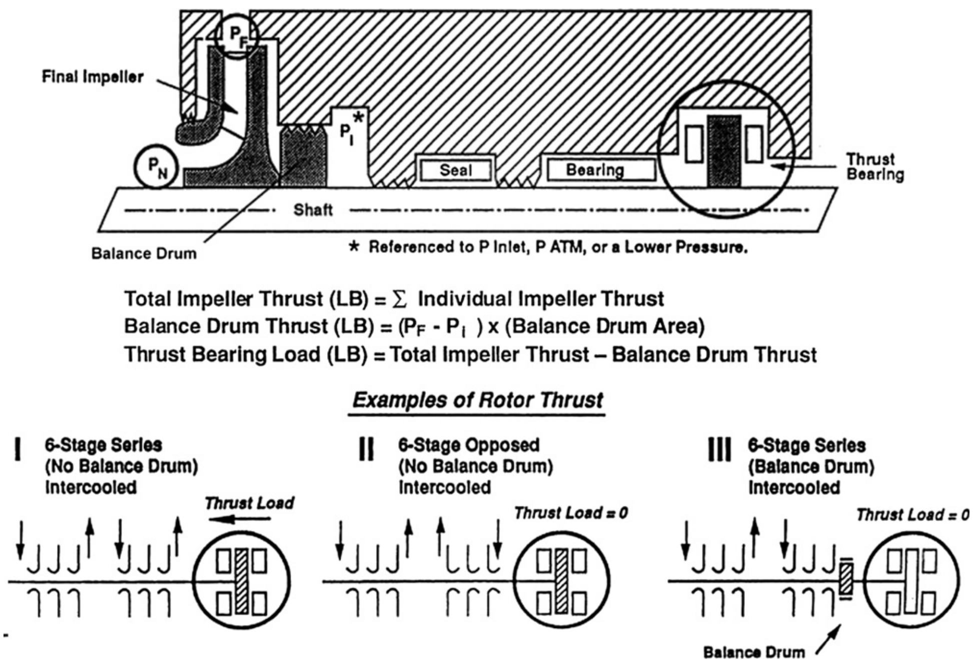


Fig 5.7.1 • Rotor thrust force

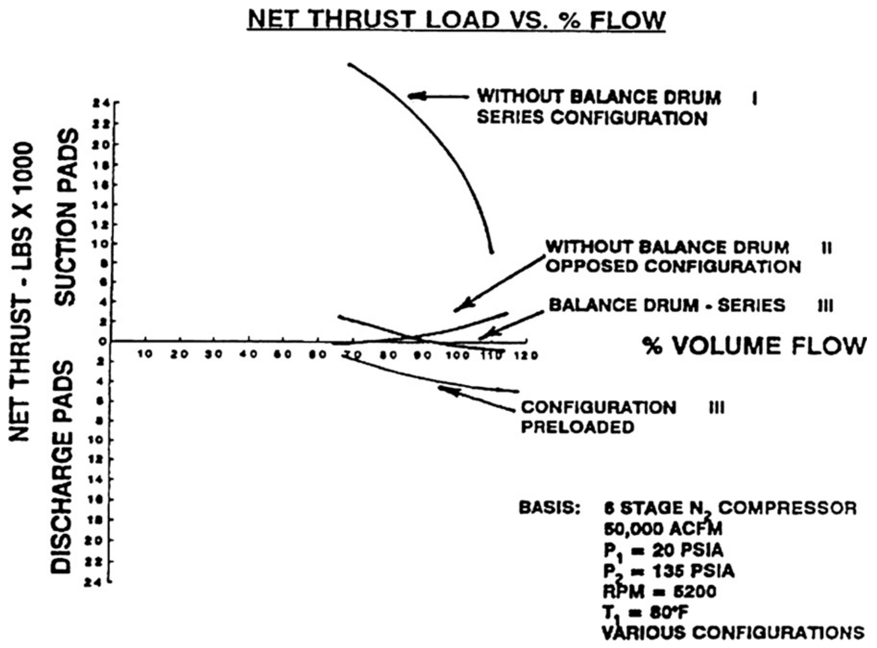


Fig 5.7.2 • Rotor system designed four different ways

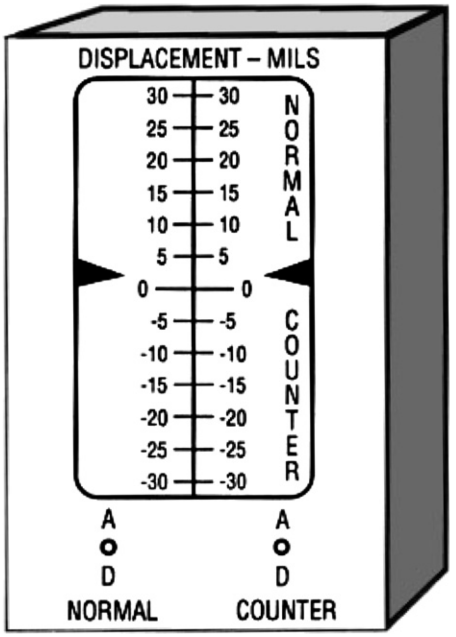


Fig 5.7.3 • Typical axial thrust monitor

Best Practice 5.8

Condition monitor steam seals on large turbines by trending gland condenser vacuum and taking corrective action when vacuum is less than -5 inches water column to positively prevent contamination of the oil system.

Most steam turbine vendors use a vacuum gland condenser system to ensure that air enters the last section of the steam seal, thus preventing stream from entering the bearing bracket and contaminating the oil system.

Monitoring the vacuum of the system will allow determination of necessary maintenance (eductor nozzle change).

Lessons Learned

Failure to monitor gland condenser vacuum on special purpose (un-spared) steam turbines has resulted in gross contamination of the oil systems and reduced bearing life.

In many installations, vacuum gauges in the steam seal system are either missing or broken.

Benchmarks

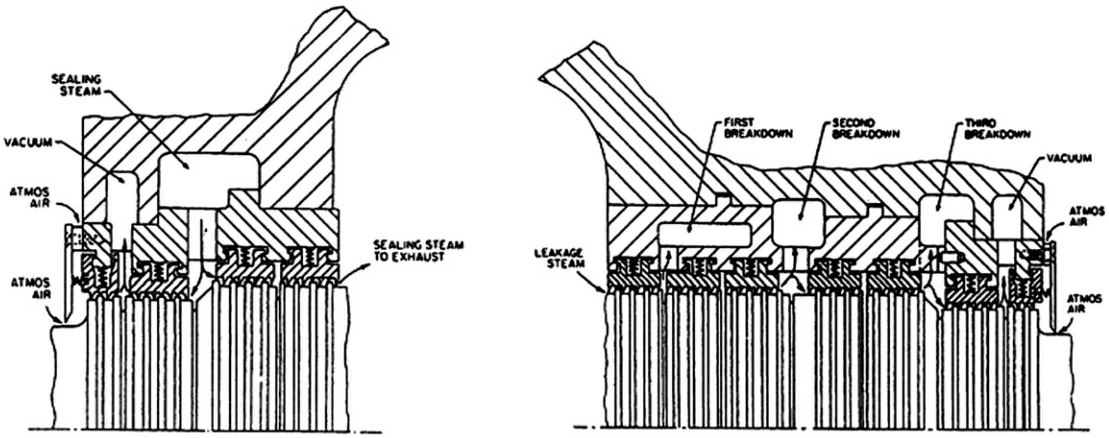
This best practice has been used since the 1990s to prevent excessive contamination of the oil system (greater than 200 ppm of H_2O). When the eductors in the steam seal system are properly monitored and maintained, steam seal MTBFs will exceed 100 months.

B.P. 5.8. Supporting Material

Shaft end seals

Facts concerning shaft end seals and sealing systems for critical service non-condensing and condensing turbines are shown in Figures 5.8.1−5.8.4.

The key to successful shaft end seal operation is to continuously maintain a slight (2-4 cm [5-10"] H_2O) vacuum in the last chamber of the seal. By maintaining a vacuum at this location, atmospheric air will enter the seal thus assuring that steam (moisture) will not enter the bearing bracket and contaminate the oil system.



Function: The labyrinth type shaft end seals safely direct the expansion vapor to a defined location and draw in a buffer gas (air) to prevent expansion vapor contact with the bearings

Fig 5.8.1 • Expansion turbine shaft end seals (Special purpose (unspared) turbines) Left: Typical exhaust end seal. Right: Typical inlet end seal (Courtesy of IMO Industries)

- Function: prevent steam from escaping to atmosphere along the shaft and entering the bearing housing
- Special purpose turbines usually employ a low vacuum (2–4 cm [5–10"] H_2O vacuum) to buffer atmospheric end labyrinth with air
- General purpose turbines usually do not employ a vacuum system and do not totally prevent moisture from entering bearing housing

Fig 5.8.2 • Steam turbine shaft sealing systems

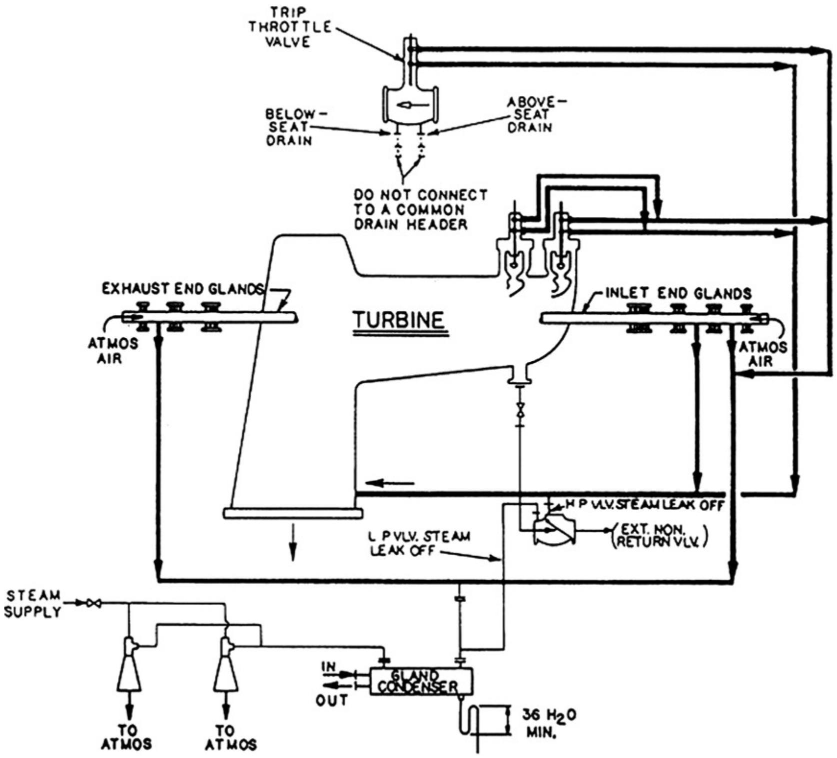


Fig 5.8.3 • Gland seals and drains: noncondensing automatic-extraction turbine (Courtesy of IMO Industries)

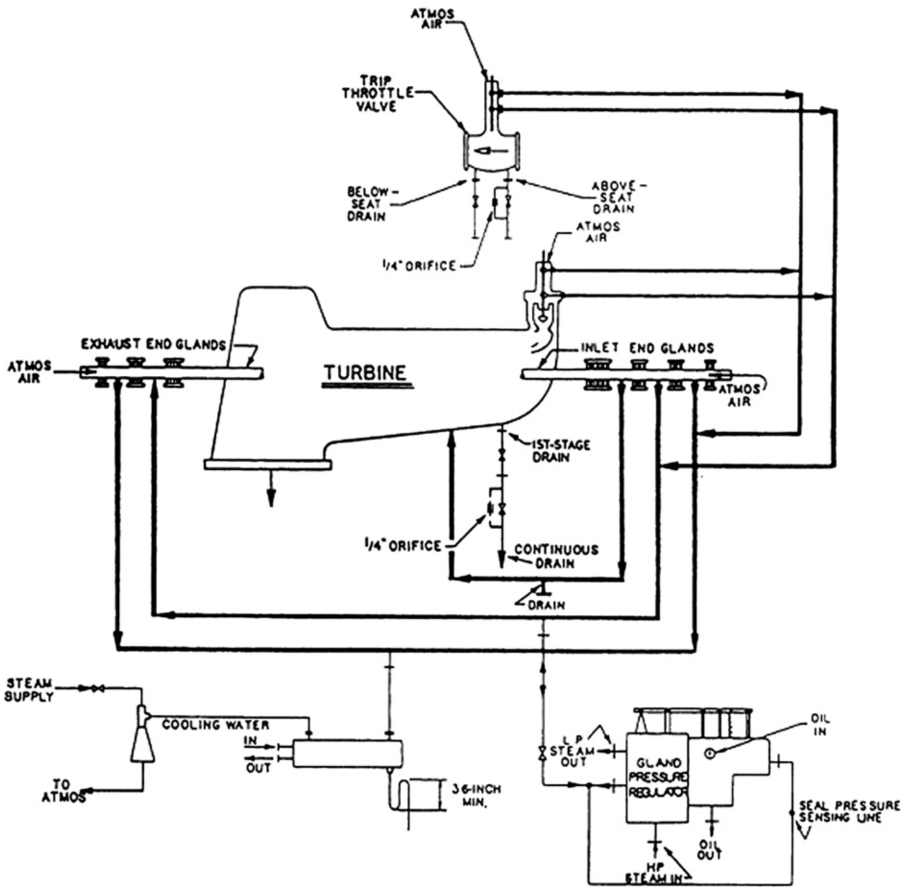


Fig 5.8.4 • Gland seals and drains: condensing turbine (Courtesy of IMO Industries)

Condition monitoring of the system vacuum is essential to maintaining moisture-free lubrication oil. Many a turbine bearing has failed because of poor seal system preventive and predictive maintenance practices. 'THINK SYSTEM' and check all components of the seal system frequently.

Best Practice 5.9

Use dual gland condenser systems in special purpose turbines to positively prevent contamination of lube oil and to optimize steam seal MTBF.

Considering that seal systems in any type of rotating equipment are the number one failure component, it is most surprising that steam seal system reliability is usually neglected.

Most special purpose turbines use a vacuum, eductor gland, condenser system to pull a 2-4 cm (5-10") vacuum in the last set of gland packing, to buffer the seal with atmospheric air in order to prevent steam condensate entering the bearing housing.

Since lube and seal systems used in special purpose (un-spared) equipment trains always use dual elements (pumps, filters, coolers, etc.), the same philosophy should be used for steam seal systems to ensure their proper operation at all times.

Installation of two vacuum eductors will allow a faulty eductor to be switched over and repaired on-line (just like a cooler or filter in an oil system).

Lessons Learned

Most special purpose turbine gland vacuum sealing systems fail during a continuous run of 4-5 years, and are not repaired on-line, resulting in oil system contamination and exposure to bearing failures.

Benchmarks

This best practice has been used for new projects since 2005, when numerous reliability issues were experienced with faulty vacuum eductor systems that were not repaired on-line.

B.P. 5.9. Supporting Material

Please refer to material in B.P. 5.8.

Best Practice 5.10

Always shop test special purpose steam turbines in spite of schedule delays, but do consider a spare rotor high speed balance machine test, to minimize total factory acceptance test (FAT) time when the delivery is behind schedule.

Schedule delays can prompt some vendors to offer field mechanical tests as an alternative to specified shop tests.

Field testing exposes turbines to situations not encountered in FAT testing (piping and foundation stresses, etc.) which lead to decision delays and possible acceptance of less than reliable equipment.

Lessons Learned

Field testing exposes the end user to many issues that can lead to endless discussions regarding acceptability and corresponding start-up delays. The necessity to correct identified field issues can delay start-ups by months.

The initially attractive option of field testing to correct manufacturing schedule delays has proven time and again to be a false hope, which leads to longer delays in field start-ups than would have resulted from a delayed FAT.

Benchmarks

This best practice has been used since the 1980s to ensure on time start-ups and steam turbine field operation of the highest possible reliability (99.7%+).

B.P. 5.10. Supporting Material

Testing phase

The testing phase is the last phase in terms of vendor and sub-supplier design and manufacturing involvement in the project – and so it is the last chance to ensure the optimum availability of the finished product.

Remember that all of the equipment addressed in this section is most likely custom designed, and no matter how much accrued design and manufacturing experience is present, the possibility of some abnormality, hopefully minor, is high. Therefore, it is imperative that this phase be carefully observed and witnessed by the end user team. Figure 5.10.1 lists important facts surrounding this phase of the project.

The shop test is an opportunity to:

- Confirm vendor proper design and manufacture
- Match field conditions
- Witness assembly and disassembly using job special tools
- Have plant personnel observe test, assembly, etc. and take pictures for purposes of emergency field maintenance excellence
- Review the instruction book
- Have the assigned vendor service engineer observe the equipment he will install
- Review all vendor field procedures

Fig 5.10.1 • The shop test

A shop test checklist is included at the end of this best practice that will be valuable in planning and executing the shop test phase. Yes – there certainly are many opportunities to ensure equipment reliability during shop test, but there are also a lot of potential lost opportunities if they are not justified to the project team early, during the pre-FEED phase, of the project. The potential lost shop test phase opportunities are noted in Figure 5.10.2.

The following opportunities will be lost if they are not justified at project inception:

- Possible full load test
- Unproven component tests
- Attendance at test by plant personnel
- Use of special tools
- Vendor permission for pictures
- Agreement that assigned vendor field service specialists will be present for tests
- Agreement that the instruction book is reviewed
- Agreement for formal field construction meeting to clearly define all vendor procedures from receipt of equipment on site to initial run in of equipment

Fig 5.10.2 • Potential lost test opportunities

The success of the shop test depends on a good test plan that is reviewed by the end user and contractor and modified as requested well in advance of the test. Figure 5.10.3 presents these facts.

- The agenda is issued for review two months prior to test
- It incorporates agreed VCM scope
- Compressor performance test conditions are per ASME PTC–10 requirements
- A sample of test calculations and report format is included
- Vendor concurs with all end user and contractor comments prior to test

Fig 5.10.3 • Shop test agenda review – key facts

I actually began my career in rotating equipment on the test floor. And I can still remember how we would see the witnesses come in with an intent to completely participate in the entire test, only to leave for a long 'test lunch' an hour or so later. Why did this occur? Usually because the concerned end user and contractor witnesses did not have the opportunity to review the test set-up and the procedure prior to the test. As a result, I have always been a proponent of a pre-test meeting.

Is it always required? I think it is but the detail and timing of the meeting depends on certain factors. These factors are noted in Figure 5.10.4.

- If the equipment is prototype
- If the equipment is complex
- If a full load test is required
- If the test facility is new

Fig 5.10.4 • When is a pre-test meeting required?

If it is decided to conduct a pre-test meeting, the key facts are noted in Figure 5.10.5.

- Conduct the meeting prior to the test day
- Send the agenda to the vendor well in advance
- A typical agenda outline:
  - Confirm test agenda requirements
  - Confirm all test parameter acceptance limits
  - Confirm instrument calibration
  - Review test set up or concept drawing
  - Review data reduction methods
  - Confirm all test program agreements

Fig 5.10.5 • Pre-test meeting – key facts

Figures 5.10.6 to 5.10.8 define recommended test activity for the mechanical, auxiliary equipment and performance shop tests respectively.

- Per API and project requirements
- Confirm all components are installed
- Confirm all accessories are installed
- Monitor progress of test, look for leaks, etc.
- Do not accept test until all requirements are met

Fig 5.10.6 • Mechanical test — key facts

- Must be per API and project requirements
- Confirm that the test agenda is followed
- Confirm all components are installed
- Confirm that all required instruments are installed
- Monitor the progress of the test – look for leaks, etc.
- Do not accept until all requirements are met

Fig 5.10.7 • Auxiliary system test — key facts

- Confirm mechanical test acceptance
- Confirm auxiliary system test acceptance
- Inspect components and confirm acceptance
- Agree to any corrective action in writing
- Accept or reject test – any corrective action requires a retest!

Fig 5.10.8 • Post test – key facts

At the conclusion of all test activities, there is still important work to be performed. These items are defined in Figure 5.10.8.

What happens if the test is not successful? Approximately 50% of the tests that I have either run or participated in over my career have not been successful in regards to one component or more not meeting test requirements. Possible rejected test action is noted in Figure 5.10.9.

- Immediately provide details to the project team
- Confirm if field conditions can handle the abnormality
- Determine if the 'as tested' machine will meet all reliability requirements
- If the decision is to reject, inform the vendor and detail the reasons
- Do not accept unrealistic delivery delays

Fig 5.10.9 • Rejected test action

Finally, do not forget the importance of test report requirements. The test report is a most important document that represents the 'baseline performance' of the unit and will be a benchmark for field operation acceptability. Test report key facts are noted in Figure 5.10.10.

- The shop test is the field baseline!
- The test report must be detailed and complete
- Review the preliminary contents of the report before leaving the test floor
- Obtain the actual test results
- When the final report is received, check the results obtained at test against the final report
- Immediately contact the vendor if there are any differences

Fig 5.10.10 • Test report – key facts

Shop test checklist

1. Scope

- Appropriate industry specs included (ANSI, API, NEMA, etc.)
- In-house and/or E&C specs included
- Project specific requirements
  - ☐ Performance test ☐ All rotors ☐ One rotor
  - ☐ Test (equivalent) conditions
  - ☐ Field (actual) conditions
  - ☐ Mechanical test ☐ All rotors ☐ One rotor
  - ☐ Test (equivalent) conditions
  - ☐ Field (actual) conditions
  - ☐ Unit test of all equipment (string test)
    - ☐ No load ☐ Includes auxiliary systems
    - ☐ Full load ☐ Does not include auxiliary systems
  - ☐ Use of job couplings and coupling guards
  - ☐ Testing of instrumentation, control and protection devices
  - ☐ Auxiliary system test
    - ☐ Lure oil ☐ Test press ☐ Full press
    - ☐ Control oil ☐ Test press ☐ Full press
    - ☐ Seal oil ☐ Test press ☐ Full press
    - ☐ Seal gas ☐ Test press ☐ Full press
    - ☐ Fuel ☐ Test press ☐ Full press
    - ☐ Flow measurement required
    - ☐ Time base recording of transient events required
  - ☐ Use of all special tools during test (rotor, removal, coupling, etc.)
  - ☐ Shop test attendance (includes assembly and disassembly)
    - ☐ Site reliability ☐ Site maintenance ☐ Site operations
    - ☐ Review of instruction book during shop test visit
    - ☐ Test agenda requirements
    - ☐ Mutually agreed limits for each measured parameter
    - ☐ Issue for approval two months prior to contract test date

2. Pre-test meeting agenda

- Meet with test department prior to test to:
  - ☐ Confirm test agenda requirements
  - ☐ Confirm all test parameters have mutually agreed established limits
  - ☐ Review all instrument calibration procedures

- ☐ Review test set-up drawing
- ☐ Review data calculation (data reduction) methods
- ☐ Define work scopes for site personnel (assembly and disassembly witness, video or still frame pictures, etc.)
- ☐ Confirm assigned vendor service engineers will be in attendance for:
  - ☐ Assembly
  - ☐ Disassembly
  - ☐ Test

Shop test activity

- ■ Review and understand test agenda prior to test
- ■ Immediately prior to test meet with assigned test engineer to:
  - ☐ Review schedule of events
  - ☐ Designate a team leader
  - ☐ Confirm test team leader will be notified prior to each event
  - ☐ 'Walk' test set-up to identify each instrumented point
  - ☐ Confirm calibration of each test instrument
  - ☐ Obtain documents for data reduction check – if applicable (flow meter equations, gas data, etc.)
- ■ During test
  Note: coordinate with test personnel to avoid interference
  - ☐ Review 'as measured' raw data for consistency
  - ☐ 'Walk' equipment – look for leaks, contract instrument, piping and baseplate vibration, etc.
  - ☐ Use test team effectively – assign a station to each individual
  - ☐ Ask all questions now, not later, while an opportunity exists to correct the problem
  - ☐ Check vendor's data reduction for rated point – if applicable
- ■ After test
  - ☐ Inspect all components as required by the test agenda (bearings, seals, labyrinths, RTD wires, etc.)
  - ☐ Review data reduction of performance data corrected to guarantee conditions
  - ☐ Review – all mechanical test data
  - ☐ Generate list of action (if applicable) prior to acceptance of test
  - ☐ Approve or reject

Best Practice 5.11

Perform coupled overspeed trip checks for turbines with electronic governors and overspeed trip systems.

Checking overspeed trip systems with turbines uncoupled exposes personnel and nearby assets to safety issues, since steam energy compared to the uncoupled power requirement can accelerate the turbine to speeds which exceed the rotor maximum design stress.

Electronic governor systems allow the overspeed trip setting to be reduced, which permits confirmation of trip system function without exposing personnel to dangerous consequences.

Lessons Learned

Machinery historical case studies are full of examples of failed turbines and personnel injury resulting from the failure of turbine overspeed trip devices during the uncoupled overspeed trip checks.

Benchmarks

This best practice has been recommended to clients since 2005, when insurance companies accepted it for turbines with electronic governors over uncoupled overspeed trip checks. Note: Since the electronic governor is supplied with overspeed detection trip capability, the mechanical trip bolts can be disabled and removed if an additional independent electrical trip system is used ("Protech", "Turbosentry" or equivalent).

B.P. 5.11. Supporting Material

Total train control and protection objectives

Figure 5.11.1 presents the total train control and protection objectives.

Regardless of the type of driven equipment, the objective of the control and protection system is to ensure that the required quantity of product or generated power is continuously

- ■ Meet driven equipment control requirements
- ■ Compressor – pressure or flow
- ■ Pump – pressure, flow or level
- ■ Generator – load
- ■ Meet above objectives when in series or parallel with other trains
- ■ Continuously protect entire train from damage due to:
  - • Overspeed
  - • Loss of auxiliaries
  - • Component mechanical failure
  - • Driven equipment upsets (surge, minimum flow, high load, etc.)

Fig 5.11.1 • Total train control/protection objectives

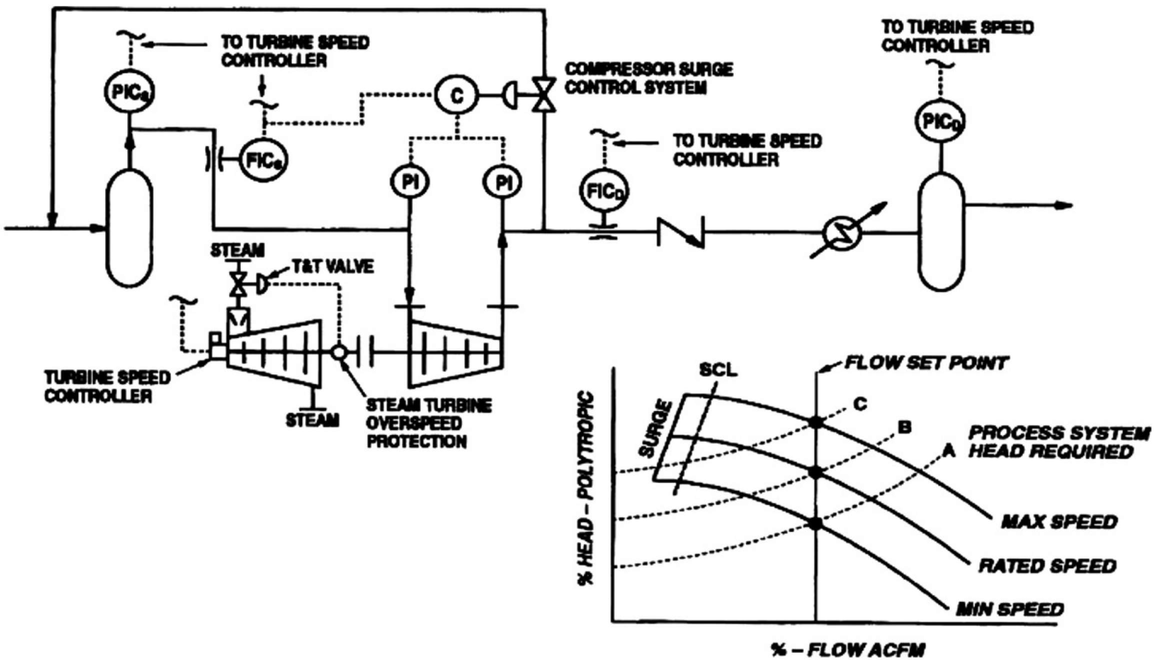


Fig 5.11.2 • Total train control

supplied, maintaining the highest possible total train efficiency and reliability.

Figure 5.11.2 presents a process diagram for a steam turbine driven compressor train.

Depending on the selected process variable and location, any PIC or FIC will continuously monitor the selected process variable, sending its signal as an input to the turbine speed controller. For this example, assume the set point is a flow controller located in the discharge line of the turbo-compressor (FIC_D). The process system head (energy) requirements A, B, C are shown. These different energy requirements can represent either increased pressure ratio requirements (suction strainer blockage exchanger ΔP, etc.) and/or gas density changes (M.W. P or T). As the process head (energy) requirements increase from A to B to C, the input flow variable will decrease if the turbo-compressor speed does not change. However, as soon as the monitored process variable, $FIC_D \neq$ flow set point, the turbine speed controller output will open the turbine inlet throttle valves. This provides more turbine power to increase the head (energy) produced by the compressor, to meet the additional process system head requirements, and therefore maintain the desired throughput.

Adjusting the speed of the driven equipment is the most efficient control method, since there no control valves are required in the system. Therefore only the exact value of head required by the process system is produced by the turbo-compressor.

Figure 5.11.2 shows the two major protection systems for the compressor and steam turbine: the surge protection and turbine overspeed protection systems. The surge system has been discussed previously, and the turbine overspeed system will be

- The governor is the heart of the control system
- The governor in simple terms compares input signal(s) to a set point and sends an output signal to achieve the desired set point.
- An example of a simple governor system is ‘cruise control’ in a car

Fig 5.11.3 • Control

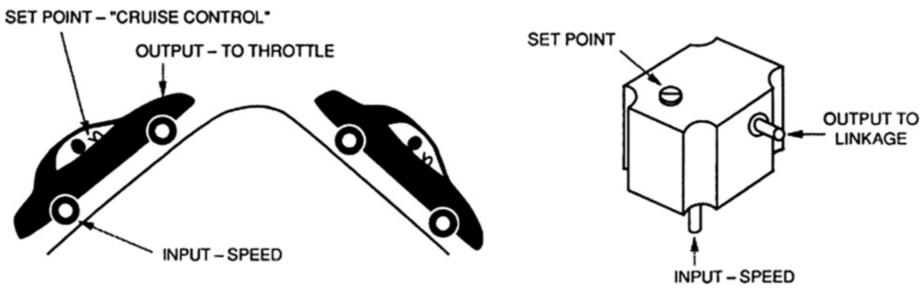


Fig 5.11.4 • A control system analogy. Left: Cruise control. Right: Steam turbine governor. In both cases, load change is inversely related to speed change. The controller compares input to set point and changes output appropriately.

discussed later in this chapter. In addition to the two major protection systems mentioned above, other typical protection systems for a rotating equipment train are:

- Shaft vibration
- Bearing bracket vibration
- Axial thrust displacement
- Bearing temperature
- Process gas temperature
- Lube oil pressure
- Seal oil ΔP
- Suction drum high liquid level (compressors)

Control

A turbine governor is a speed controller. Important facts concerning expansion turbine governors are shown in Figure 5.11.3.

Regardless of type, all controllers have three identical parameters:

- Input
- Set point
- Output

Some familiar controllers are:

- Pressure
- Flow
- Level
- Temperature
- Surge
- Speed

As an example, refer to Figure 5.11.4 – a speed controller that may be familiar.

In Figure 5.11.4, we compare an auto 'cruise control' to a steam turbine governor (typical single stage mechanical/hydraulic). Both are speed controllers and have an:

- Input
- Set point
- Output

The table below shows a comparison of these parameters.

Parameter	C.C. (Cruise control)	T.G. (Turbine governor)
Input	Actual speed from speedometer	Actual speed from speed pick-up
Set point	Selected by driver	Selected by operator
Output	To fuel control system	To steam throttle valve

Figure 5.11.5 is a schematic of a steam turbine governor system.

Note that the set point can either be a manual set point, similar to a driver setting a "speed" in a cruise control system or a process variable. Examples of process variable set points would be:

- Pressure
- Flow
- Level (pump applications)

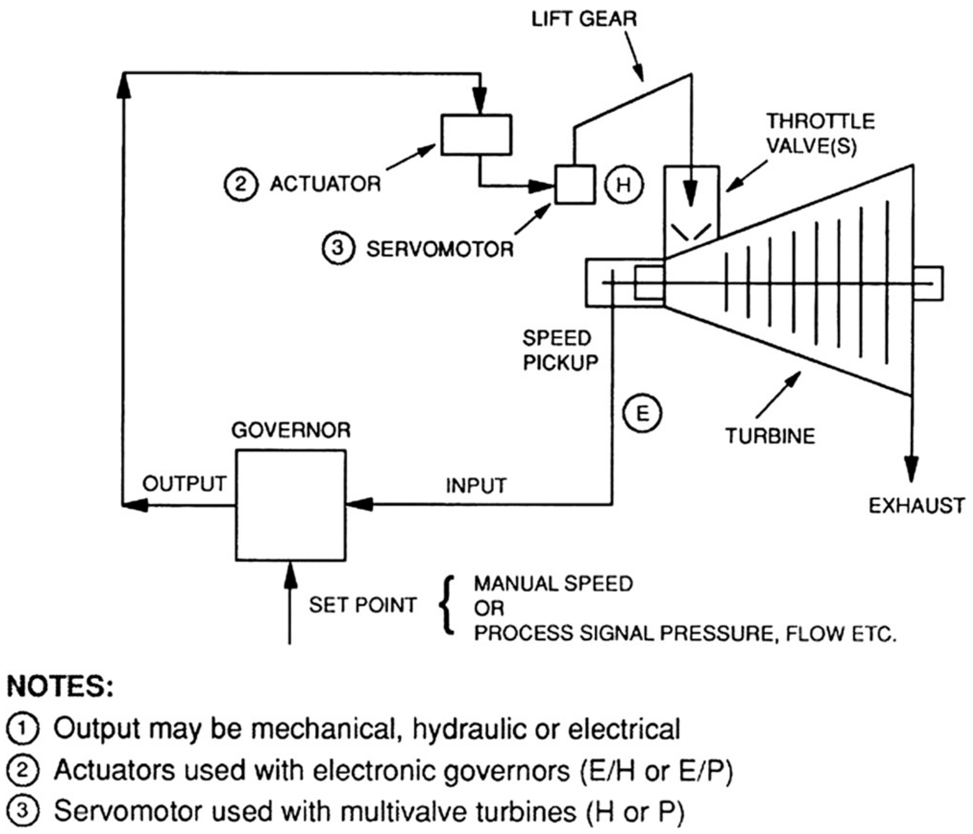
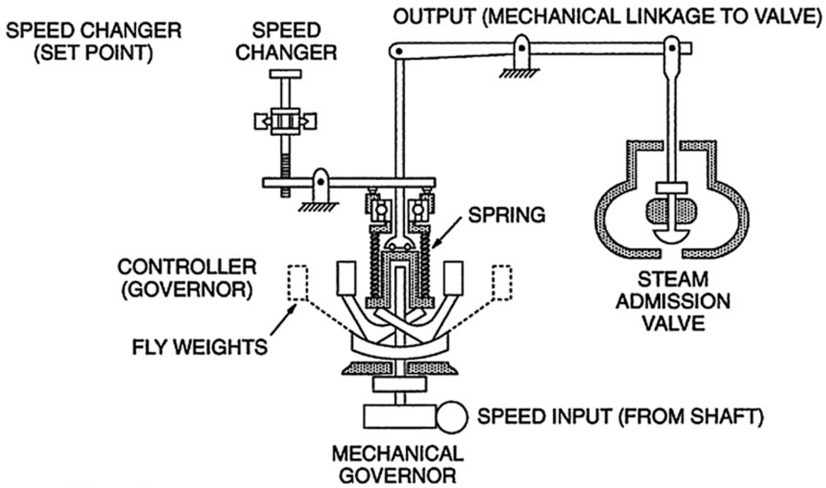


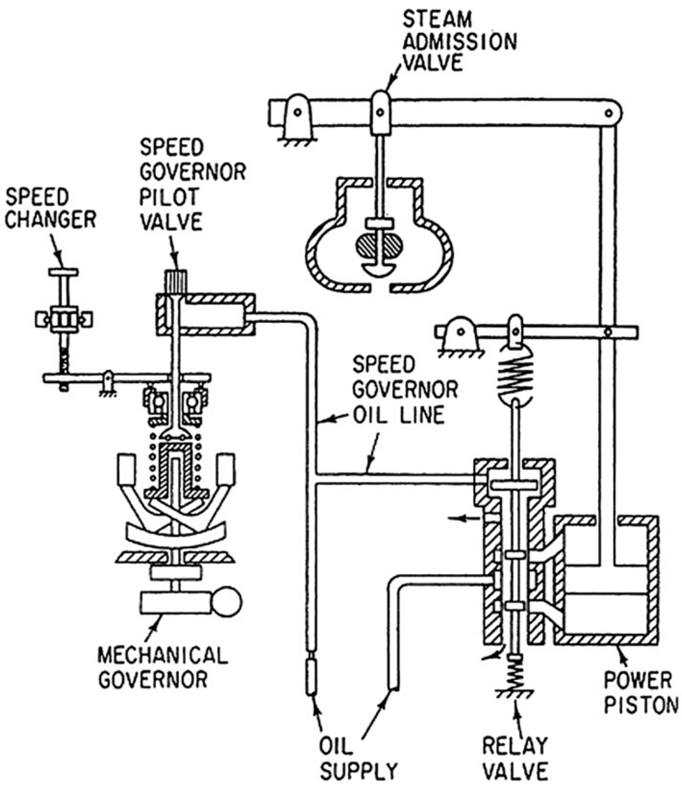
Fig 5.11.5 • Steam turbine control (Courtesy of M.E. Crane, Consultant)



Component Function:

- **Input** – *Continuously Advises Value of Turbine Speed*
- **Set Point** – *Defines the Desired Value of Speed or Process Variable*
- **Controller** – *Produces an Output Force (Signal) Proportional to Weight Mass, Radius of Rotation and Rotation SPD^2*
- **OutPut System** – *Accepts Controller Output Signal and Provides Sufficient Force to Modulate Steam Admission Valve*

Fig 5.11.6 • A mechanical governor system



FUNCTION: OUTPUT OF CONTROLLER MODULATES RELAY VALVE WHICH PROVIDES HYDRAULIC SIGNAL TO MOVE OUTPUT LINKAGE. NOTE: HAS ITS OWN INTERNAL PUMP

Fig 5.11.7 • A mechanical hydraulic governor system

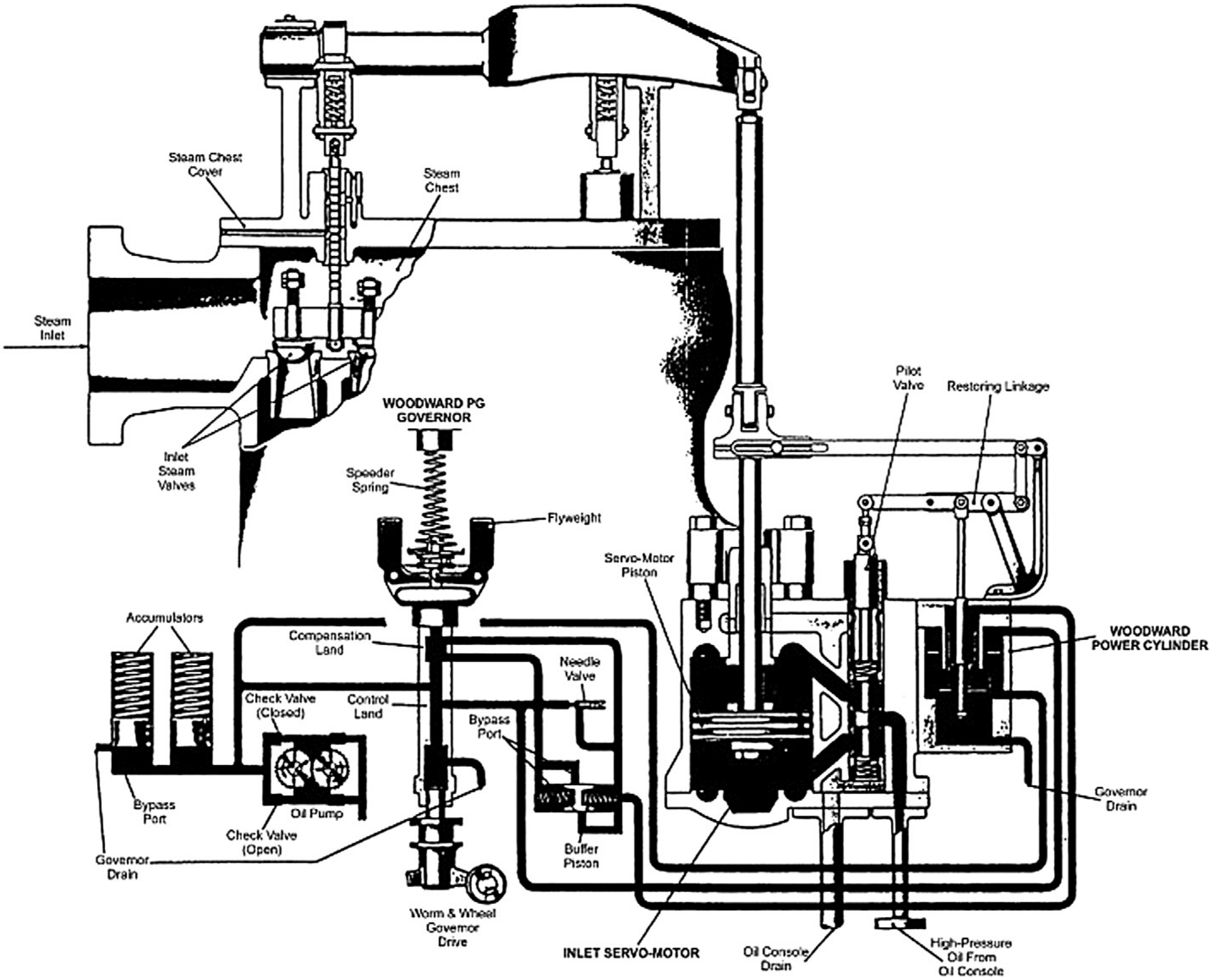


Fig 5.11.8 • Typical mechanical-hydraulic governor for turbine drive (Courtesy of Elliott/Woodward)

There are many controller designs. Historically, the first controllers were entirely mechanical. An example of a mechanical speed controller is shown in Figure 5.11.6. Commonly called 'fly ball governors', they worked by having the **input** shaft from the driver rotating the weights through a gear set. As the weights rotated, centrifugal force would move the weights outward, compressing the spring and thus moving the **output** linkage. The tension on the spring from the speed changer **(set point)** would control the speed as the equilibrium point of the input and set point values.

Many mechanical governors are still in use today on older, small single valve steam turbines. Since the output force from such systems is limited, the mechanical-hydraulic governor, pictured in Figure 5.11.7, was developed.

The mechanical-hydraulic governor uses the same mechanical mechanism to determine the output signal. However, the output shaft moves a pilot valve, which allows hydraulic fluid (usually oil) to provide the output signal to the throttle valve(s). The common Woodward 'TG' and 'PG' governors are examples of mechanical/hydraulic governors. These governors have internal positive displacement oil pumps driven by the governor input shaft.

All mechanical-hydraulic governors require hydraulic fluid, and site preventive maintenance practices must include these governors. They are provided with a sight glass to indicate the operating level of the hydraulic fluid. Typical fluids used are turbine oil and automatic transmission fluid 'ATF'. Governor instruction books must be consulted for specific hydraulic specifications. In larger systems, the governor hydraulic fluid reservoir may not be large enough to provide sufficient fluid to fill all of the speed governor oil lines. **Readers are cautioned that additional hydraulic fluid may have to be added during initial start-up and whenever work has been done on the governor system during a turnaround**.

- Do not require a mechanical input signal
- Provide extremely accurate control
- Provide self diagnostics, fault tolerance and auto-start capability
- Require actuator to convert electric output signal to control signal (hydraulic or pneumatic)
- Types:
  - Analog
  - Digital
- Either type can be:
  - Non-redundant
  - Redundant
  - Triple redundant

Fig 5.11.9 • Electro-hydraulic governors

Figure 5.11.8 is a representation of a mechanical-hydraulic governor system for a multi-valve steam turbine.

The system shows a Woodward 'P.G. - P.L.' governor system. These systems, common in the 1960s and 1970s, are still in use today, and have provided extremely reliable service. However, both mechanical and mechanical-hydraulic governors receive their input signal via a gear arrangement, so they cannot be repaired or removed while the turbine is operating. During the 1970s, refinery, petrochemical and gas plant capacities increased significantly. As a result, the lost product revenue for one day downtime for governor repair became very large (typically from $500,000 to over $1,000,000 U.S. dollars!). Therefore, there was an urgent need for a governor system that could be maintained without having to shut down the turbine. The electro/hydraulic governor met this need. Figure 5.11.9 presents the important facts concerning this system.

Since they did not require a mechanical (gear or shaft drive) input signal, these governors could be exchanged while the operators kept the turbine in the manual mode. As an analogy, exchanging automatic control valves is the same procedure. In this case, the operator maintains process conditions by manually throttling the bypass valve while the automatic control valve undergoes repair.

The first electronic governors were analog, and needed significant maintenance to change out cards. Digital governors were introduced in the late 1970s, and are the only type of speed control used today. As micro-processors became popular, digital governors also offered the great advantage of redundancy. Redundant and triple redundant governors became very popular, because they could now automatically transfer on-line to allow control to be maintained while the other governor required maintenance. Operator assistance was no longer required. Figure 5.11.10 presents a block diagram for an electro-hydraulic governor system.

Function: satisfy driven equipment control requirement and provide required extraction steam quantity at desired flow or pressure.
An extraction control system consists of multiple governors with feed back.

Fig 5.11.11 • Extraction control

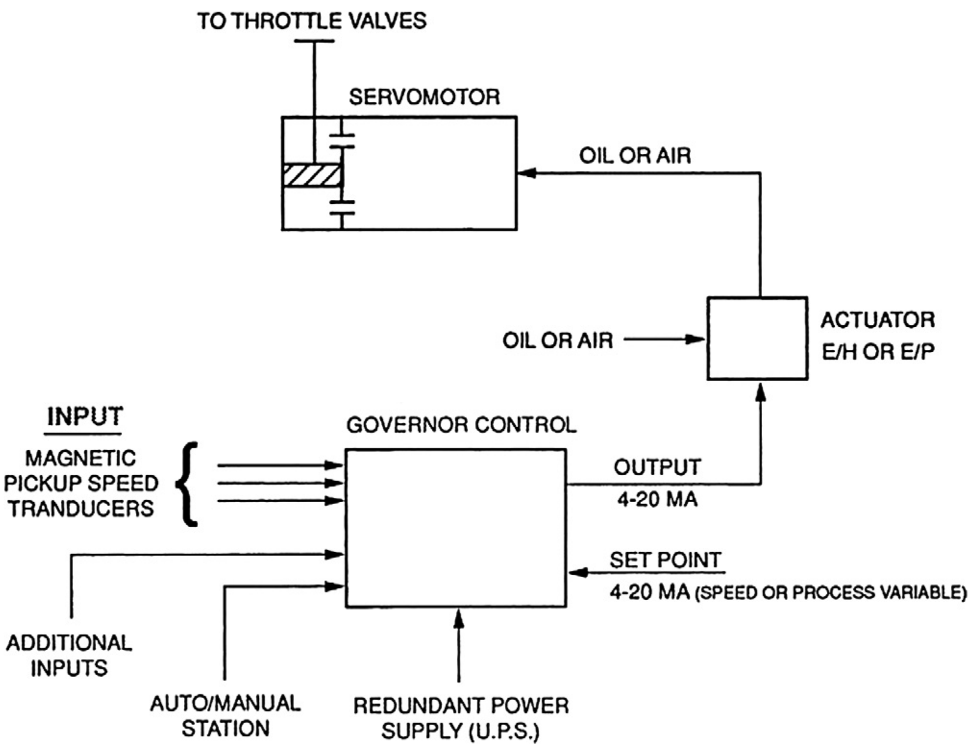


Fig 5.11.10 • Electrohydraulic governor block diagram (Courtesy of M.E. Crane, Consultant)

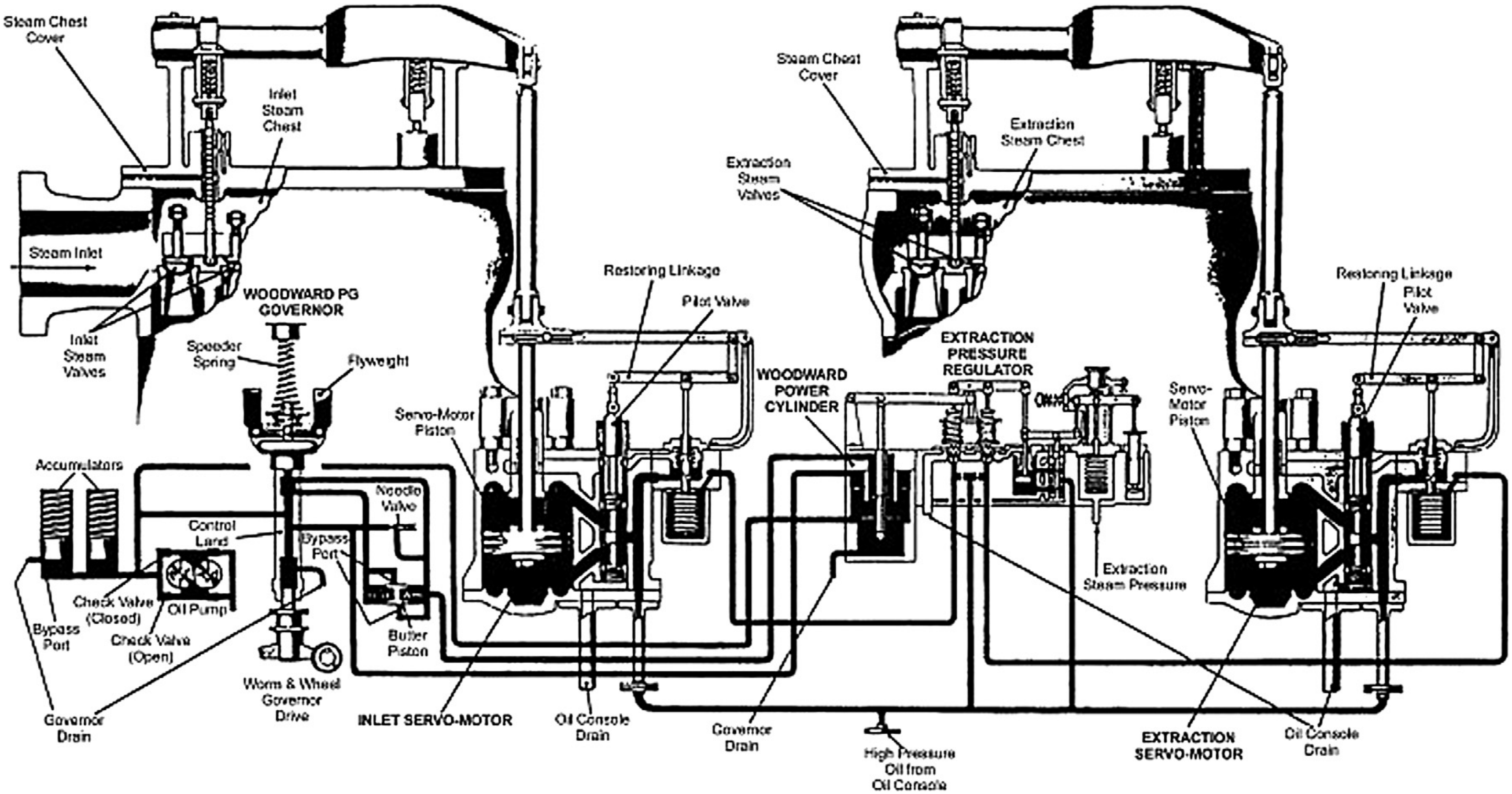


Fig 5.11.12 • Mechanical/hydraulic extraction control (Courtesy of Elliott/Woodward)

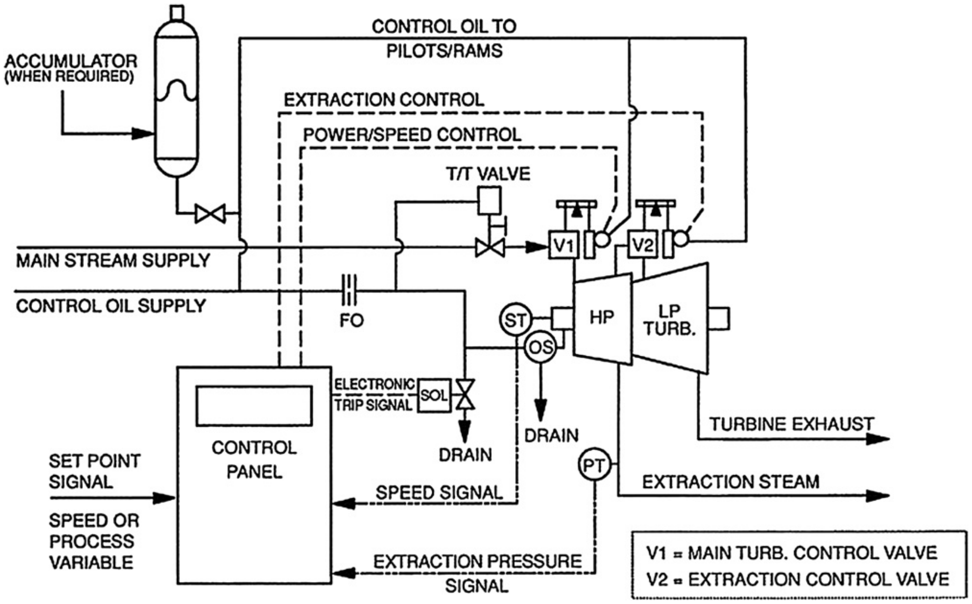


Fig 5.11.13 • Electro-hydraulic extraction control and protection system (Courtesy of M.E. Crane, Consultant)

In the 1990s, the trend was to control all process and machinery functions through the plant central distributed control system. A new chemical plant in South America is presently designing a DCS system that will control all critical system functions:

- Turbine speed control
- Process control
- Surge protection
- ESD systems
- On-line monitoring
- Emergency pump auto-start

In this design, all critical functions are actuated on the basis of a two-out-of-three voting system.

As previously discussed, extraction turbines are used to optimize plant steam balance and overall steam cycle efficiency. Figure 5.11.11 defines the function of an extraction steam turbine control system.

Both mechanical-hydraulic and electro-hydraulic extraction control systems are successfully operating in the field. Either design incorporates two or more governors operating together to meet the control system objectives. Each governor's output controls a specific set of throttle valves. In addition, each governor in an extraction or admission system continuously receives an input signal from the other governors in the system. Each governor will respond to this input signal as required to meet all of the control objectives of the governor system.

Mechanical-hydraulic extraction or admission systems need a significant amount of adjustment and maintenance, due to the high amount of friction in the system. Please refer to Figure 5.11.12, which shows a mechanical-hydraulic single extraction governor system. As a result, all new systems

Function: Continuously provide cool, clean control oil to control and protection system at proper pressure, flow rate and temperature.
Frequent problem areas:
- Main to auxiliary pump transfer
- Control oil valve instability
- Instantaneous flow requirement changes (need for accumulator)

Fig 5.11.14 • Control oil system

Application-driven equipment	Speed regulation %	Type of governor system
Spared pump	NEMA A ± 10%	Mechanical (older applications) Mechanical (hydraulic) Electro-hydraulic (optional)* Non-redundant
Fan(s)	NEMA A ± 10%	Mechanical (older hydraulics) Mechanical hydraulic
Lube/seal oil pump(s)	NEMA A ± 10%	Mechanical/hydraulic
Turbo-compressor	NEMA D ± 0.5%	Electro-hydraulic (post 1980) Non-redundant Optional-redundant, triple redundant
Generator	NEMA D ± 0.5%	Isochronous (0% droop) Mechanical/hydraulic Present Electro-hydraulic

Fig 5.11.15 • Steam turbine governor system application chart

The protection system monitors steam turbine total train parameters and ensures safety and reliability by the following action:

- Start-up (optional) provides a safe, reliable fully automatic start-up and will shut down the turbine on any abnormality
- Manual shutdown
- Trip valve exerciser allows trip valve stem movement to be confirmed during operation without shutdown
- Rotor overspeed monitors turbine rotor speed and will shut down turbine when maximum allowable speed (trip speed) is attained
- Excessive process variable signal monitors all train process variables and will shut down turbine when maximum value is exceeded

Fig 5.11.16 • Protection

incorporate electro-hydraulic governor arrangements as shown in Figure 5.11.13.

Coupled with redundant features, these systems offer high reliability and efficient process control. Regardless of the type of governor utilized, mechanical-hydraulic and electro-hydraulic governors must be supplied with a reliable control oil system. Figure 5.11.14 presents the function and frequent problem areas of hydraulic control systems.

Usually, the hydraulic control system is integral with the lubrication system. Typical pressure operating ranges for these systems are:

Low pressure: 276–690 kPa (40–100 PSI)
Medium pressure: 827–4,137 kPa (120–600 PSI)
High pressure Above: 4,137 kPa (600 PSI)

Figure 5.11.15 is an application chart showing type of governor classification, speed regulation and type of governor used.

In general, NEMA A governors are used in general purpose (spared) applications and NEMA D governors are used in special purpose (un-spared) applications.

Protection

The function of the steam turbine protection system is often confused with the control system, but in fact the two systems are entirely separate. The protection system operates only when any of the control system set point parameters are exceeded, and the steam turbine will be damaged if it continues to operate. Figure 5.11.16 defines the typical protection methods.

A schematic of a multi-valve, multi-stage turbine protection system is shown in Figure 5.11.17. This system incorporates a mechanical overspeed device (trip pin) to shut down the turbine on overspeed (10% above maximum

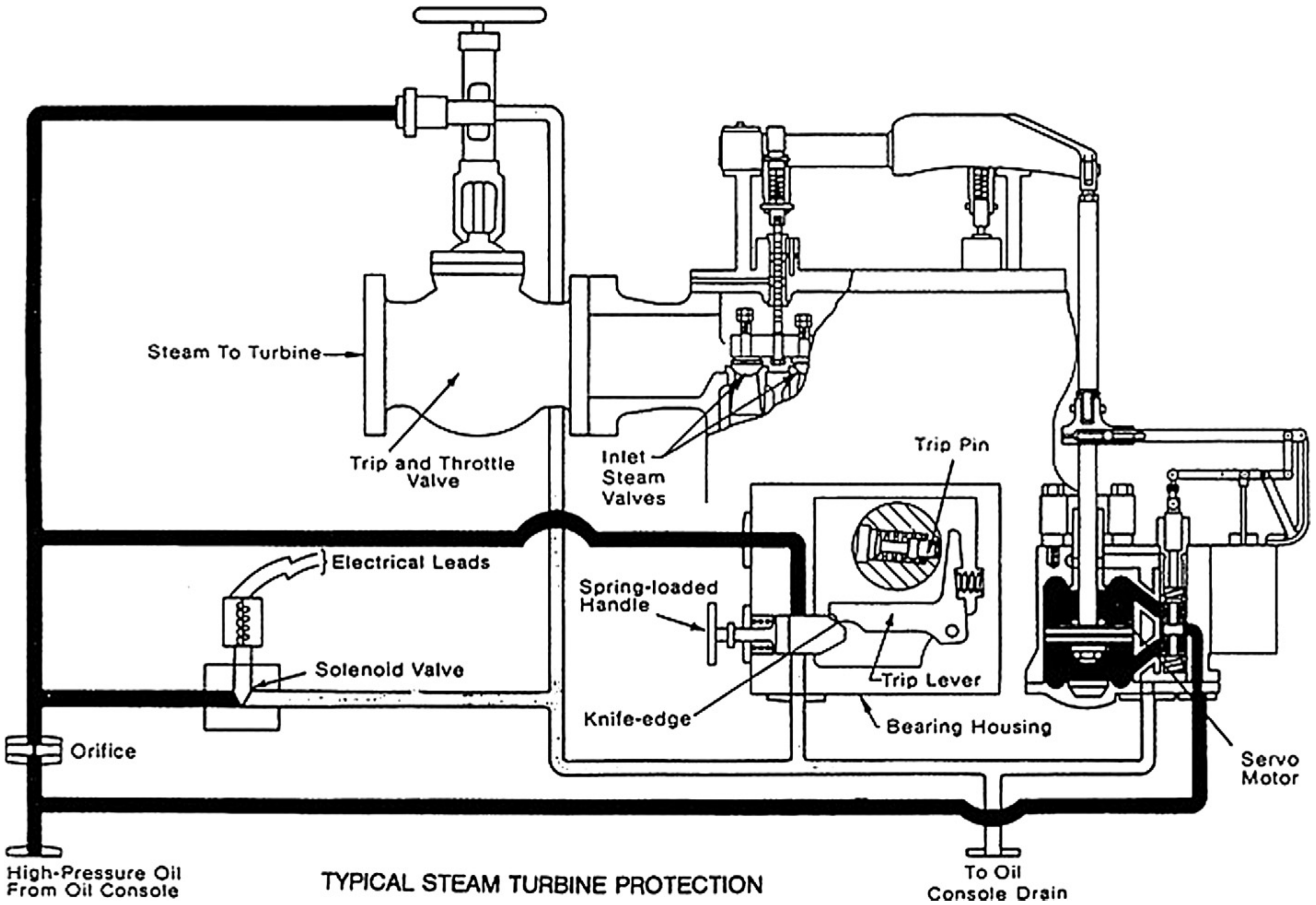


Fig 5.11.17 • Typical steam turbine protection (Courtesy of Elliott Co.)

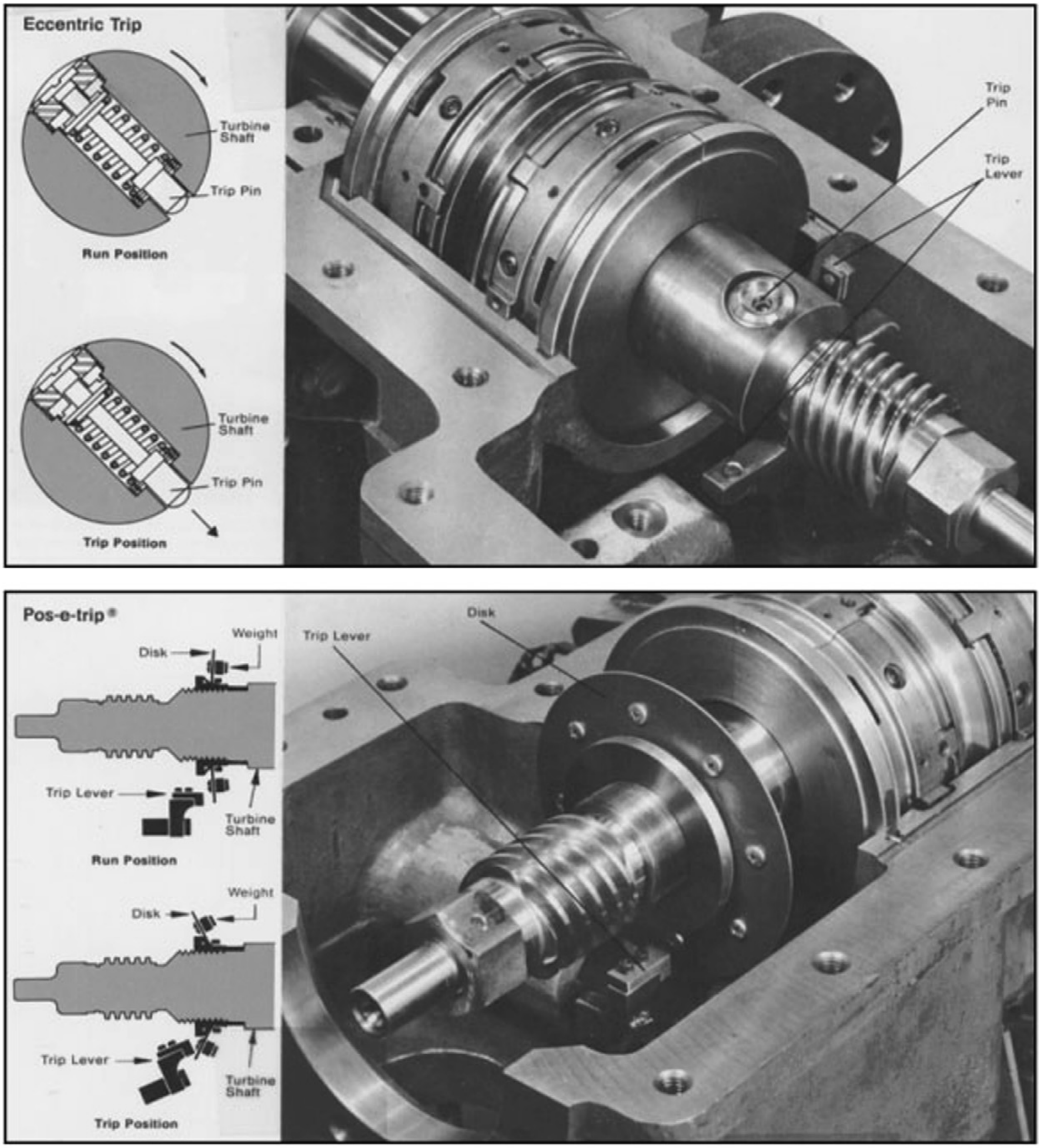
CHOICES:

MECHANICAL

- OVERSPEED BOLT

- DISC/ SPRING

ELECTRICAL



- MAGNET PICKUP FROM MULTI-TOOTH GEAR ON SHAFT

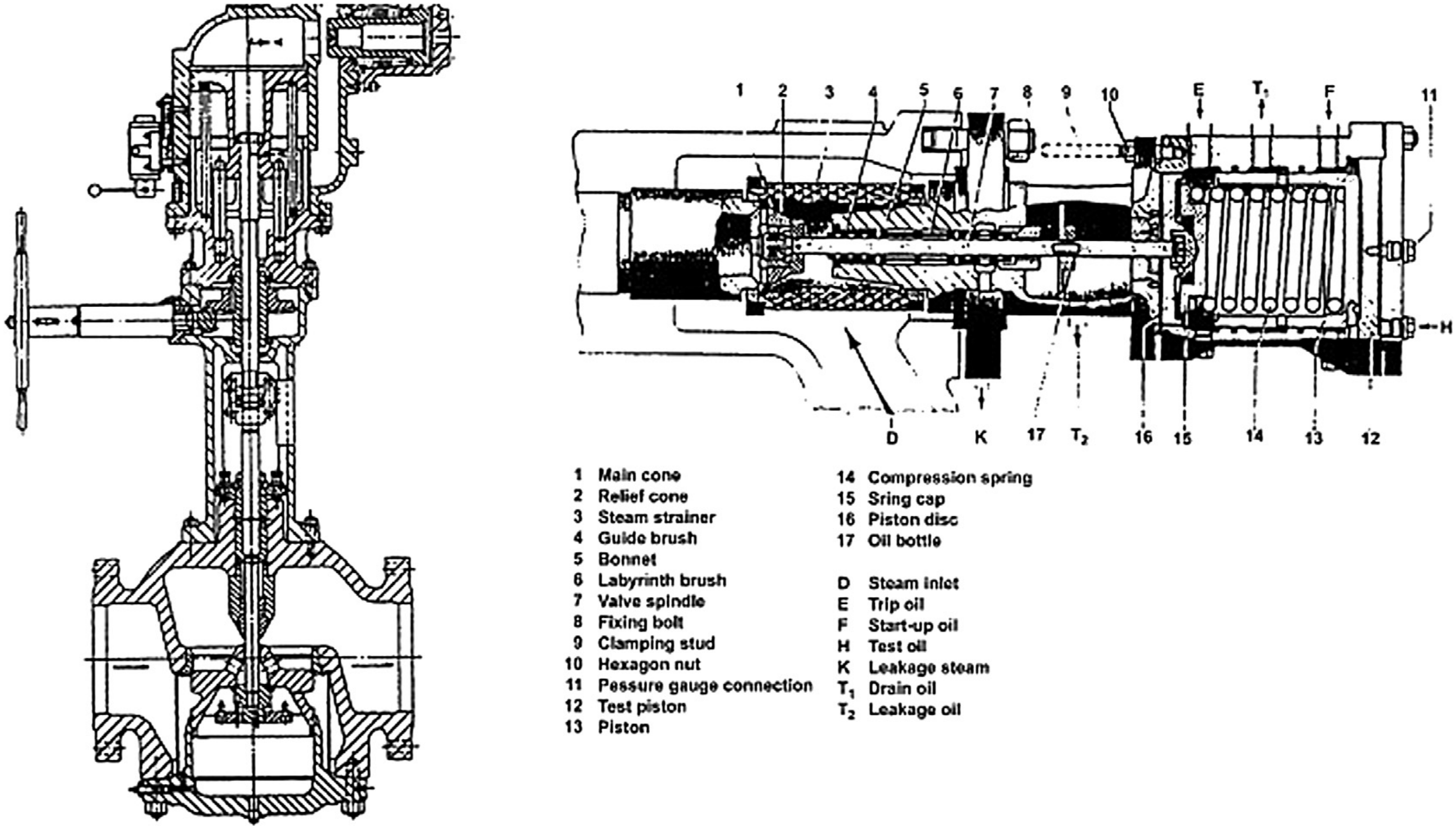
Fig 5.11.18 • Overspeed detection (Courtesy of Elliott Co.)

- Loss of control oil pressure
  Spring force automatically overcomes oil force holding valve open (approximate set point 50–65% of normal control oil pressure)
- Manual trip (panic button)
  Manually dumps control oil on command
- Optional
  Turbine excessive axial movement

Fig 5.11.19 • Internal protection

continuous speed). Centrifugal force resulting from high shaft speed will force the trip lever, which will allow the spring-loaded handle to move inward. When this occurs, the port in the handle stem will allow the control oil pressure to drain and drop to zero. The high energy spring in the trip and throttle valve, normally opposed by the control oil pressure, will close suddenly (less than one second). In this system there are two other means of tripping the turbine (reducing control oil pressure to zero):

- Manually pushing spring loaded handle
- Solenoid valve opening



- FUNCTION RAPIDLY TO CUT OFF STEAM TO TURBINE ON OVERSPEED OR ANY DESIGNATED UPSET. STRONG SPRING FORCE RAPIDLY (ONE (1) SECOND) CLOSES VALVE.

Fig 5.11.20 • Steam turbine shut-off valves. Left: Trip and throttle. (Courtesy of Gimple Corp.) Right: Trip (Courtesy of Siemens)

The solenoid valve will open on command when any trip parameter set point is exceeded. Solenoid valves are designed to be normally energized to close.

In recent years the industry has required parallel and series arrangements of solenoid valves to ensure increased steam turbine train reliability. Figure 5.11.18 shows two popular methods of overspeed protection used in the past.

Today, most speed trip systems incorporate magnetic speed input signals and two-out-of-three voting for increased reliability. Figure 5.11.19 presents the devices that trip the turbine internally. That is, they directly reduce the control oil pressure causing a trip valve closure without the need of a solenoid valve (external trip method).

Two popular types of steam turbine shutoff valves are shown in Figure 5.11.20. Both types use a high spring force, opposed by control oil pressure during normal operation, to close the valve rapidly on loss of control oil pressure.

It is very important to note that the trip valve will only close if the spring has sufficient force to overcome valve stem friction. Steam system solid build-up, which increases with system pressure (when steam systems are not properly maintained) can prevent the trip valve from closing.

To ensure the trip valve stem is free to move, all trip valves should be manually exercised on-line. The recommended frequency is once per month.

All turbine trip valves should be provided with manual exercisers to allow this feature. Figure 5.11.21 presents facts concerning manually exercising a turbine while on-line.

- Trip valve is only as reliable as valve to move
- Should periodically (minimum one per month) exercise valve to ensure movement
- **Exercises will not trip turbine**
- If valve does not move, must be remedied immediately

Fig 5.11.21 • On-line manual exercise of trip valve

Protection system philosophies have tended to vary geographically with steam turbine vendors. Figure 5.11.22 presents these facts.

- Most domestic vendors rely only on trip valve to shut off steam supply (throttle valves remain open)
- European vendors close both trip and automatic throttle valve on trip signal

Fig 5.11.22 • Protection system philosophies

Best Practice 5.12

Exercise very high pressure turbine trip valves (inlet steam pressures exceeding 100 barg) daily to positively prevent their failure to close upon command.

It can be hard to maintain VHP (very high pressure) steam systems, and to prevent contaminants (calcium, silica, etc.) from forming inside the turbine.

Trip valve packing is essentially a filter that will trap any contaminants between the trip valve and the packing which can prevent the trip valve form closing.

Failure of the trip valve to close on command can cause catastrophic machine failure and expose personnel to safety issues.

Periodic or infrequent exercise of trip valves can result in failure of the valve to move which, considering plant safety requirements, will necessitate immediate turbine shutdown.

Daily exercise of VHP trip valves will ensure freedom of movement of the trip valve and positively prevent unnecessary unit shutdowns.

Lessons Learned

Failure to exercise VHP trip valves daily or HP trip valves monthly has resulted in catastrophic machinery failure, personnel lost time and loss of life.

Benchmarks

This best practice has been recommended since the 1990s. When followed, it has resulted in zero lost time accidents and failure-to-trip incidents. When not followed, more than one catastrophic machine outage in critical (un-spared) machinery has occurred, that has exceeded three months in repair time.

B.P. 5.12. Supporting Material

Protection

The function of the steam turbine protection system is often confused with the control system, but in fact the two systems are entirely separate. The protection system operates only when any of the control system set point parameters are exceeded, and the steam turbine will be damaged if it continues to operate. Figure 5.12.1 defines the typical protection methods.

A schematic of a multi-valve, multi-stage turbine protection system is shown in Figure 5.12.2. This system incorporates a mechanical overspeed device (trip pin) to shut down the turbine on overspeed (10% above maximum continuous speed).

The protection system monitors steam turbine total train parameters and ensures safety and reliability by the following action:

- Start-up (optional) provides a safe, reliable fully automatic start-up and will shut down the turbine on any abnormality
- Manual shutdown
- Trip valve exerciser allows trip valve stem movement to be confirmed during operation without shutdown
- Rotor overspeed monitors turbine rotor speed and will shut down turbine when maximum allowable speed (trip speed) is attained
- Excessive process variable signal monitors all train process variables and will shut down turbine when maximum value is exceeded

Fig 5.12.1 • Protection

Centrifugal force resulting from high shaft speed will force the trip lever, which will allow the spring loaded handle to move inward. When this occurs, the port in the handle stem will allow the control oil pressure to drain and drop to zero. The high energy spring in the trip and throttle valve, normally opposed by the control oil pressure will close suddenly (less than one second). In this system there are two other means of tripping the turbine (reducing control oil pressure to zero):

- Manually pushing spring loaded handle
- Solenoid valve opening

The solenoid valve will open on command when any trip parameter set point is exceeded. Solenoid valves are designed to be normally energized to close.

In recent years the industry has required parallel and series arrangements of solenoid valves to ensure increased steam turbine train reliability. Figure 5.12.3 shows two popular methods of overspeed protection used in the past.

Today, most speed trip systems incorporate magnetic speed input signals and two-out-of-three voting for increased reliability. Figure 5.12.4 presents the devices that trip the turbine internally. That is, they directly reduce the control oil pressure causing a trip valve closure without the need of a solenoid valve (external trip method).

Two popular types of steam turbine shutoff valves are shown in Figure 5.12.5. Both types use a high spring force, opposed by control oil pressure during normal operation, to close the valve rapidly on loss of control oil pressure.

It is very important to note that the trip valve will only close if the spring has sufficient force to overcome valve stem friction. Steam system solid build up, which increases with system pressure (when steam systems are not properly maintained) can prevent the trip valve from closing.

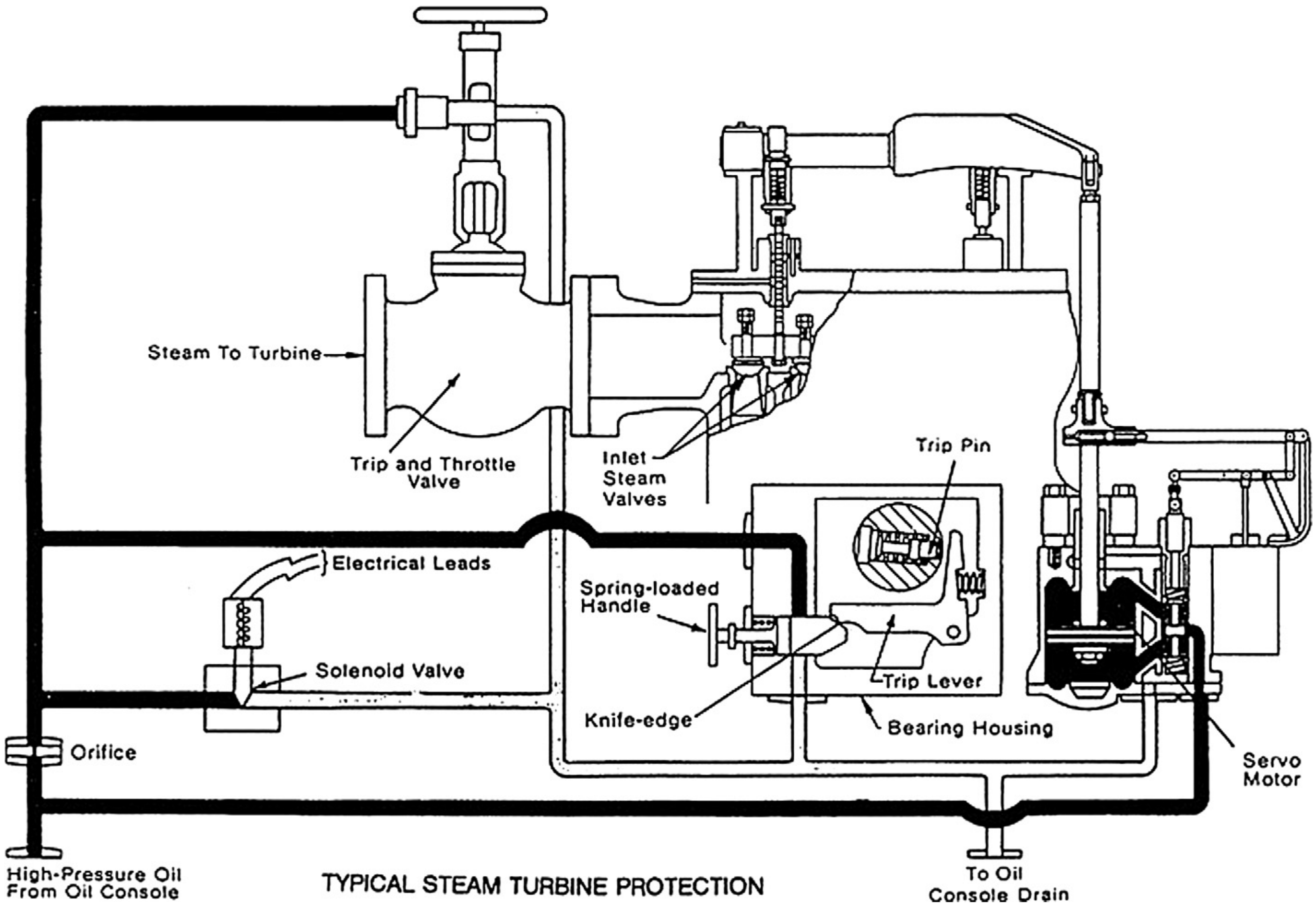


Fig 5.12.2 • Typical steam turbine protection (Courtesy of Elliott Co.)

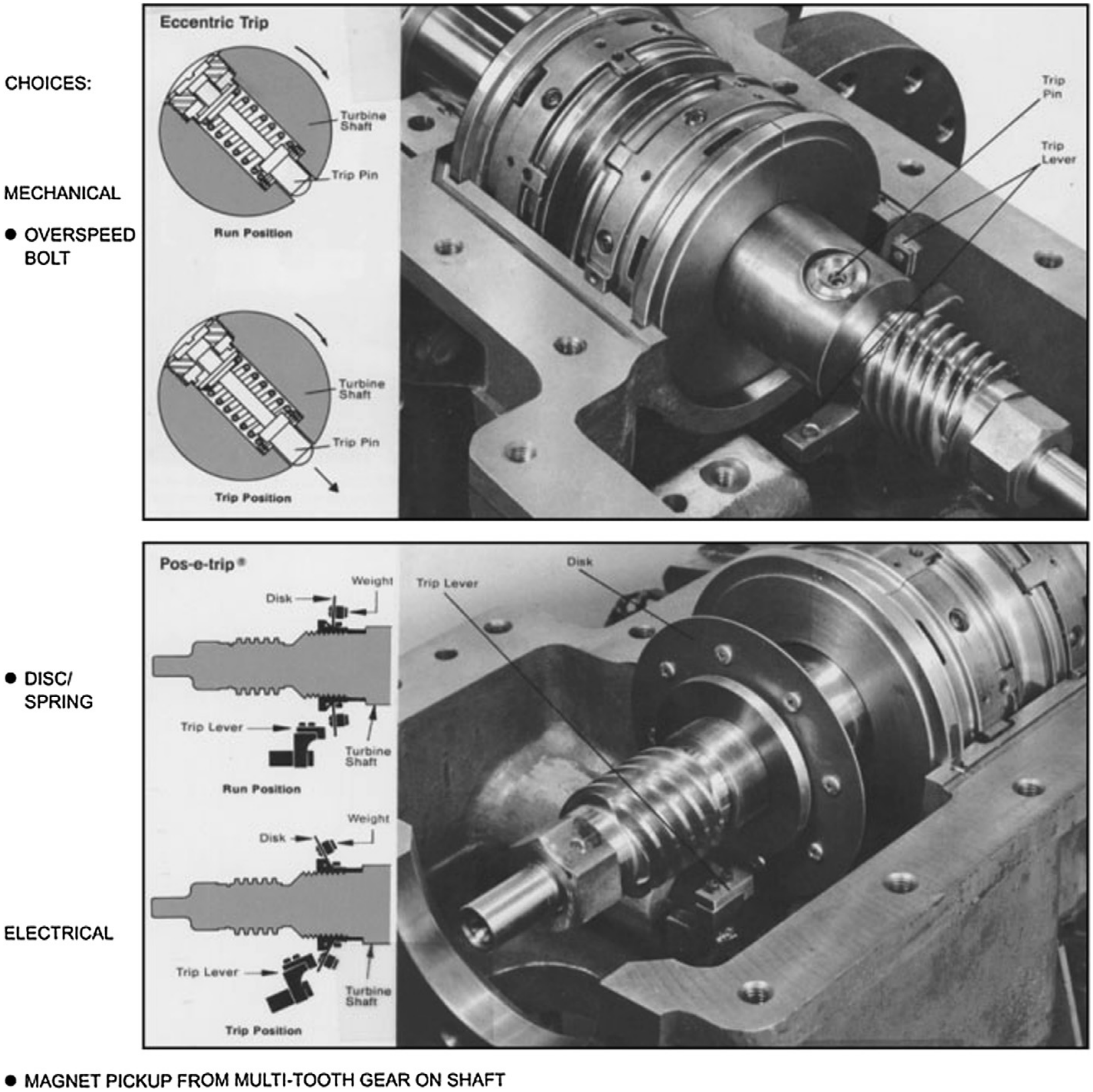


Fig 5.12.3 • Overspeed detection (Courtesy of Elliott Co.)

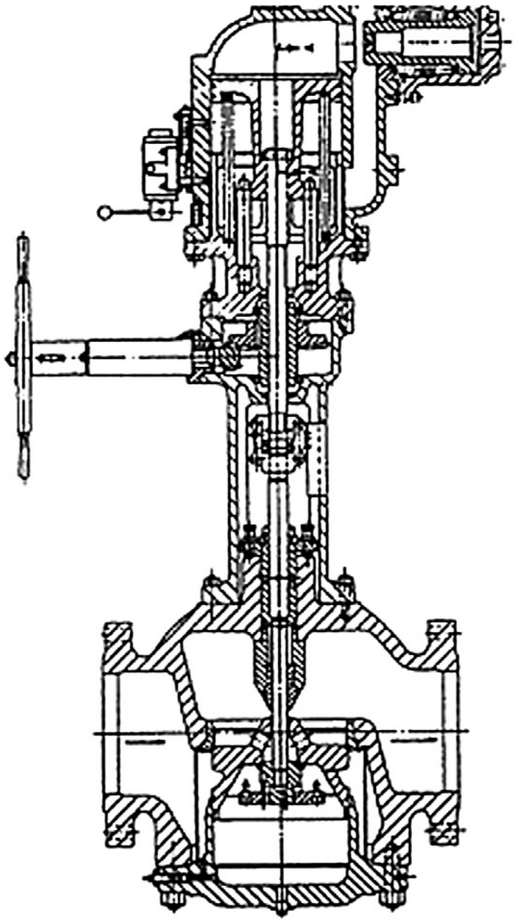
- Loss of control oil pressure
  Spring force automatically overcomes oil force holding valve open (approximate set point 50–65% of normal control oil pressure)
- Manual trip (panic button)
  Manually dumps control oil on command
- Optional
  Turbine excessive axial movement

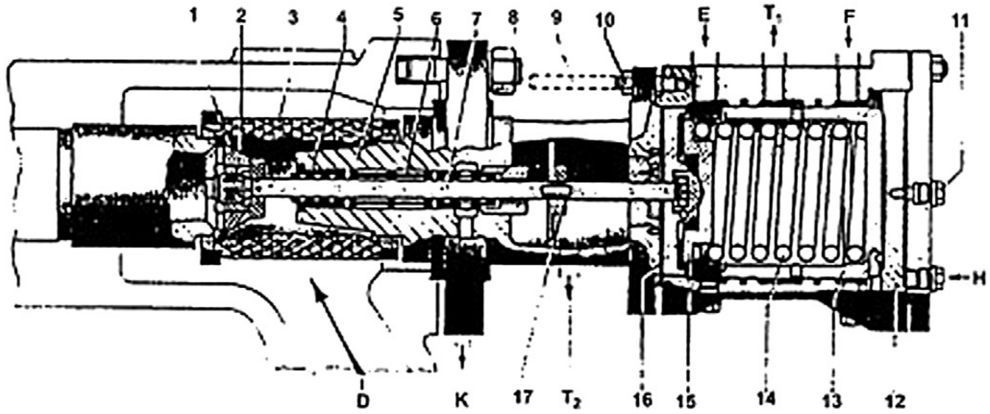
Fig 5.12.4 • Internal protection

To ensure the trip valve stem is free to move, all trip valves should be manually exercised on-line. The recommended frequency is once per month for High Pressure (40 bar) steam systems and daily for very high pressure (1000 bar +) steam systems.

All turbine trip valves should be provided with manual exercisers to allow this feature. Figure 5.12.6 presents facts concerning manually exercising a turbine while on line.

Protection system philosophies have tended to vary geographically with steam turbine vendors. Figure 5.12.7 presents these facts.





- FUNCTION RAPIDLY TO CUT OFF STEAM TO TURBINE ON OVERSPEED OR ANY DESIGNATED UPSET. STRONG SPRING FORCE RAPIDLY (ONE (1) SECOND) CLOSES VALVE.

Fig 5.12.5 • Steam turbine shut-off valves. Left: Trip and throttle (Courtesy of Gimple Corp.). Right: Trip (Courtesy of Siemens)

- Trip valve is only as reliable as valve to move
- Should periodically exercise valve to ensure movement (minimum one per month for high pressure (40 bar) steam systems and daily for very high pressure (1000 bar +) steam systems)
- **Exercises will not trip turbine**
- If valve does not move, must be remedied immediately

Fig 5.12.6 • On-line manual exercise of trip valve

- Most domestic vendors rely only on trip valve to shut off steam supply (throttle valves remain open)
- European vendors close both trip and automatic throttle valve on trip signal

Fig 5.12.7 • Protection system philosophies

Best Practice 5.13

Accurately trend mechanical drive special purpose turbine efficiency, speed and flow rate using an installed torquemeter for condensing and extraction condensing steam turbines.

Condensing and extraction steam turbine efficiency determination requires the calculation of the driven equipment power. This calculation can be inaccurate due to the many variables involved.

Installing a torquemeter during the project phase will ensure accurate efficiency calculations for all condensing and extraction condensing steam turbines.

Torquemeters can be installed in existing trains but will require:

- Rotor response study
- Torsional study
- Coupling and coupling guard modifications

Lessons Learned

Condensing and extraction condensing steam turbine efficiency calculations will be erroneous and non-conclusive in determining maintenance requirements when torquemeters are not installed.

Benchmarks

The installation of torquemeters has been recommended as part of project specifications since 1995. This has resulted in the accurate determination of critical (un-spared) steam turbine maintenance to be carried out during turnarounds only and during planned operation.

B.P. 5.13. Supporting Material

Steam turbine performance characteristics

In this section we will discuss how energy is extracted from the vapor (steam) in a steam turbine. Power output is determined by the following factors in a steam turbine:

- The energy available from the vapor
- The external efficiency of the turbine
- The steam flow rate

Note: The external efficiency is equal to the internal efficiency (steam path efficiency) of the turbine factored by the mechanical losses and steam leakage losses from the turbine.

The energy available from the vapor is determined by the steam conditions of the particular application. That is, the pressures and temperatures at the turbine inlet and exhaust flanges. To determine the energy available from the vapor, steam tables or a Mollier diagram are used. We have included a Mollier diagram for steam in this section. We will review in practical terms the use of a Mollier diagram to determine:

- Theoretical steam rate
- Actual steam rate
- Turbine efficiency

An example will be presented for both non-condensing and condensing turbines. It should be noted that the exercises in this section will deal with overall steam turbine performance only. In the next section, we will discuss individual blade performance and efficiencies, and determine, as an exercise, the number of stages required for a given turbine application.

In this section we will also observe the effect of design steam conditions on steam turbine performance and reliability. Finally, typical turbine efficiencies and performance curves will be presented and discussed.

Steam conditions

Steam conditions determine the energy available per pound of steam. Figure 5.13.1 explains where they are measured and how they determine the energy produced.

Frequently, proper attention is not paid to maintaining the proper steam conditions at the flanges of a steam turbine. Failure

- The steam conditions are the pressure and temperature conditions at the turbine inlet and exhaust flanges
- They define the energy per unit weight of vapor that is converted from potential energy to kinetic energy (work)

Fig 5.13.1 • Steam conditions

to do this will affect power produced, and can cause mechanical damage to turbine internals resulting from blade erosion and/or corrosion. Figure 5.13.2 presents these facts.

Inlet steam conditions should be as close as possible (+/− 5%) to specified conditions because:

- Power output will decrease
- Exhaust end steam moisture content will increase, causing blade, nozzle and diaphragm erosion

Fig 5.13.2 • Steam condition limits

A Mollier diagram or steam tables allow determination of the energy available in a pound of steam for a specific pressure and temperature. Figure 5.13.3 describes the Mollier diagram and the parameters involved.

Describes the energy per unit mass of fluid when pressure and temperature are known.

- Enthalpy (energy/unit mass) is plotted on Y axis
- Entropy (energy/unit mass degree) is plotted on X axis
- Locating P_1, T_1 gives a value of enthalpy (H) horizontal and entropy (S) vertical
- Isentropic expansion occurs at constant entropy ($\Delta S = 0$) and represents an ideal (reversible) expansion

Fig 5.13.3 • The Mollier Diagram

Refer to Figure 5.13.4 – an enlarged Mollier Diagram. As an exercise, plot the following values on the Mollier Diagram in this section and determine the corresponding available energy in BTUs per pound.

1. $P_1 = 600$ PSIG, $T_1 = 800°F$ $\qquad h_1 = \dfrac{BTU}{LB_M}$
2. $P_2 = 150$ PSIG, $T_2 = 580°F$ $\qquad h_2 = \dfrac{BTU}{LB_M}$
3. $P_1 = 1500$ PSIG, $T_1 = 900°F$ $\qquad h_1 = \dfrac{BTU}{LB_M}$
4. $P_2 = 2$ PSIG, % moisture = 9% $\qquad h_2 = \dfrac{BTU}{LB_M}$

Having plotted various inlet and exhaust conditions on the Mollier diagram to become familiar with its use, please refer to Figure 5.13.5, which presents the definitions and uses of steam rate.

Theoretical steam rate

The theoretical steam rate is the amount of steam, in LBS per hour required to produce one (1) horsepower if the isentropic efficiency of the turbine is 100%. As shown in Figure 5.13.5, it is determined by dividing the theoretical enthalpy $\Delta h_{isentropic}$ into the amount of BTUs/HR in horsepower.

Actual steam rate

The actual steam rate is the amount of steam, in LBS per hour, required to produce one (1) horsepower based on the actual turbine efficiency. As shown in Figure 5.13.5, it is determined by dividing the theoretical steam rate (TSR) by the turbine efficiency. Alternately, if the turbine efficiency is not known and the turbine inlet and exhaust conditions are given (P_2, T_2 or % moisture), the actual steam rate can be obtained in the same manner as theoretical steam rate but substituting ΔH_{actual} for $\Delta H_{isentropic}$.

Turbine efficiency

As shown in Figure 5.13.5, turbine efficiency can be determined either by the ratio of TSR to ASR or Δh_{actual} to $\Delta H_{isentropic}$.

It is relatively easy to determine the efficiency of any operating turbine in the field if the exhaust conditions are superheated. All that is required are calibrated pressure and temperature gauges on the inlet and discharge, and a Mollier diagram or steam tables. The procedure is as follows:

1. For inlet conditions, determine h_1
2. For inlet condition with $\Delta S = 0$, determine h_{2ideal}
3. For outlet conditions, determine $h_{2actual}$
4. Determine $\Delta h_{ideal} = h_1 - h_{2ideal}$
5. Determine $\Delta h_{actual} = h_1 - h_{2actual}$
6. Determine efficiency

$$\text{Efficiency} = \frac{\Delta H_{actual}}{\Delta H_{ideal}}$$

However, for turbines with saturated exhaust conditions, the above procedure cannot be used because the actual exhaust condition cannot be easily determined. This is because the % moisture must be known. Instruments (calorimeters) are available, but results are not always accurate. Therefore the suggested procedure for turbines with saturated exhaust conditions is as follows:

1. Determine the power required by the driven equipment. This is equal to the power produced by the turbine.
2. Measure the following turbine parameters using calibrated gauges:
   - P_{in}
   - T_{in}
   - $P_{exhaust}$
   - Steam flow (in lbs/hr)
3. Determine the theoretical steam rate by plotting P_{in}, T_{in}, $P_{exhaust}$ @ $\Delta S = 0$, and dividing $\Delta h_{isentropic}$ into the constant.
4. Determine the actual steam rate of the turbine as follows:

$$\text{Actual Steam Rate (A.S.R.)} = \frac{\text{Steam Flow (lb/hr)}}{\text{BHP required by driven equipment}}$$

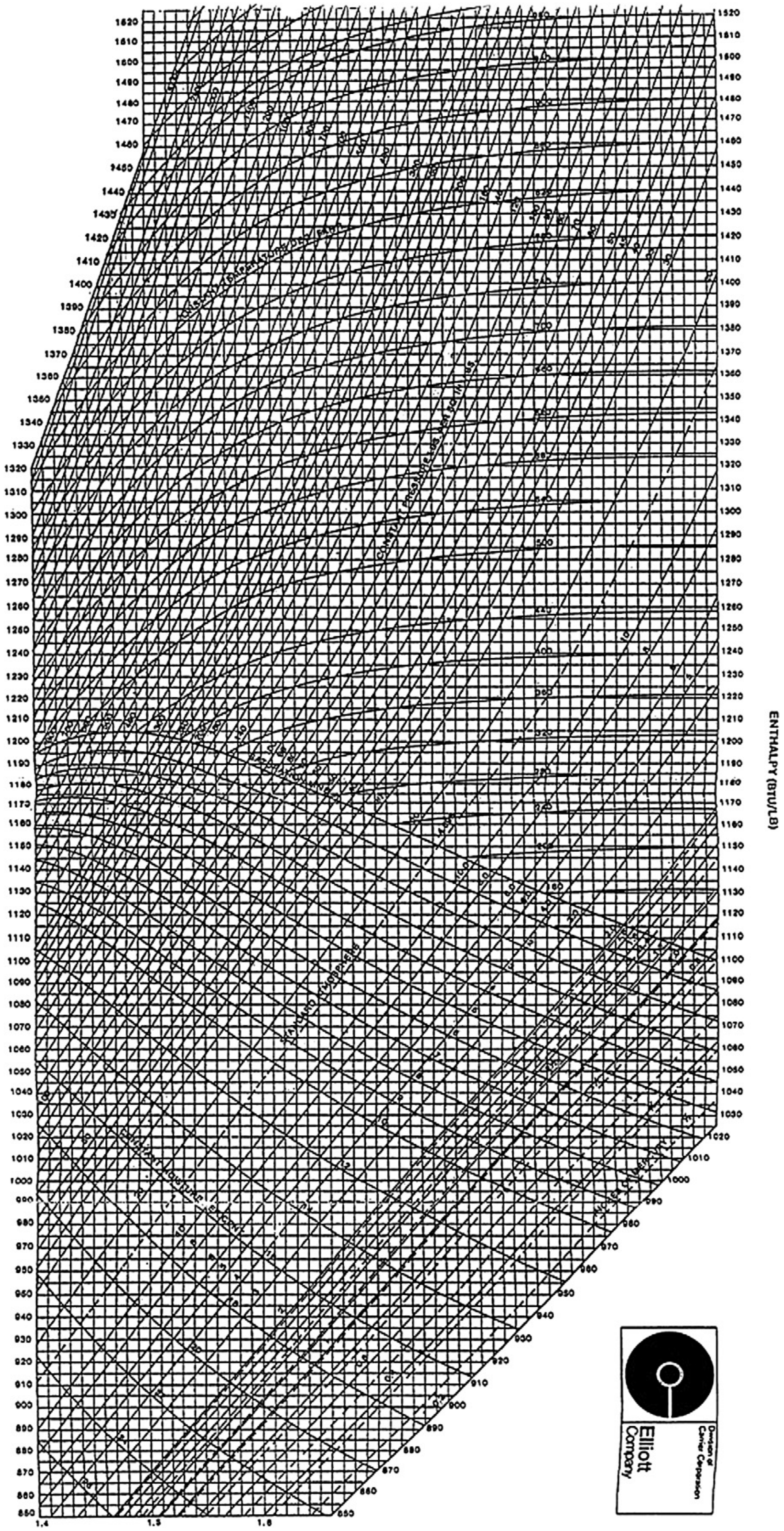


Fig 5.13.4 • Mollier steam diagram (Courtesy of Elliott Company)

Uses:

- Determine the amount of steam required per hour
- Determine the amount of potential kW (horsepower)

Required:

- Steam conditions
- Theoretical steam rate table or Mollier diagram
- Thermal efficiency of turbine

Formula:

Metric Units

- Theoretical steam rate

$$\text{TSR (kg/kWhr)} = \frac{3600\ \text{kJ/kW-hr}}{\Delta H_{\text{ISENTROPIC}}}$$

- Actual steam rate

$$\text{A.S.R. (kg/kWhr)} = \frac{\text{T.S.R.}}{\text{Efficiency}} = \frac{3600\ \text{kJ/kW-hr}}{\Delta H_{\text{ACTUAL}}}$$

- Turbine efficiency

$$\text{Efficiency} = \frac{\text{T.S.R.}}{\text{A.S.R}} = \frac{\Delta H_{\text{ACTUAL}}}{\Delta H_{\text{ISENTROPIC}}}$$

U.S. Units

$$\text{TSR (lb/HP-hr)} = \frac{2545\ \text{BTUS/HP-hr}}{\Delta H_{\text{ISENTROPIC}}}$$

$$\text{A.S.R. (lb/HP-hr)} = \frac{\text{T.S.R}}{\text{Efficiency}} = \frac{2545\ \text{BTU/HP-hr}}{\Delta H_{\text{ACTUAL}}\ \text{HP/hr}}$$

Fig 5.13.5 • Determining steam rate

5. Determine efficiency

$$\text{Efficiency} = \frac{\text{T.S.R.}}{\text{A.S.R.}}$$

Figures 5.13.6 to 5.13.8 present the advice and values concerning steam turbine efficiencies. The efficiencies presented can be used for estimating purposes.

- Quoted turbine efficiencies are external efficiencies; they include mechanical (bearing, etc.) and leakage losses
- Turbine efficiency at off load conditions will usually be lower than rated efficiency
- Typical efficiencies are presented for impulse turbine:
  - Condensing multi-stage
  - Non condensing multi-stage
  - Non condensing single state

Fig 5.13.6 • Typical steam turbine efficiencies

Why steam turbines are not performance tested

When purchasing large steam turbines that do not use proven components, keep in mind that it is not cost effective to performance test the turbine prior to field installation. If the turbine does not meet predicted output horsepower values, the field modifications will be lengthy and costly in terms of lost product revenue resulting from reduced output horsepower. In some cases, the output power predicted may never be attained. Figure 5.13.9 presents the reasons why steam turbines are not performance tested.

Performance curves

The performance curve format for steam turbines is to plot steam flow on the y axis and produced shaft horsepower on the x axis. Figure 5.13.10 presents important facts concerning steam turbine performance curves.

In Figure 5.13.11, a typical performance curve is presented for a single stage turbine with manual hand valves.

Note that this turbine contains three manual hand valves (x, y, and z). Closing hand valves for low horsepower loads increases the efficiency of the turbine. However, please note that closed hand valves limit the steam flow through a turbine, and therefore the horsepower produced. Hand valves are not modulating – that is, they are either fully open or fully closed. Throttling a hand valve will destroy the valve seat and may damage the valve stem, thus rendering it immovable. Normally hand valves are manually actuated, however, modern electronic governor systems provide outputs to open or close hand valves based on power requirements.

Figure 5.13.12 shows a performance curve for a typical extraction steam turbine. This performance curve plots inlet flow and extraction flow vs. turbine horsepower produced. When selecting an extraction turbine, care must be taken to be sure the turbine produces the horsepower required during the start-up of the process. The cost of an extraction steam turbine can be significantly reduced if the size of the exhaust section (LP steam section) is reduced. Figure 5.13.12 shows an extraction turbine capable of producing 100% power with 0% extraction flow. Usually, extraction turbines are sized to only provide the process start-up horsepower with 0% extraction. These values may be as low as 50-60% of full load horsepower.

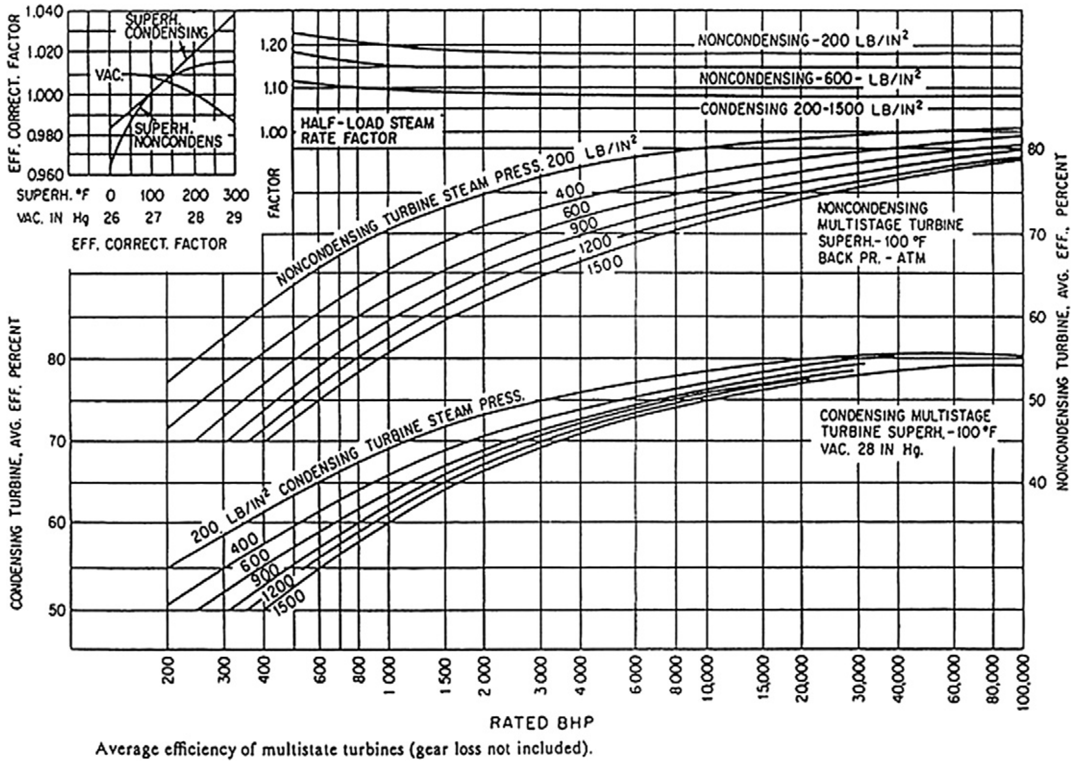


Fig 5.13.7 • Efficiency of multistate turbines (Courtesy of IMO Industries)

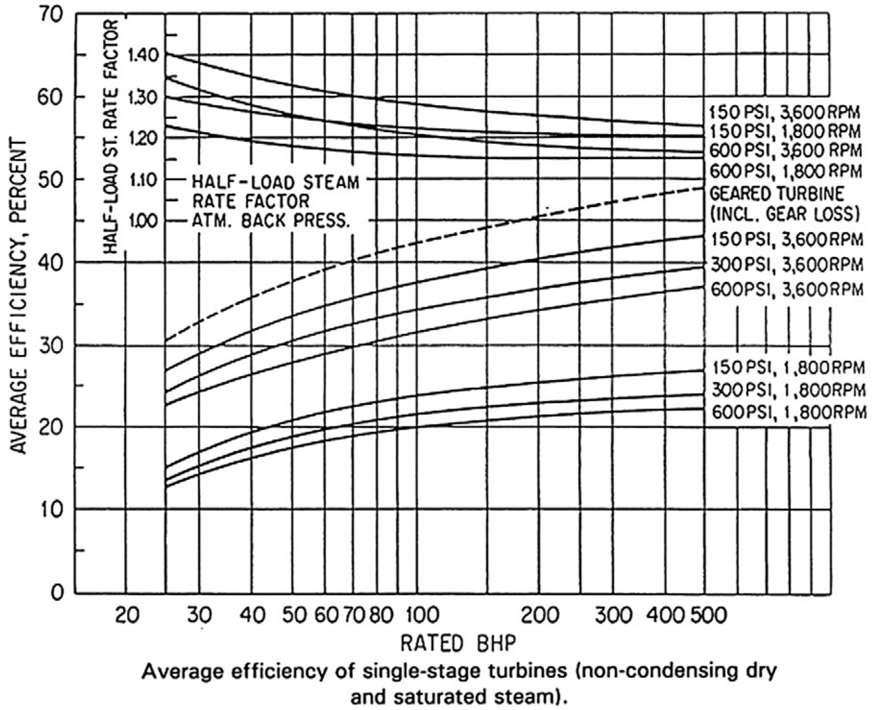


Fig 5.13.8 • Efficiency of single-stage turbines (Courtesy of IMO Industries)

- Steam turbines are not usually shop performance tested
- They are only tested mechanically (vibration, bearing temperatures, etc.) at no load

  Because:
- Performance (steam rate, efficiency) varies directly with steam velocity
- Steam velocity varies directly with steam flow
- Steam flow varies directly with power requirement because:

  Power = ΔH (energy per unit mass) $\times$ mass flow rate
- Testing at full load is not cost effective

Fig 5.13.9 • Steam turbine performance testing

- Non-extraction turbines present performance on a Willans-line
- A Willans-line describes the amount of steam flow (throttle flow) required for a given load at a given speed
- An extraction map describes throttle flow for a given load and extraction flow

  Note: all curves are for a specific set of steam conditions

Fig 5.13.10 • Steam turbine performance curves

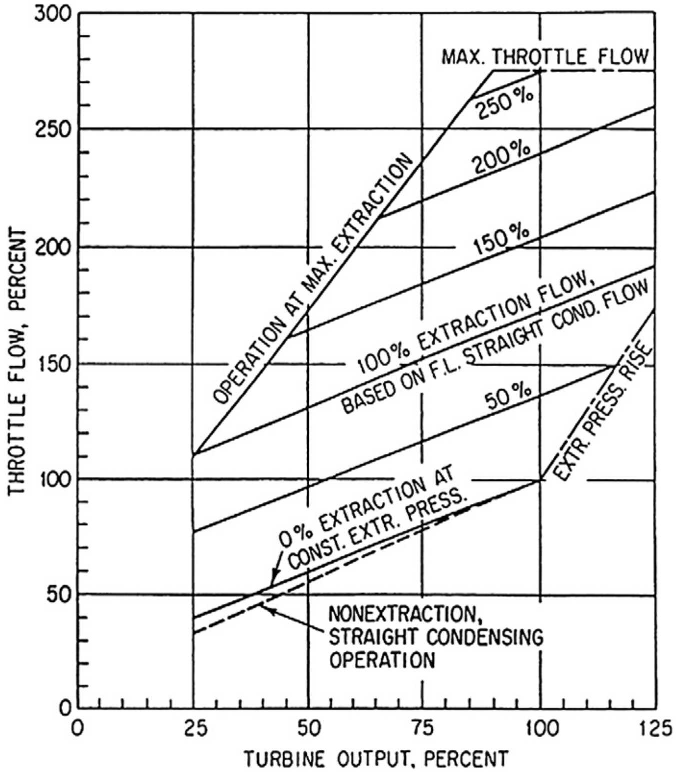
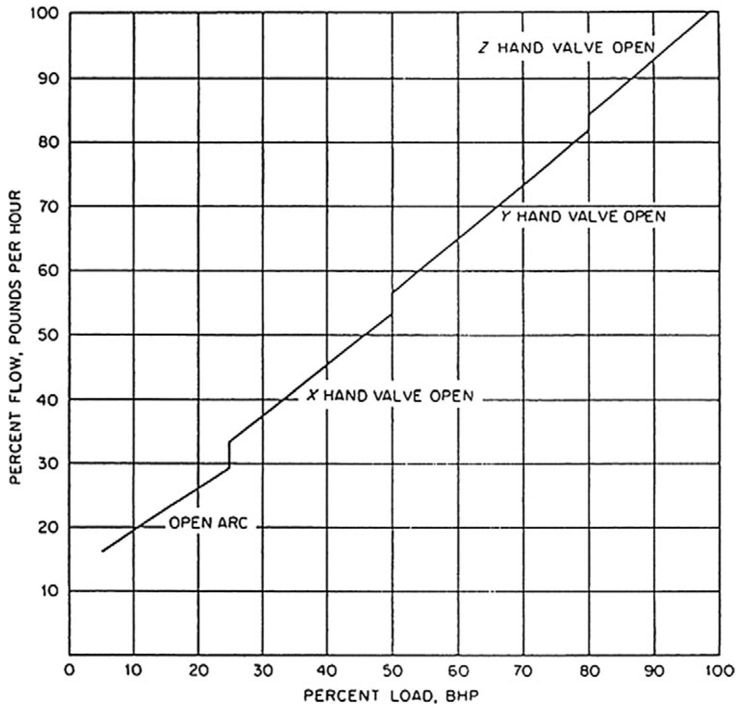


Fig 5.13.12 • Performance curve for a typical extraction steam turbine (Courtesy of IMO Industries)



Partial-load Willans lines.

Fig 5.13.11 • Typical performance curve for a single stage turbine with manual hand valves (Courtesy of IMO Industries)

Best Practice 5.14

Install a low pressure steam seal eductor and oil condition monitoring bottles on single stage turbines to prevent and monitor bearing oil contamination.

A common reliability issue with single stage steam turbines is the undetected steam seal wear and entrance of condensate into the small volume bearing housings.

Steam seal wear will rapidly displace bearing oil with condensate causing Babbitt-lined, sleeve bearing wear with potentially dangerous consequences.

Installation of a low pressure steam eductor and associated piping with a vacuum gauge from each seal leak off port will function in the same manner as special purpose steam turbine seal systems (see BP 5. 8), and positively prevent the entry of condensate into the bearing housing.

Installation of oil condition monitoring bottles at the bottom of each bearing bracket will allow operations to monitor bearing oil condition by observing water accumulated in the bottom of the bottle.

Lessons Learned

Failure to design single stage turbine steam seal systems for vacuum systems and bearing bracket oil condition monitoring bottles has resulted in frequent turbine shutdowns for seal and oil changes and catastrophic bearing failure and fires (resulting from 'red hot' bearing housings).

Benchmarks

This best practice has been used since the mid 1990s for recommended field modifications and project requirements. Major oil companies have adapted this best practice to produce steam seal MTBFs in excess of 100 months.

B.P. 5.14. Supporting Material

Single stage turbine guidelines

The five common problems with single stage turbines are noted in Figure 5.14.1.

- Bearing bracket oil contamination (inadequate carbon ring steam seal design)
- Slow governor system response (inadequate governor linkage maintenance and governor power)
- Hand valve(s) closed on critical services
- Bearing bracket oil viscosity reduction and bearing wear (high pressure service)
- Use of sentinal valves on turbine cases

Fig 5.14.1 • Single stage steam turbines: common reliability problems

We will now discuss each problem in detail. Please refer to Figure 5.14.2 which has each problem area circled.

Bearing bracket oil contamination

Please refer to Item 1 in Figure 5.14.2.

The most common reliability problem with single stage steam turbines is the contamination, with water, of the oil in the bearing housing. The root cause of the problem is the ineffectiveness of the floating carbon ring shaft seal system.

Unless site systems are modified to eliminate the root cause, the best plan is to minimize the effect of the contamination so a bearing failure will not occur. Such an action plan is presented in Figures 5.14.3, 5.14.4 and 5.14.5.

Slow governor system response

Please refer to Item 2 in Figure 5.14.2. Another very common reliability problem is the slow or non-movement of the governor system linkage during start-up, and during normal operation when steam conditions change. It appears as if the governor is not responding because speed will not be controlled when it should. Typical examples are:

- Speed will continue to increase when throttle valve is opened; turbine will trip on over speed
- Speed will increase or decrease when:
  - Steam conditions change
  - Driver equipment changes

These facts are presented in Figure 5.14.6.

Since most single stage steam turbines are not supplied with tachometers, it is difficult, if not impossible, to condition monitor this problem. A condition monitoring action plan is provided in Figure 5.14.7.

The usual root cause of the problem is that the friction in the mechanical linkage and/or valve stem packing exceeds the maximum torque force that the governor output lever can deliver. The governor designations TG-10, TG-13 and TG-17 simply mean 'turbine governor with FT-LB torque'. Therefore, if a TG-10 governor is installed, and the torque required to move the valve steam exceeds the value of 10 FT-LBs, the governor system will not control speed. Taking the governor to the shop will not solve the problem. Causes of excessive friction are shown in Figure 5.14.8, and a plan to eliminate this reliability problem is presented in Figure 5.14.9.

Hand valve(s) closed on critical services

Most single stage steam turbines are supplied with one or more hand valves in the steam chest. Refer to Figure 5.14.2, Item 3. The purpose of the hand valves is to allow more or fewer inlet steam nozzles to be used during operation. Optimizing the steam nozzles used maintains turbine efficiency during load changes. **However, the efficiency of single stage steam turbines is only 35% at best!** Therefore, adjustment of hand valves, other than during start-up or during slow roll, should not be required. Figure 5.14.10 is a top view of a one hand valve.

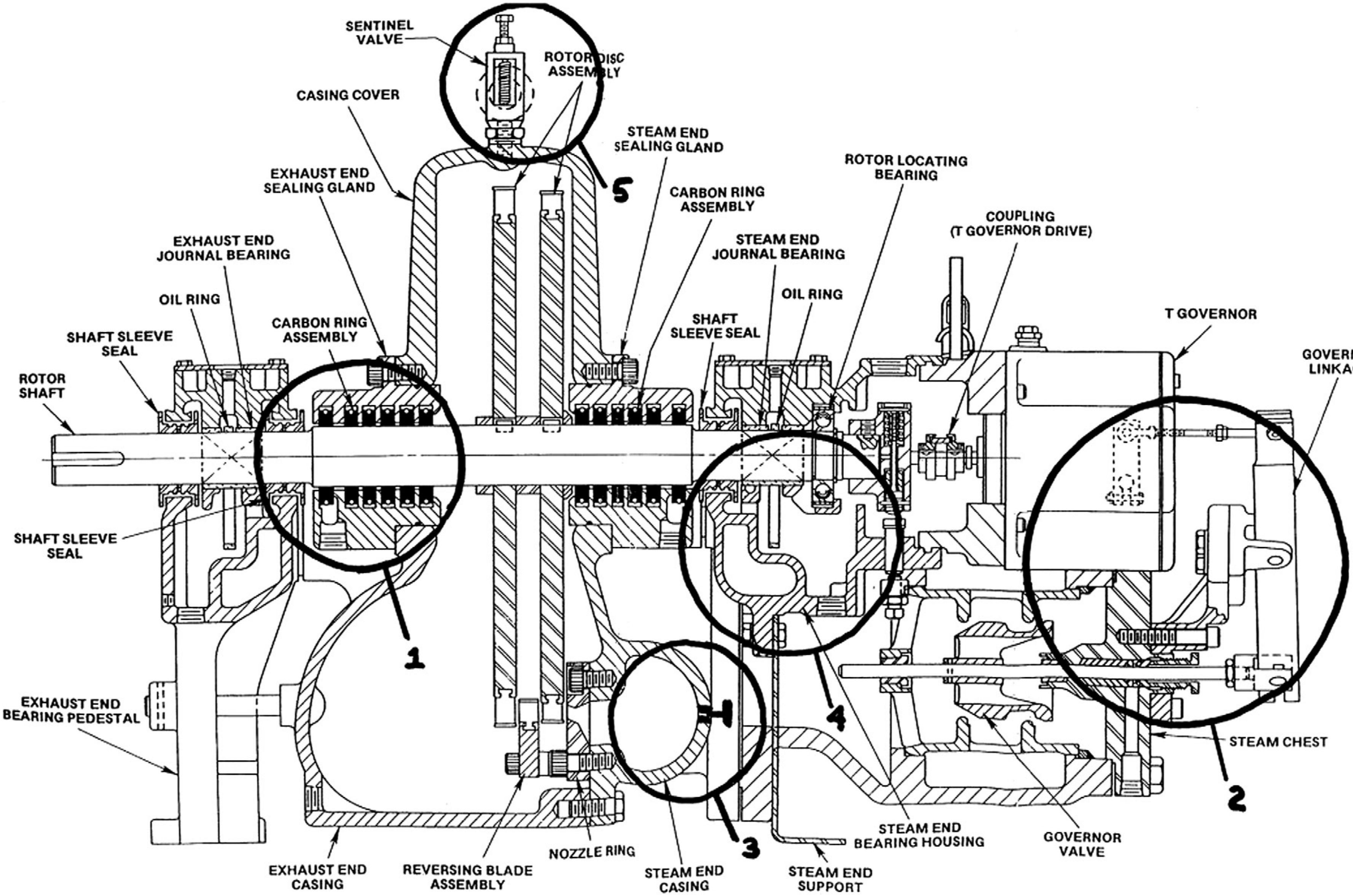


Fig 5.14.2 • Single stage steam turbines: common reliability problems

- Shaft carbon ring seal cannot positively prevent steam leakage

Fig 5.14.3 • Bearing bracket oil contamination (root cause)

- Install oil condition site glasses in bearing bracket drain connection
- Inspect once per shift
- Drain water as required
- Sample oil monthly initially

Fig 5.14.4 • Steam turbine bearing bracket oil contamination monitoring action plan

- Install steam eductor on each seal chamber leak off drain (between 4th and 5th carbon ring)
- Design eductor to pull 5–10" of H_2O vacuum at this point
- Alternative approach – install bearing housing isolation seal ('Impro' or equal)

Fig 5.14.5 • How to correct carbon ring seal ineffectiveness

1. Rapid speed change and trip on start-up
2. Speed increase or decrease on steam condition or load condition change
3. Governor instability (hunting) around set point

Note: #1 usually occurs on "solo", #2 occurs during steady state operation

Fig 5.14.6 • Slow governor system response

- Install tachometer on all single stage steam turbines
- Always test speed control on "solo run"(1)
- Monitor turbine speed once per shift. Take corrective action if speed varies +/− 5% (200 rpm)

Note: (1) since load is very low, test acceptance is the ability to stabilize speed and prevent overspeed trip when throttle valve is slowly opened.

Fig 5.14.7 • Slow governor system response condition monitoring action plan

We have witnessed many unscheduled shutdowns of critical (un-spared) compressor units, because the general purpose steam turbine that is the main lube oil pump driver had the hand valves closed. An upset in the steam system reduced

- Linkage bushings not lubricated with high temp. grease
- Valve steam packing too tight
- Steam deposits in valve and/or packing after extended shut down (turbine cold)
- Bent steam valve stem

Fig 5.14.8 • Causes of excessive governor mechanical linkage system and valve friction

- If problems occur (Box 5.47), disconnect linkage and confirm ease of valve movement
- Replace bushings and/or lubricate with "molycote" or equivalent
- Clean deposits from valve and packing as required
- If above action does not correct problem, replace governor (inspection and/or adjustment of governor droop is required)

Fig 5.14.9 • Slow governor system response condition monitoring action plan

steam supply pressure, and caused the turbine and lube pump to slow down. This was because hand valves were closed and the throttle valve, even when fully open, could not meet steam flow requirements. When the speed of the steam turbine decreased, the lube oil pressure dropped and – guess what? – the auxiliary pump did not start in time and the unit tripped.

Figure 5.14.11 presents the recommended action plan refinery for single stage steam turbine hand valves.

Bearing bracket oil viscosity reduction and bearing wear on high pressure single stage steam turbines

Please refer to Figure 5.14.2, Item 4. Observe the jacket in the bearing housings. The purpose of this jacket is to cool the oil in the bearing bracket. When the inlet steam pressure is high, the heat of the steam is transmitted to the steam end inlet bearing through the shaft. Although the jacket in the bearing housing does reduce the oil temperature in the bearing housing, it cannot effectively reduce the oil temperature at the shaft/bearing interface. Figure 5.14.12 presents these facts.

This problem is a design issue. A small, single stage turbine is not provided with an oil system effective enough to remove the heat between the shaft and bearing when the turbine is operating on high temperature 400°C (750°F) steam. The solution is to require pressure lubrication for this application.

Naturally, it is difficult, and not cost effective, to retrofit these turbines for pressure lubrication. The solutions to this problem that are proven in the field are presented in Figure 5.14.13.

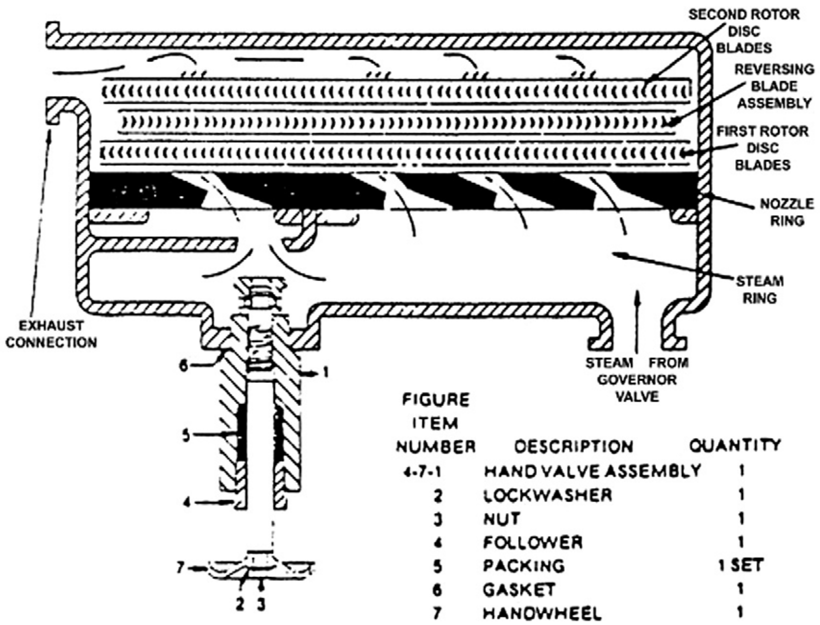


FIGURE ITEM NUMBER	DESCRIPTION	QUANTITY
4-7-1	HAND VALVE ASSEMBLY	1
2	LOCKWASHER	1
3	NUT	1
4	FOLLOWER	1
5	PACKING	1 SET
6	GASKET	1
7	HANDWHEEL	1

Hand Nozzle Valve Assembly

FEATURES:

- **NOZZLE BLOCK IN LOWER HALF OF CASE**
- **ONE OR MORE HAND VALVES FOR EFFICIENCY AND INCREASED LOAD**
- **HAND VALVES NOT MODULATING!**
  (FULL OPEN OR FULL CLOSED)

Fig 5.14.10 • Single valve turbine admission path

Continued use of sentinel valves on turbine cases

Please refer to Figure 5.14.2, Item 5. Sentinel valves were used, years ago, as alarm devices to indicate that the steam turbine case (low pressure part) was under excessive pressure. These devises are not pressure relief valves and will not protect the case from failure during over pressure events.

It is a known fact that the sentinel valves wear, leak, and require steam turbine shutdown for repair. Most large company specifications prevent the use of sentinel valves, and require full relief-valve protection on the inlet and exhaust of all single stage turbines. These facts are presented in Figure 5.14.14.

- Never throttle hand valves
- Hand valves should be open on main oil pump and auto-start steam turbines

Fig 5.14.11 • Single stage steam turbine hand valve recommendations

- Sleeve bearings (usually steam inlet end) wear out quickly
- Oil viscosity is reduced and difficult to maintain

Fig 5.14.12 • High pressure single stage steam turbine bearing problems and oil viscosity reduction

- Assuring bearing housing jacket passages are open (flushed)
- Consulting with turbine vendor for bearing material change
- Using special high temperature service oil (synthetic based oil)

Fig 5.14.13 • Eliminate bearing wear and oil viscosity reduction (high pressure service)

- Removing sentinel valves
- Assuring that inlet and exhaust casings are protected by properly sized and set pressure relief valves

Fig 5.14.14 • Prevent excessive sentinel valve maintenance

Best Practice 5.15

Single valve steam turbines in all services should have dedicated tachometers to monitor governor linkage system condition.

Single stage steam turbines have governor mechanical linkages and packing that can create friction loads which exceed governor actuator forces. This can render the governor system unable to accommodate steam system condition changes.

Dedicated tachometers allow operators to quickly detect reduced speeds, and request maintenance activity to free up friction bound bushings and packing before the reduced turbine speed can affect plant operation.

Note that mechanical/hydraulic governors can be easily modified to include local tachometers at a minimum cost.

Lessons Learned

Frequent, critical (un-spared), compressor train trips have resulted from undetected friction-bound governor systems. They resulted in reduced main oil pump reduced turbine speed, which required the oil system motor driven auxiliary oil pump to start. In these cases, the auxiliary pump did not start in time, causing a unit trip and revenue losses that approached and in some instances exceeded $1m USD.

Benchmarks

This best practice has been recommended since the 1990s. Clients have reported that past unit trips have been eliminated when tachometers are installed and operators are instructed regarding the monitoring of these systems.

B.P. 5.15. Supporting Material

Please refer to material in B.P. 5.14.

Best Practice 5.16

Always keep single valve turbine hand valves open and do not throttle them.

There is a misconception that closing certain hand valves on single valve turbines will save steam and increase efficiency.

While it is true that steam consumption can be reduced by closing hand valves, the efficiency of single stage turbines does not exceed 35%.

Single stage turbines are generally used to protect the plant against electrical power outages that would shut down critical pump or fan services.

Closed hand valves expose the steam turbines to reduced speed during steam header upsets and should always be fully open.

Finally, hand valves are designed to be only fully open or closed and should never be throttled.

Lessons Learned

Keeping hand valves closed in critical services (oil system main pump drivers, BFW, FD fan drivers, etc.) has resulted in process unit ESDs costing the plant millions of USD in lost revenue.

Hand valves are too often observed to be closed on main lube oil pump steam turbines. This exposes the turbine to slow down on steam header interruptions, which will require the auxiliary motor driven pump to immediately start without a drop in oil pressure to the trip level. Many process unit shutdowns have occurred due to this scenario.

Benchmarks

This best practice has been used for all field auxiliary system audits since 1990, and has resulted in compressor train reliabilities exceeding 99.7% in critical compressor trains.

B.P. 5.16. Supporting Material

Please refer to material in B.P. 5.14.

Best Practice 5.17

Check the freedom of movement of a single valve, steam turbine, governor system every three months, by slowly closing the turbine inlet block valve and observing governor valve stem movement. Note: First confirm that the governor valve is not fully open.

Single stage steam turbine linkages can easily become friction bound.

Check their freedom of movement every three months and free up linkages and coat all bushings with Moly Cote (high temperature grease), or equivalent if required.

Linkages commonly become friction bound after a turnaround or extended shutdown, since any steam deposits can set up and harden in the valve stem packing area.

Lessons Learned

Failure to exercise single valve steam turbines has led to critical compressor train trips because:

- A tachometer was not installed
- Friction bound governor linkages have resulted in undetected speed reductions during steam header upsets
- The motor driven auxiliary pump did not start in time to prevent a low lube oil pressure trip

Benchmarks

This best practice has been recommended since 1985, when multiple friction-bound linkage issues were experienced during the start-up of a chemical plant in the Middle East.

B.P. 5.17. Supporting Material

Please refer to material in B.P. 5.14.

6

Gas Turbine Best Practices

Introduction

Of all available alternatives, the gas turbine has been the slowest to be accepted as a viable driver for all possible applications. While it is true that the gas turbine is the newest prime mover, its rise from pipeline compressor drivers in the 1950s has been slow and bumpy, somewhat because of unfamiliarity with its operation, and reputation for high maintenance. The present generation of gas turbines offers viable alternatives, however, with maintenance intervals as high as 60,000 hours for all possible applications.

This chapter will cover the best practices for these important prime movers.

Best Practice 6.1

Always consider aero-derivative gas generator/industrial power turbine (hybrid type) gas turbine units.

Aero-derivative gas turbines (initially designed for flight and using anti-friction bearings) have been thought to have lower MTBFs than industrial types. End user fleet experience has shown anti-friction bearing life to exceed 100 months.

The use of an aero-derivative gas generator while using an industrial type power turbine (with hydrodynamic bearings) has the following advantages:

- Shorter start-up and shut down time
- Equal or greater bearing MTBF than industrial type gas turbines
- Module replacement of the gas generator (as low as 36 hours guaranteed) without the necessity for gas generator to power turbine alignment

Lessons Learned

Industrial gas turbines are used for mechanical drive applications based, on the pre-conceived idea that they have a higher reliability than aero-derivative/industrial power turbine units. There has finally been a realization by end users that greater MTTR times, longer start-up and cool-down times for industrial types outweigh any reliability advantages for the industrial – if one actually exists.

Benchmarks

The use of hybrid (aero-derivative/industrial) gas turbines, for mechanical dive and generator applications above 15 MW site power rating, has been recommended in project design phases since 1990.

B.P. 6.1. Supporting Material

In this section, we will discuss functions and types of gas turbines. In my personal experience, the gas turbine is the most misunderstood rotating equipment item. Because of its many support systems and various configurations, the gas turbine is often approached with mystery and confusion. In order to thoroughly explain the gas turbine from a functional standpoint, we will build on prior knowledge. We will also compare the gas turbine to an automotive engine in terms of its combustion cycle. Having done this, we will then use a building block approach to explain the total configuration of a gas turbine and conclude with a brief history of its evolution.

Gas turbine classifications will then be presented, specifically:

- Design type
- Number of shafts
- Drive and number of shafts
- Cycle
- Drive and location

We will discuss the major design differences between aero-derivative and hybrid (aero-derivative gas generator/industrial power) turbines. Single and multiple shaft gas turbines will be discussed and reviewed. The three major application cycles for gas turbines: simple, regenerative, and combined will be presented and discussed.

Finally, we will present applications of different gas turbine types, and provide information concerning where the different types are used.

Comparison to a steam turbine

Figure 6.1.1 shows a typical condensing steam turbine and an industrial gas turbine. The major difference between the two is that a steam turbine is an external combustion engine, whereas a gas turbine is an internal combustion engine. That is, the motive fluid for a steam turbine is generated external (in the boiler) to the engine. In the case of a gas turbine, the motive fluid is generated internal to the engine (air compressor and combustor).

Figure 6.1.2 compares the gas and steam turbine cycles; the latter is known as the 'Rankine cycle'. As shown, the hot vapor is generated in the boiler, which is external to the steam turbine (expander).

The gas turbine cycle is known as the 'Brayton cycle'. Here, air is brought into the engine by the axial compressor, and combined with fuel and an ignition source in the combustor to produce a hot vapor, which then is expanded through the HP (high pressure) turbine. The combination of the compressor, combustor and HP turbine is commonly known as the gas generator. This is because its function is to generate or produce a hot vapor from the combination of an air and fuel mixture. Essentially, a gas generator can be considered to have the same function as a boiler – both produce a hot vapor. One can think of the gas generator as a 'rotating boiler'. After the hot vapor is generated, it then is expanded additionally in the power turbine. The power turbine, therefore, serves exactly the same function as the steam turbine. That is, both components are hot gas expanders.

Comparison to an automotive engine

Figure 6.1.3 shows the similarities between an automotive engine and a gas turbine. If one considers a gas turbine as only a dynamic internal combustion engine, understanding it becomes significantly easier. As shown in the figure, an automotive engine is a positive displacement internal combustion engine, having an intake, compression, combustion and exhaust stroke. A gas turbine engine is a dynamic internal combustion engine. The process in this case is continuous – and not intermittent as

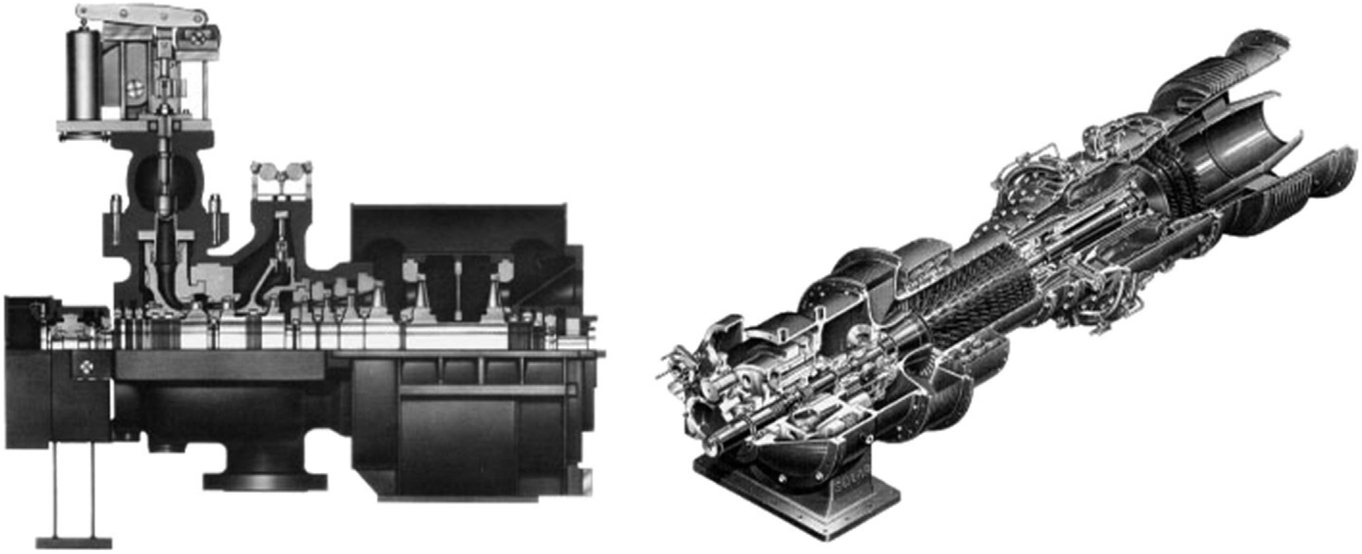


Fig 6.1.1 • A gas turbine vs. a steam turbine. Left: Steam turbine (Courtesy of General Electric Co.). Right: Mars gas turbine (Courtesy of Solar Turbines, Inc.)

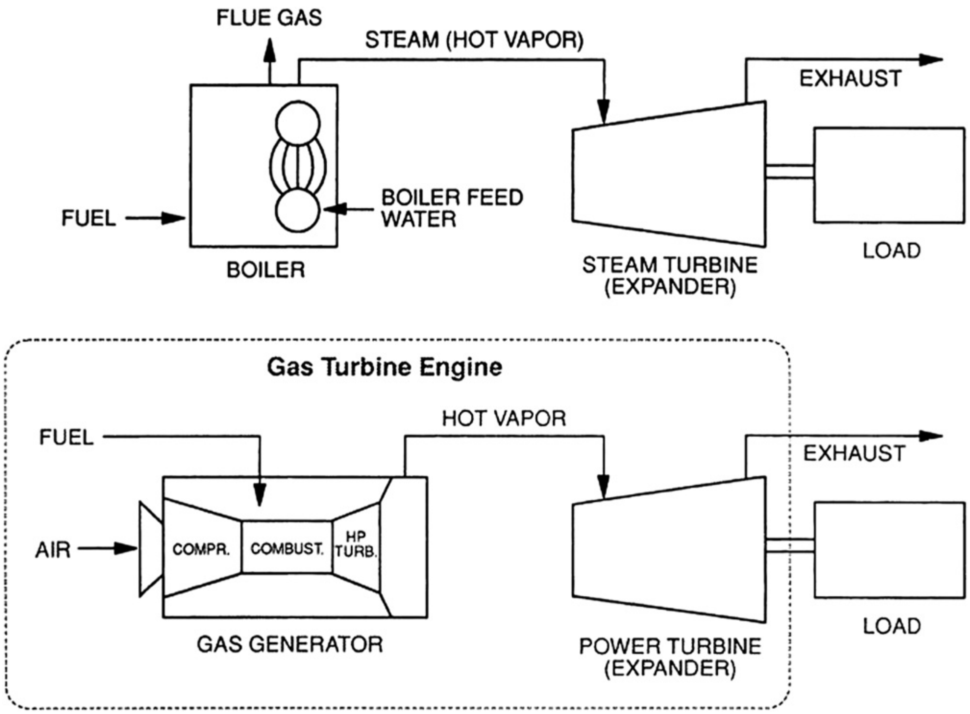


Fig 6.1.2 • Comparison – gas turbine vs. steam turbine cycles

is the case for the automotive engine. Both engines have compression, combustion and exhaust sections. When one considers the similarities of these engines in this manner, it can be seen that both require starters, ignition sources, inlet air filters, inlet fuel systems, cooling systems, and monitoring systems.

Building a gas turbine

We have already discussed turbo compressors and expansion turbines. A gas turbine is a combination of these components, with the addition of a combustor that produces the hot gas for expansion. Figure 6.1.4 presents these facts.

Figure 6.1.5 shows a gas turbine configuration for a gas turbine with a regenerator. The function of the regenerator is to use gas generator exhaust vapors to preheat the air exiting the air compressor, thus reducing the amount of fuel required by the gas turbine. Figure 6.1.5 also shows the changes of gas temperature, pressure, energy, and the horsepower produced in the different sections of the gas turbine. Note that the power produced in a gas turbine is typically three times the output

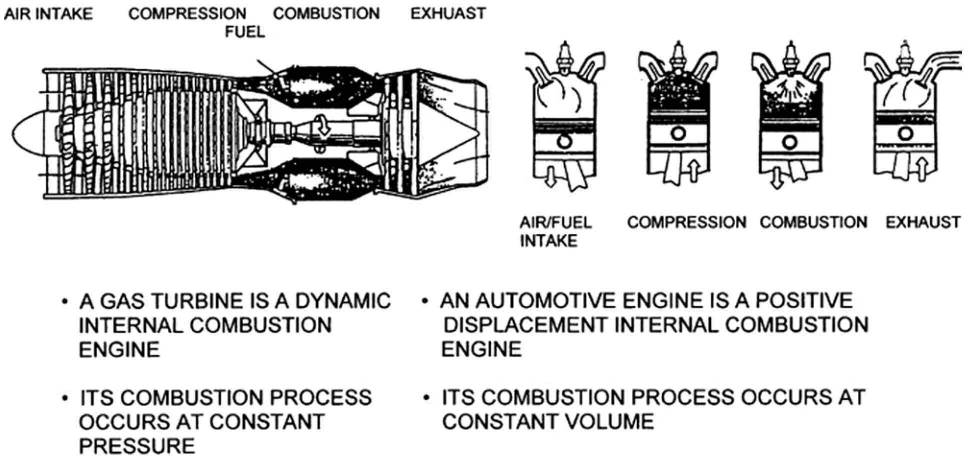


Fig 6.1.3 • Gas turbine vs. automotive engine (Courtesy of Dresser-Rand)

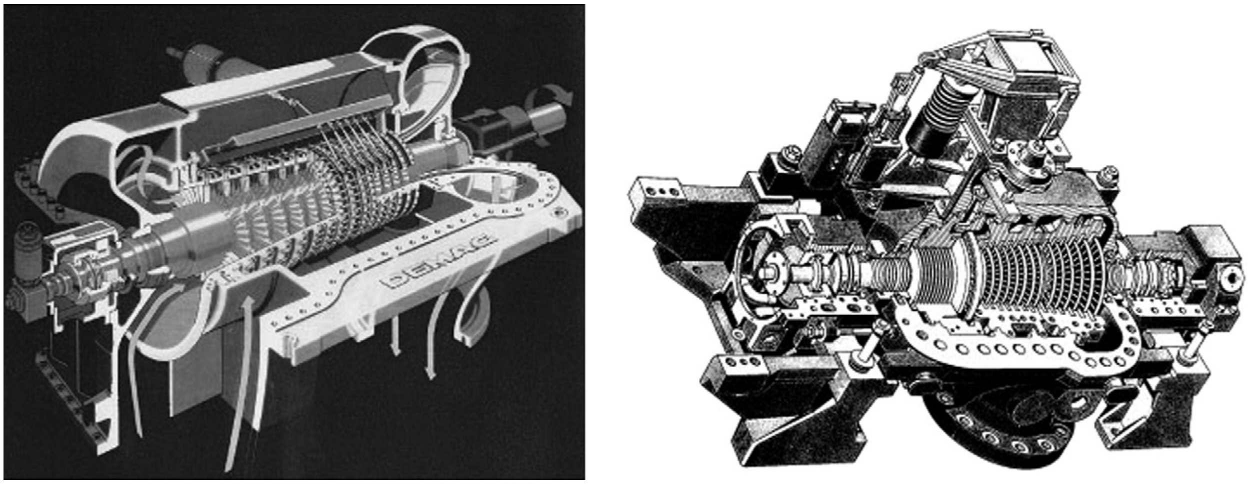


Fig 6.1.4 • Building a gas turbine

power. This is because the air compressor typically requires two thirds of the produced power.

History of gas turbine development

A brief history of gas turbine development is presented in Figure 6.1.6. Gas turbines were initially used in the early 1940s for military purposes. In the 1950s, gas turbines first entered mechanical service applications. Since gas turbines are production type equipment and not custom designed, we usually refer to generations of gas turbines. The first generation of gas turbines began in the 1950s, and progressed through the 1970s (second generation) and 1980s (third generation) to present day efficiency improvements.

Gas turbine classifications

In this section, we will discuss the different gas turbine classifications in terms of design type, number of shafts, drive location, and cycle.

Classification by design type

Figure 6.1.7 presents the two types of industrial gas turbine. The older type was grass roots industrial; that is, it was never built to function as an aircraft engine. The more modern approach (late 1960s) is the aero-derivative influenced industrial gas turbine. This design evolved from the aircraft industry, and is lighter in weight. Maintenance is easier than the grass roots industrial type, since components are modularized and are changed out as opposed to the individual parts in the grass roots industrial. Both industrial types are differentiated from aero-derivative types by the fact the radial bearings are always hydrodynamic. Facts, advantages, and disadvantages concerning industrial turbines are presented in Figure 6.1.8. In recent years, the emphasis on ease of maintenance has favored the aero-derivative type gas turbine over either type of industrial gas turbine presented here. This is because the maintenance times for the aero-derivative gas turbines in the field are significantly less than industrial type gas turbines, since the aero-derivative unit can be easily exchanged with a similar unit in the field. Therefore, the field maintenance time is significantly lower for the aero-derivative gas turbine (typically 72 hours as opposed to 360 hours or more).

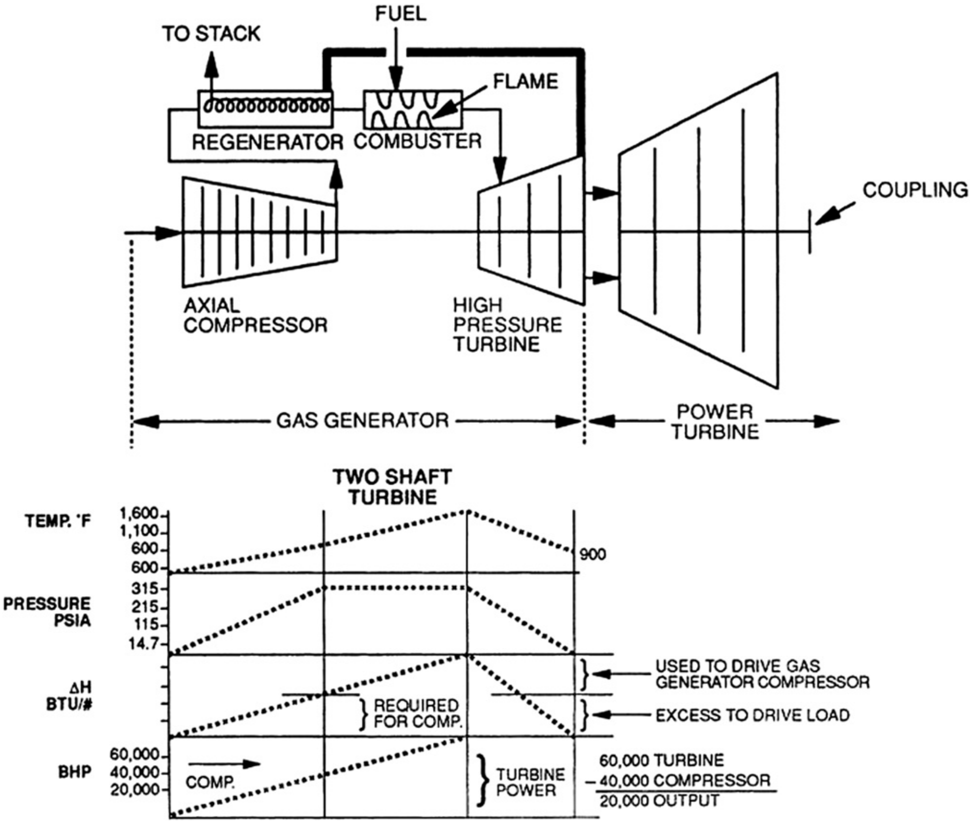


Fig 6.1.5 • Gas turbine configuration

Year	Milestone
1900–30	Various works – expansion turbines (Delaval, Parsons, etc.)
1930	Sir Frank Whittle granted gas turbine patent
1943	First successful gas turbine (jet engine)
1950s	Industrial gas generators and power turbines used in pipeline service (1st generation)
1960s	Improved efficiency through use of material and cooling improvements. Use in power generation and industrial plants
1970s	■ Development of 2nd generation – larger sizes, higher efficiency (higher firing temperatures) ■ First uses of aero-derivative gas generators and power turbine for 'off shore' applications ■ Increased availability ■ Increased preventive maintenance cycle time
1980s	■ Extensive use of gas turbines in combined cycles for co-generation ■ Aero-derivative types gain further acceptance ■ Continued efficiency increase (higher firing temperatures) – retrofits of 1st generation units ■ Development of 3rd generation gas turbines (use of advanced materials, processing, coating and cooling techniques)
1990s	■ Further acceptance of gas turbine as an industrial prime mover ■ Simple cycle efficiencies approach 45%. Firing temperatures approach 1400°C (2500°F)

Fig 6.1.6 • Gas turbines – history of development

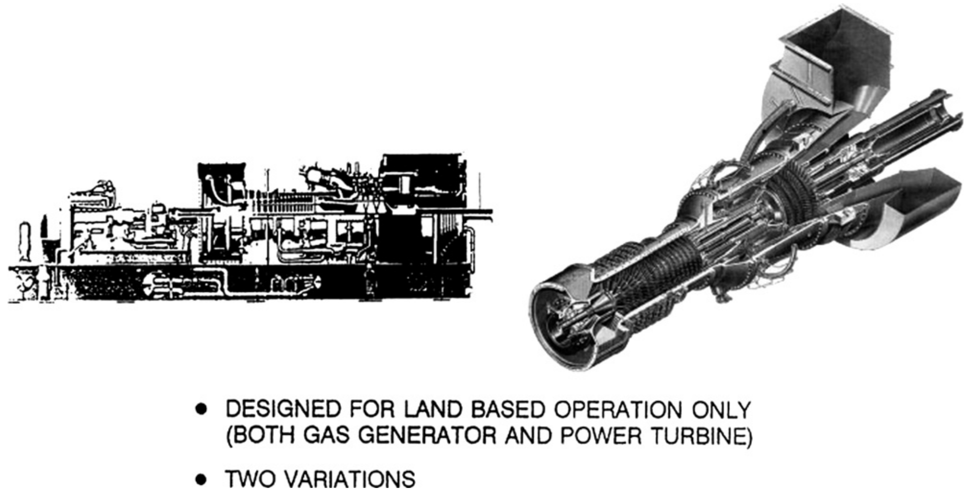


Fig 6.1.7 • Gas turbine classifications (industrial type). Left: Grass roots industrial (never built to fly) (Courtesy of General Electric Co.). Right: Aero-influenced industrial (lighter weight hydrodynamic bearings) (Courtesy of Solar Turbines, Inc.)

Advantages	Disadvantages
■ Longer cycle time between maintenance	■ Longer maintenance times
■ Longer bearing life (hydrodynamic bearings)	■ Large foot print
■ Greater tolerance to upsets	■ High specific weight
	■ Lower efficiency (1st and 2nd generation)
	■ Longer start sequences

Fig 6.1.8 • Industrial type gas turbines

A single shaft grass roots industrial gas turbine – General Electric model 7000 (Frame 7) – is shown in Figure 6.1.9. This turbine has been used for both generator drives and mechanical drives. Nominal ISO horsepower is in the 100 MW (135,000 BHP) range. Efficiency is approximately 35%.

Figure 6.1.10 shows an example of a two shaft aero-derivative gas turbine – Solar Mars gas turbine – used for mechanical drive applications (compressor and pump drives). Nominal ISO horsepower is in the 11 MW (15,000 BHP) range. Efficiency is approximately 35%.

Figures 6.1.11, 6.1.12 and 6.1.13 are examples of various aero-derivative gas turbines. In Figure 6.1.11, a General Electric LM 2500 gas turbine is shown in two applications. The first is a gas generator for a Dresser Rand DJ 270R power turbine, and the second uses the LM 2500s six-stage power turbine on a separate shaft. Nominal ISO horsepower is in the 20 MW (25,000 BHP) range. Efficiency is approximately 37%.

Figure 6.1.12 is a drawing of a Rolls Royce RB211 two-shaft gas turbine. It has intermediate and high pressure axial compressors mounted on separate shafts for increased efficiency. Nominal ISO horsepower is in the 23 MW (30,000 BHP) range. Approximate efficiency is 38%.

The newest aero-derivative gas turbine used for generator and mechanical drive is the LM 6000 two-shaft gas turbine shown in

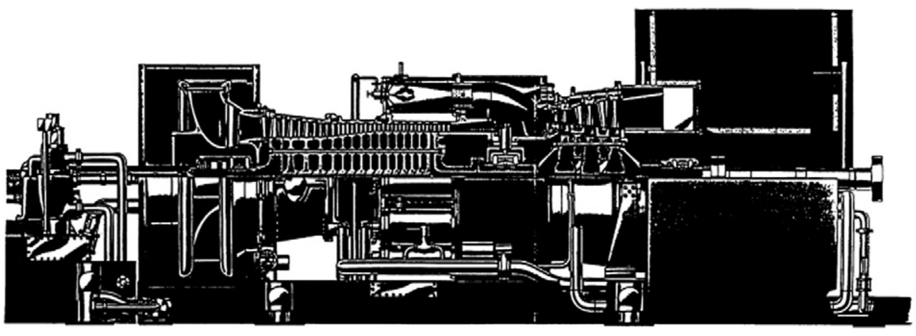


Fig 6.1.9 • Single shaft industrial gas turbine (Courtesy of General Electric)

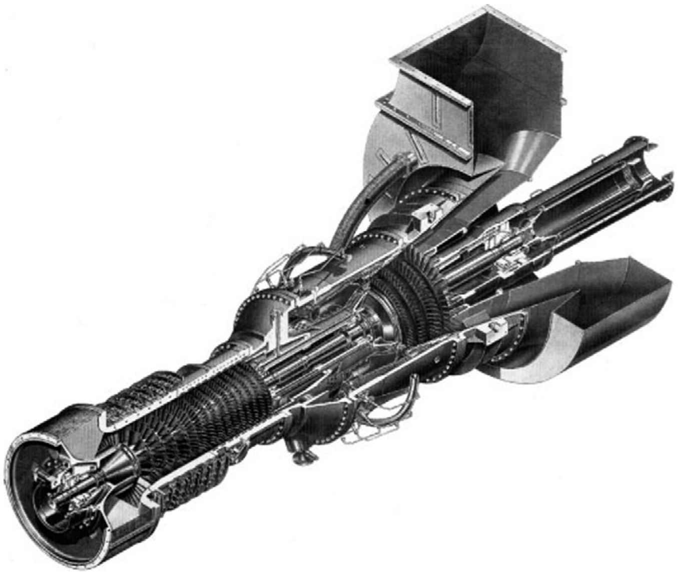
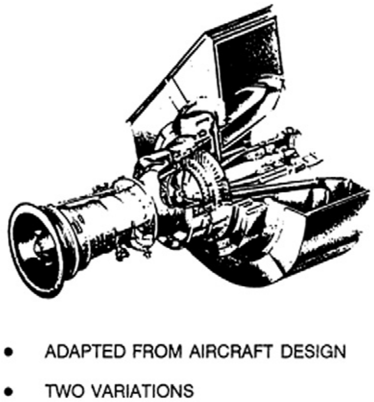


Fig 6.1.10 • Mars gas turbine (Courtesy of Solar Turbines)



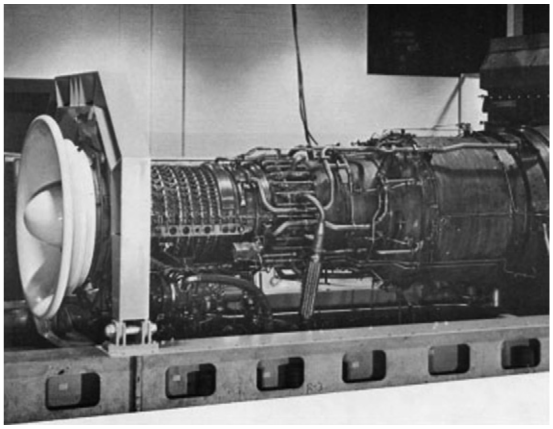


Fig 6.1.11 • Gas turbine classifications (aero-derivative). Left: used as gas generator only (Courtesy of Dresser Rand). Right: entire engine adapted (gas generator and power turbine). Note: power turbine is either turbo-prop or bypasss fan drive in aero-engine version (Courtesy of General Electric).

Figure 6.1.13. This produces 45 MW (60,000 ISO horsepower), has an efficiency of 43% and can drive a load on either or both ends.

The advantages and disadvantages of aero-derivative gas turbines are presented in Figure 6.1.14.

Figure 6.1.15 shows a hybrid type gas turbine, which is a combination of an aero-derivative gas generator and an industrial power turbine. This design offers the advantage of maintainability on the 'hot section' of the gas turbine and high reliability in the power turbine. These facts are presented in Figure 6.1.16.

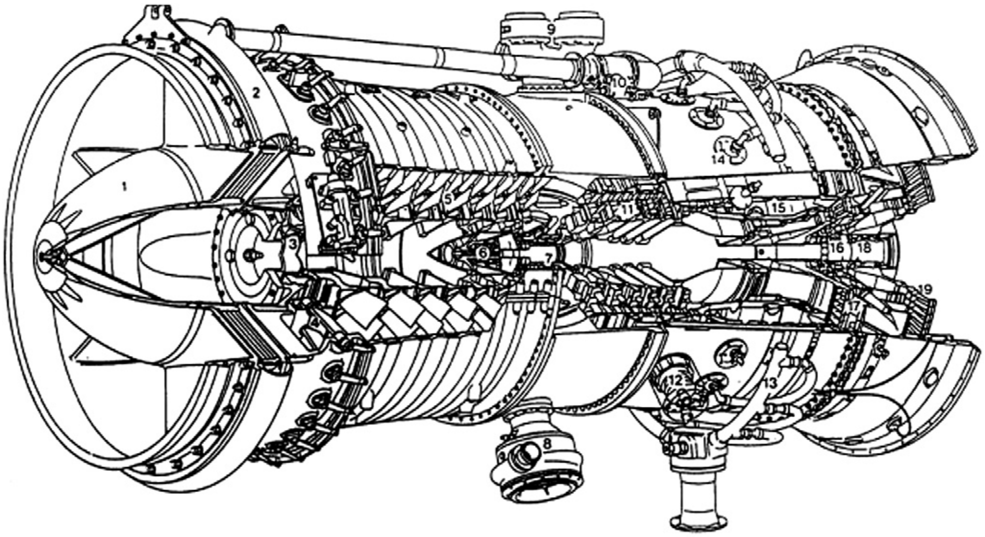
Aero-derivative and industrial facts are discussed in Figure 6.1.17.

The number of gas turbine shafts

Gas turbines are configured as single, dual or triple shaft designs. The advantages and disadvantages of each type are presented in Figure 6.1.18. Most modern gas turbines are of the triple shaft design. Figure 6.1.19 shows a single shaft gas turbine where the gas generator and power turbine are mounted on the same shaft. This figure also shows a dual shaft gas turbine, where the gas generator and power turbine are mounted on different shafts. Single shaft gas turbines are usually limited to generator drive applications, since the starting turbine load is significantly lower for a generator application, because a generator is started under zero load. Dual shaft turbines are used for mechanical drive, pump and compressor applications.

Gas turbine drive configurations

Gas turbines can be designed as hot end drive, or cold end drive. Figure 6.1.20 presents these facts. The majority of first and second generation gas turbines were of a hot end drive, but most third generation gas turbines are of the cold and dry type. A cold end drive configuration is more reliable, in my opinion, since the



1 Hose fairing
2 Anti-locking duct
3 IP compressor front bearing'
4 Variable inlet guide vane
5 IP compressor
6 IP location bearings
7 HP location bearings
8 Air starter
9 IP handling bleed valves
10 Anti-locking hot air valves
11 HP compressor
12 Starting bleed valve
13 Gas fuel manifold
14 Gas fuel burner
15 Combustion chamber
16 HP turbine bearing
17 HP turbine
18 IP turbine bearing
19 IP turbine

Fig 6.1.12 • The industrial RB211 (Courtesy of Rolls Royce)

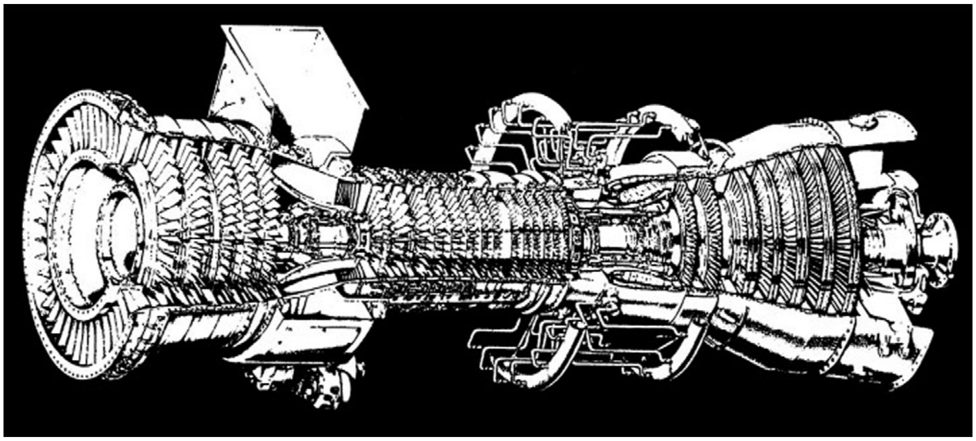


Fig 6.1.13 • LM6000 gas turbine (Courtesy of General Electric)

Advantages	Disadvantages
■ Shorter maintenance times	■ *Shorter cycle time between maintenance
■ Small foot print	■ Less tolerance to upsets
■ Low specific weight	■ Shorter bearing life (anti-friction bearings)
■ Higher efficiency	
■ Faster start sequence	

*Note: maintenance cycle time is increasing and approaching industrial types

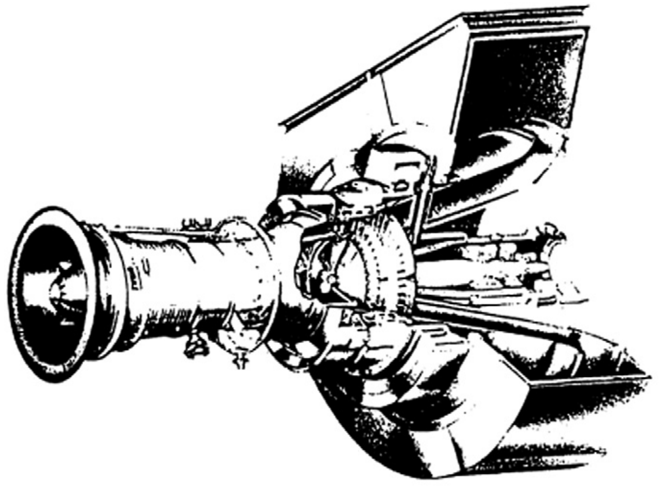
Fig 6.1.14 • Aero-derivative type gas turbines

coupling environment is significantly reduced in terms of temperature. This results in a much lower axial expansion of the drive coupling and subsequently increases the reliability of the gas turbine.

Gas turbine cycles

Gas turbine cycles are presented in Figure 6.1.21. There are essentially three types of gas turbine cycles: first is the simple cycle, where the gas is exhausted directly to atmosphere; secondly, the regenerative cycle, where the exhaust gas is used in an exchanger (regenerator) to preheat the compressor discharge air prior to the combustor; and finally there is the combined cycle, where the exhaust gas is used in a heat recovery steam generator (HRSG) to either generate steam for plant use or as an expansion fluid is a steam turbine. Typical efficiencies are as follows:

- Simple cycle 20% to 43%
- Regenerative cycle 30% to 45%
- Combined cycle 55% to 60%



- LOAD (POWER) TURBINE IS INDUSTRIAL TYPE
- GAS GENERATOR IS AERO-DERIVATIVE TYPE

Fig 6.1.15 • Gas turbine classifications (hybrid type industrial)
(Courtesy of Dresser Rand)

- Usually based on power turbine type
- Depends on types of bearings
- Anti-friction = aero
- Hydrodynamic = industrial
- With time, both types will converge to a ‘hybrid’
- Current 3rd generation designs are moving in this direction

Fig 6.1.16 • Classification of industrial and aero-derivative gas turbines

Item	Aero-derivative	Industrial
Casing weight	Light	Very heavy
Casing material yield	3 times higher yield strength	–
Rotor weight	15–20 times lighter	–
Bearing type	Anti-friction	Hydrodynamic
Bearing life	50,000 hours	50,000–100,000 hours
Start-idle times	1–2 minutes	15–30 minutes
Boroscope locations	More than industrial	

Fig 6.1.17 • Aero-derivative vs. industrial facts

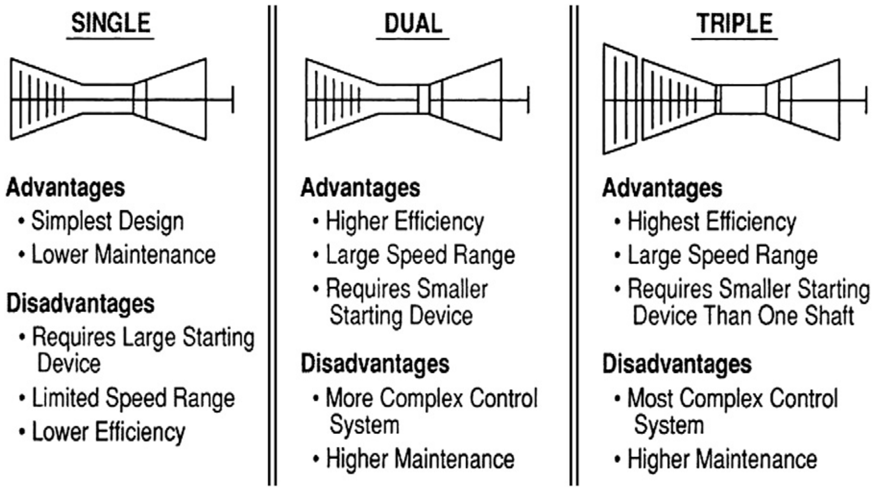
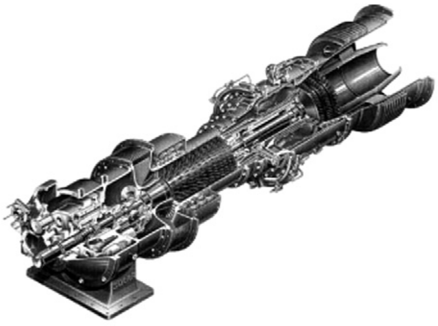
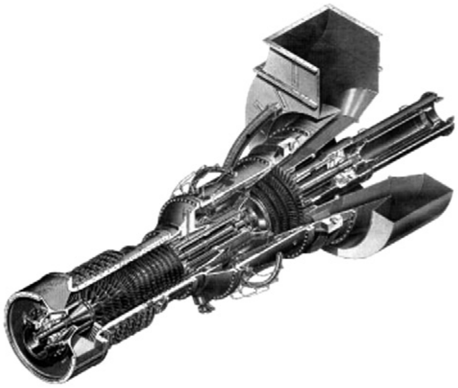


Fig 6.1.18 • The number of gas turbine shafts advantages/disadvantages



■ Single shaft – Gas generator and power turbine on one shaft (only industrial types) (left)



■ Dual shaft – Gas generator and power turbine on individual shaft (industrial and aero types) (right)

Fig 6.1.19 • The number of gas turbine shafts (Courtesy of Solar Turbines, Inc.)

Hot end drive (exhaust end)	Cold end drive (inlet end)
■ Majority of 1st, 2nd generation	■ Some 2nd generation ■ Most 3rd generation
Disadvantages	**Advantages**
■ Longer drive coupling spacer ■ Driver coupling in hot environment	■ Shorter drive coupling spacer ■ Minimized thermal expansion effects

Fig 6.1.20 • Gas turbine drive configurations

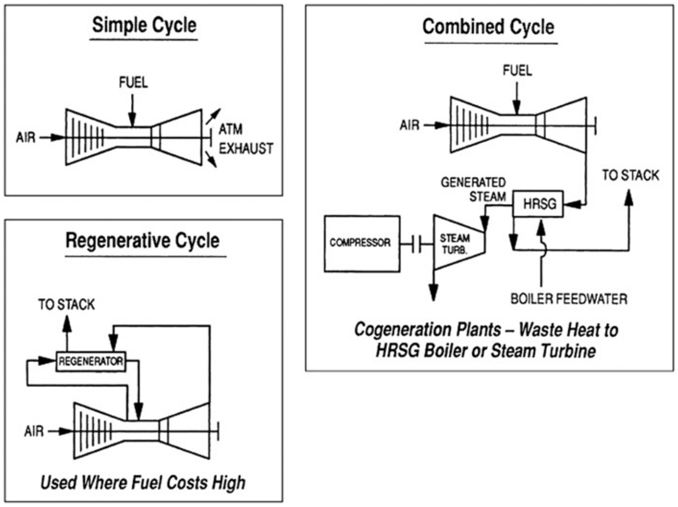


Fig 6.1.21 • Gas turbine cycles

Industrial single shaft	Industrial dual shaft
■ Generator drive ■ *Limited mechanical drive applications (Compressor – pump)	■ Land based mechanical drive ■ Pipeline ■ Refinery, gas plant, petrochemical plant land based ■ Generator drive

*Usually used only when a multi-shaft alternative in the designed power range is not available

Aero multi-shaft (dual or triple)

- Co-generation
- Off-shore
- Generator drive
- Compressor drive
- Pipeline
- Pump drive
- Compressor drive

Fig 6.1.22 • Gas turbine applications

Best Practice 6.2

Accurately define site conditions to be sure that sufficient driver power will be available at site.

Gas turbine site power is determined by elevation, temperature, inlet conditions, outlet conditions, humidity and fuel conditions.

Confirm in the design phase that all site conditions are correct, and confirm ambient temperature conditions for high sites, and an accurate fuel gas analysis. Consult other end users in the area to confirm that the anticipated conditions are correct.

Perform a life cycle cost analysis to determine possible lost revenue costs that would arise from using an undersized driver, in order to justify a larger power driver.

Lessons Learned

Failure to consider actual site conditions and fuel gas composition has resulted in power deficient gas turbines that reduce product revenue for the life of the plant.

Insufficient driver power will restrict maximum possible pump or compressor flow rates and generated power. The associated revenue losses can exceed over $100 MM over the life of the plant.

Benchmarks

This best practice has been used since 1990 to ensure that sufficient gas turbine power is available at site conditions. A combination of rigorous checks of anticipated site conditions, consultation with other end users in the geographic area and a life cycle cost analysis to justify a larger driver selection, if warranted, has been performed to result in maximum driver reliability and product revenue.

B.P. 6.2. Supporting Material

Gas Turbine Performance

A gas turbine is a dynamic internal combustion engine. When we compare the performance of a gas turbine to that of a steam turbine, it becomes immediately evident that steam turbine performance is much easier to calculate, since both the vapor and the vapor conditions are fixed. For a gas turbine, the vapor condition depends on the type of fuel used and the atmospheric conditions. This is because the inlet to the gas turbine engine is from the atmosphere, and any change in temperature, humidity or pressure will affect the mass flow into, and consequently the power produced by the gas turbine. The gas turbine cycle (Brayton) is open.

As a result, steam turbine performance can be expressed rather easily in terms of steam rate (pounds of steam per horsepower or kilowatt hour) and external efficiency. Since the gas turbine vapor conditions are variable however, its performance must be expressed in terms of heat rate, BTUs per horsepower or kilowatt hour, thermal efficiency and fuel rate. All of the above also must be expressed in standard terms.

A set of standardized conditions has been established by ISO (The International Standards Organization) to rate all gas turbines. We will discuss the various ISO standard requirements, and how a site rating is obtained by using vendor ISO derating data for each turbine design. A performance example for an actual gas turbine will be presented, and the effect of varying inlet conditions (temperature, pressure and humidity) on performance will also be explored. Finally, the exhaust gas composition will be discussed, and the emission products examined. In addition, various alternatives for meeting local emission requirements will be presented and discussed.

Figure 6.2.1 presents a comparison between gas turbine and steam turbine performance.

A gas turbine is an internal combustion engine in that the hot vapor is produced internal to the engine. The cycle is open, since both inlet and exhaust conditions are 'open' to the atmosphere and vary with atmospheric conditions.

The steam turbine is an external combustion engine since the hot vapor is produced external to the engine. The steam turbine cycle is closed, in that both inlet and exhaust conditions are controlled by the steam generation system (boiler), therefore steam turbine conditions are constant and do not vary.

Figure 6.2.2 presents performance parameters for steam turbines. Since inlet and exhaust conditions are controlled and the steam turbine is an external combustion engine, steam rate and external efficiency can be used to express performances.

Since the gas turbine Brayton cycle is open, vapor conditions are variable and performance must be expressed as:

- Heat rate
- Thermal efficiency
- Fuel rate

These facts are shown in Figure 6.2.3.

Gas turbine ISO conditions

Since gas turbine performance varies as a function of fuel and inlet conditions, a set of standard conditions has been established by the International Standards Organization to define gas turbine performance. These facts are presented in Figure 6.2.4.

Gas turbine vendors publish their performance data in terms of ISO power rating and ISO heat rate. Typical vendor data is shown in Figure 6.2.5.

Site rating correction factors

Gas turbine site performance is directly affected by inlet air density and air environmental conditions as shown in Figures 6.2.6, 6.2.7 and 6.2.8 respectively.

Since produced power and heat rate vary as a function of inlet temperature, pressure and inlet duct and exhaust duct pressure

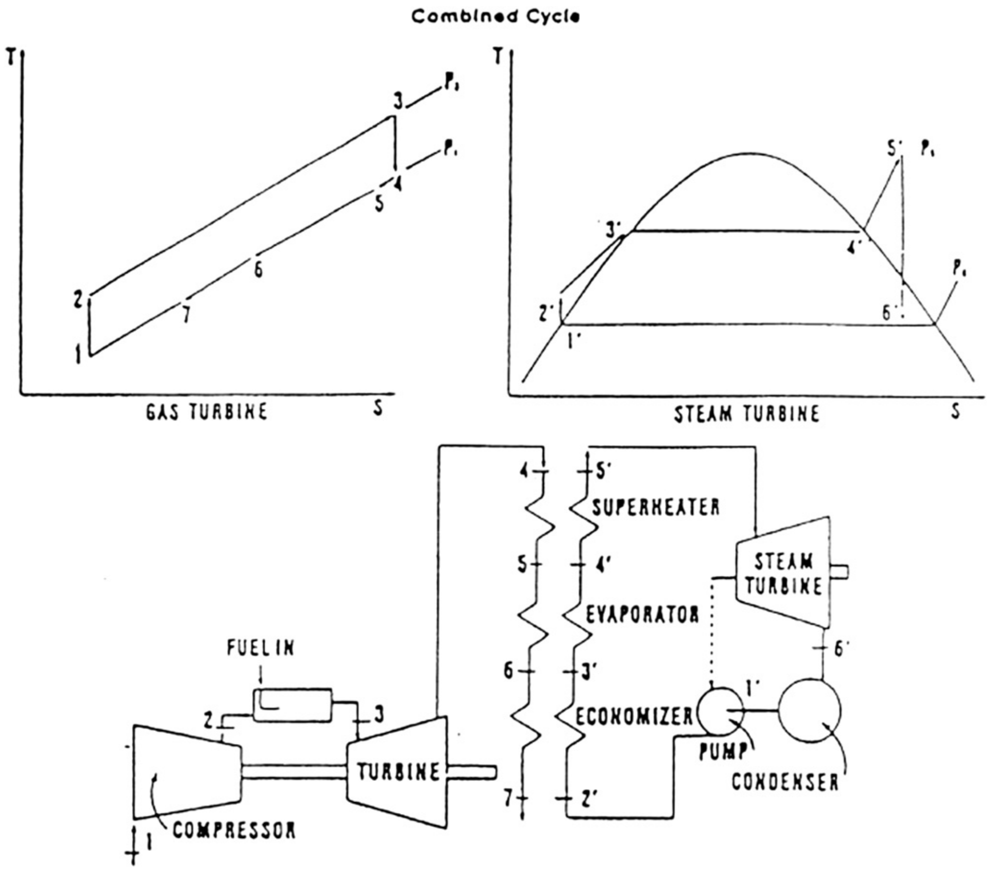


Fig 6.2.1 • Gas turbine vs. steam turbine performance (Reprinted with permission of Gas Producers Suppliers Association GPSA)

- Since the steam turbine (Rankine) cycle is closed, the vapor and vapor conditions are constant!
- Therefore performance can be expressed in terms of:

  $$\text{Steam Rate} = \frac{\text{kg or lbs of steam}}{\text{kW or bhp-hr}}$$

  $$\text{External efficiency} = \frac{\text{actual work} \times \text{loss factor}}{\text{ideal work}}$$

- Mechanical and leakage losses

Fig 6.2.2 • Gas turbine vs. steam performance

- Since the gas turbine (Brayton) cycle is open, both the vapor and vapor conditions are variable.
- Therefore performance is expressed in terms of:
- $$\text{heat rate (ISO)} = \frac{\text{kJ or btu}}{\text{kW or bhp-hr}}$$
- $$\text{thermal efficiency} = \frac{\text{kJ/kW-hr}}{\text{heat rate (ISO) kJ/kW-hr}} \quad \text{or} \quad \frac{\text{btu/hp-hr}}{\text{heat rate (ISO) btu/hp-hr}}$$
- $$\text{fuel rate} = \frac{\text{(heat rate) (kW)}}{\text{fuel heating value (kJ/kg)}^{*}} \quad \text{or} \quad \frac{\text{(heat rate)} \times \text{(horsepower)}}{\text{fuel heating value (btu/lb)}^{\dagger}}$$

Note: *ISO conditions – standardized fuel, inlet conditions at design speed – no losses
† kJ/Nm^3 (normal cubic meter) for gas fuel or BTU/SCF (standard cubic foot) for gas fuel

Fig 6.2.3 • Gas turbine vs. steam performance (continued)

drop, vendors supply correction curves to convert ISO conditions to site conditions.

Figures 6.2.9A to 6.2.9F present an example of a typical gas turbine site rating exercise.

The effect of firing temperature on power and efficiency

A small increase in firing temperature has a significant effect on produced horsepower and on engine efficiency. These facts are shown in Figure 6.2.10.

- Since gas turbine performance varies as a function of fuel and inlet conditions, a set of standardized conditions has been established by ISO (International Standards Organization) to rate all gas turbines.
- ISO standard conditions are:
  - T inlet = 15°C (59°F)
  - P inlet = sea level
  - Inlet and exhaust losses = 0" H_2O
  - Based on stated fuel heating value
  - Relative humidity = 0%
  - Design speed of rotors
  - Power losses = 0
  - Compressor bleed air = 0

Fig 6.2.4 • Gas turbine performance ISO conditions

Dresser-Rand
Turbo Products Division

Olean, New York-USA

MODEL	POWER RATING ISO Base Load Gas Fuel (hp)	HEAT RATE Lower Heating Value (LHV) (Btu/hp-hr	POWER SHAFT SPEED (RPM)	PRESSURE RATIO	NUMBER OF COMBUSTORS	AT ISO RATING CONTINUOUS Turbine Inlet Temp. (°C)	Exhaust Flow (kg/sec)	Exhaust Temp (°C)
DR-22C	5,278	8,850	13,820	9.9	6	1,035	15.6	579
DR-990	5,900	8,350	7,200	12.2	1	1,082	20.0	482
DR-60G	18,750	6,840	7,000	21.5	1	1,216	45.6	482
DR-61G	31,200	6,777	3,600	18.8	1	1,235	69.0	523
DR-61	30,800	6,800	5,500	18.8	1	1,235	69.0	520
DR-63G	56,840	6,135	3,600	30.0	1	1,154	122.5	452

- ALL VENDORS PUBLISH ISO PERFORMANCE AND DE-RATING DATA SO THAT SITE PERFORMANCE (AT ACTUAL SITE CONDITIONS, FUEL AND LOSSES) CAN BE DETERMINED
- TYPICAL VENDOR'S ISO DATA

Fig 6.2.5 • Gas turbine performance ISO and site performance (Reprinted with permission of Turbomachinery International Handbook 1993, Vol. 34, No. 3)

- A given engine design limits air volume flow capacity.
- Produced power is a function of actual energy extracted per pound of vapor and mass flow of vapor.
- For a given engine therefore, produced power varies directly with inlet air density.
- Produced power does become limited by low volume (stall and surge) flow.

Fig 6.2.6 • The effect of inlet air density on produced power and heat rate

Care must be taken when selecting gas turbines to ensure sufficient shaft power is available at:
- High temperature conditions
- Fouled inlet conditions

Gas turbine applications tend to be ‘fully loaded’ since gas turbines (unlike steam turbines) are not custom designed.

Fig 6.2.7 • Gas turbine performance effect of inlet conditions on performance

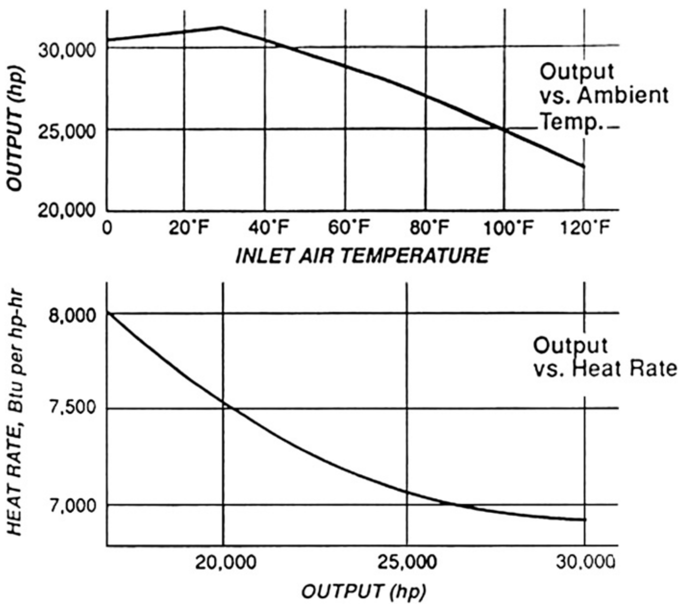


Fig 6.2.8 • Typical gas turbine output power and heat rate vs. ambient temperature

1. Scope
The purpose of this specification is to estimate the site shaft horsepower and heat rate for a given set of site conditions.
2. Applicable documents
Figures 6.2.9D–6.2.9F
3. Requirements
3.1 The following site condition must be known:
A. Elevation (ft.)
B. Inlet temperature (°F)
C. Inlet duct pressure loss (inches of water)
D. Exhaust duct pressure loss (inches of water)
4. Procedure
4.1 Read the shaft horsepower (SHP) and heat rate (HR) for the site inlet temperature (from Figure 6.2.9D).
4.2 Read the elevation correction factor (d) for the site elevation (from Figure 6.2.9E).
4.3 Site shaft horsepower:
A. Read the inlet correction factor (Ki) for the site inlet duct pressure loss (from Figure 6.2.9F).
B. Read the exhaust correction factor (Ke) for the site exhaust duct pressure loss (from Figure 6.2.9F).
C. Calculate the site shaft horsepower:
Site SHP = SHP (from Figure 6.2.9D) · δ · Ki · Ke
4.4 Site heat rate:
A. No elevation correction factor (δ) is used for the heat rate.
B. Read the heat rate correction factor (Kh) from Figure 6.2.9F for the duct pressure loss (sum of site inlet and exhaust duct pressure losses).
C. Calculate the site heat rate:
Site HR = HR (from Figure 6.2.9E) · Kh
5. Sample calculation
5.1 Assume the following site conditions:

A. Elevation (ft.)	1000 ft
B. Inlet temperature (°F)	59°F
C. Inlet duct pressure loss (inches of water)	3.5 inches of water
D. Exhaust duct pressure loss (inches of water)	4.5 inches of water

5.2 Read the shaft horsepower (SHP) and heat rate (HR) with no inlet or exhaust duct pressure losses for the site inlet temperature (from Figure 6.2.9D).

59°F site inlet temperature	
Shaft horsepower (SHP)	29,200
Heat rate (HR)	7,035 BTU/HP-HR

5.3 Read the elevation correction factor (δ) for the site elevation (from Figure 6.2.9E).

1000 ft. site elevation:	
Elevation correction factor (δ)	0.964

5.4 Site shaft horsepower:
A. Read the inlet correction factor (Ki) for the site inlet duct pressure loss (from Figure 6.2.9F).

3.5 inches of water site inlet duct pressure loss:	
Inlet correct factor (Ki)	0.9845

B. Read the exhaust correct factor (Ke) for the site exhaust duct pressure loss (from Figure 6.2.9F).

4.5 inches of water site exhaust duct pressure loss:	0.991
Exhaust correction factor (Ke)	

C. Calculate the site shaft horsepower:
Site SHP = SHP (from Figure 6.2.9D) · δ · Ki · Ke
= 20.200 · 0.964 · 0.9845 · 0.991
Site SHP = 27,463
5.5 Site heat rate:
A. Read the heat rate correction factor (Kh) from Figure 6.2.9F for the duct pressure loss (sum of site inlet and exhaust duct pressure losses).
Duct pressure loss is the sum of 3.5 inches of water site inlet duct pressure loss and 4.5 inches of water site exhaust duct pressure loss, equaling 8.0 inches of water duct pressure loss:

Heat rate correction factor (Kh)	1.016

B. Calculate the site heat rate:
Site HR = HR (from Figure 6.2.9D) · Kh
= 7,035 · 1.016
Site HR = 7,148 BTU/HP-HR

Fig 6.2.9A-C • Typical gas turbine site rating exercise

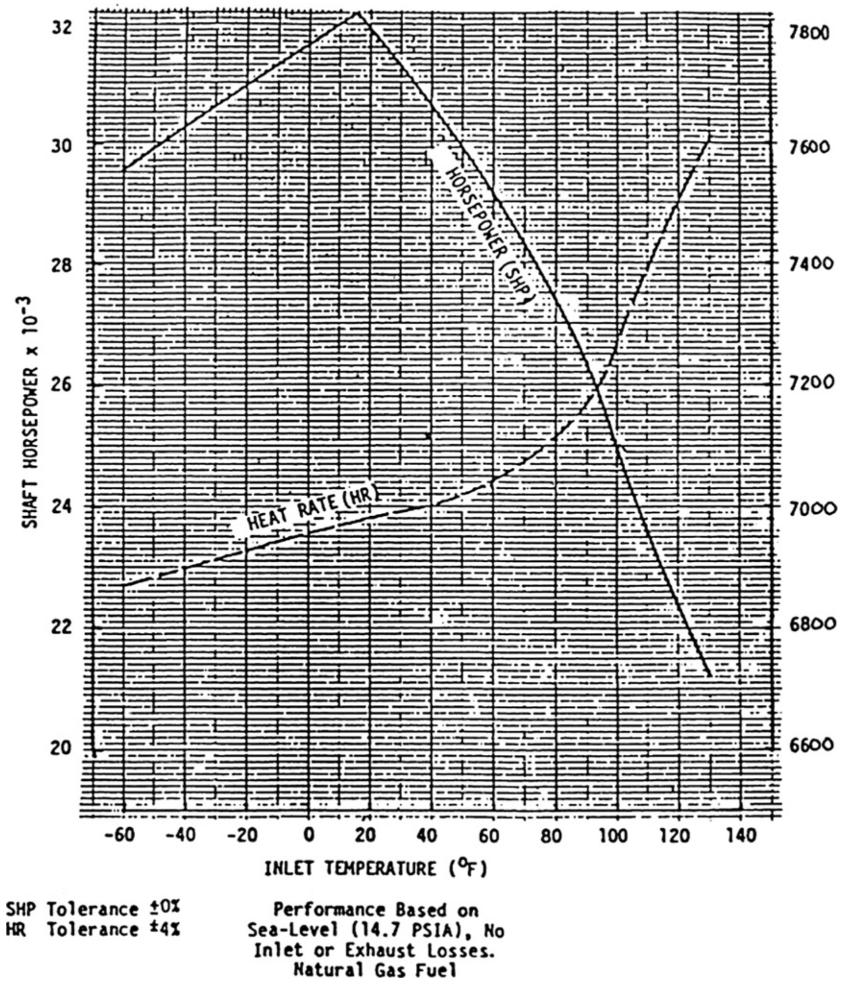


Fig 6.2.9D • Figure for typical gas turbine site rating exercise (Courtesy of General Electric)

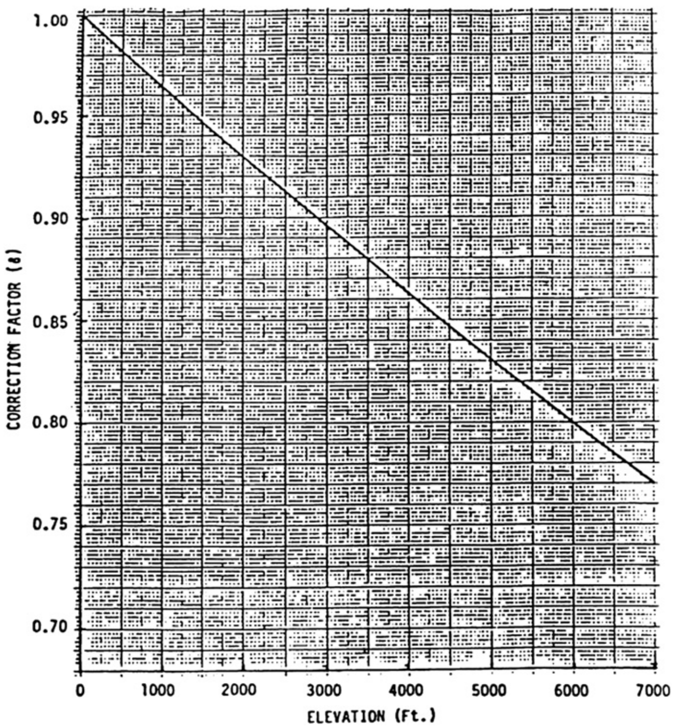


Fig 6.2.9E • Figure for typical gas turbine site rating exercise (Courtesy of General Electric)

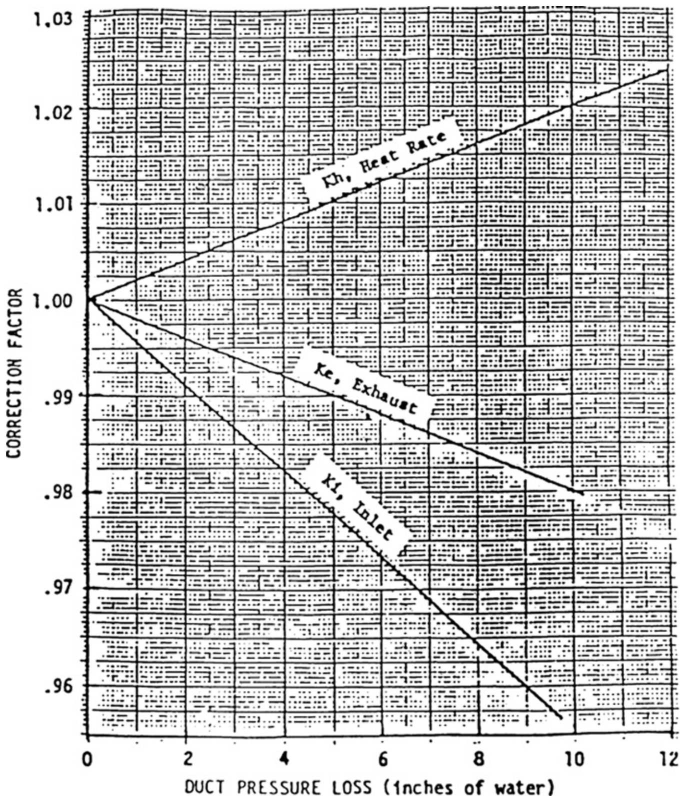


Fig 6.2.9F • Figure for typical gas turbine site rating exercise (Courtesy of General Electric)

- THE HOTTER THE VAPOR (FIRING TEMPERATURE) THE GREATER AMOUNT OF ENERGY EXTRACTED FROM EACH LB OF VAPOR.

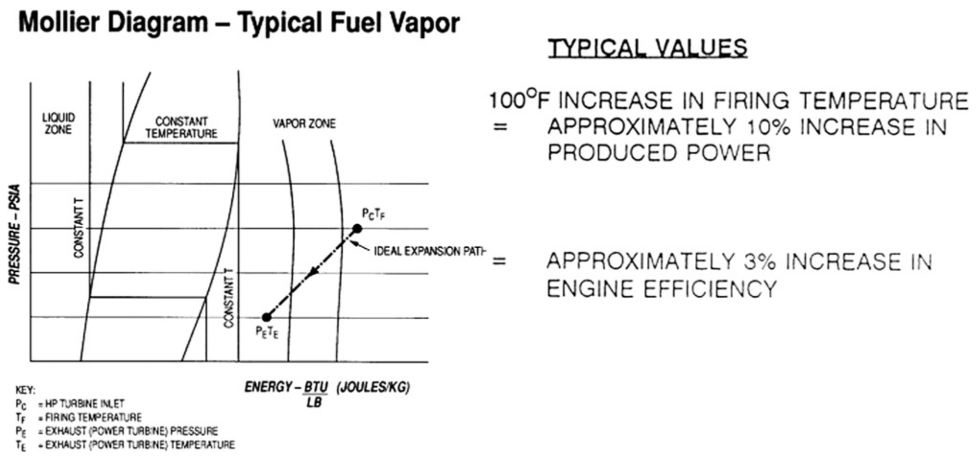


Fig 6.2.10 • The effect of increased firing temperature on produced power and engine efficiency

Best Practice 6.3

Size the gas turbine output power for a minimum of +10% at rated site conditions (see B.P. 6.2 for site condition rating).

Inaccurate site condition specifications, variances in fuel gas composition and fouling of the gas generator can significantly reduce its output power and result in significant revenue losses.

Insufficient output power will directly reduce maximum attainable flow rates, resulting in significant revenue losses for the life of the project.

Perform a life cycle cost analysis, to determine possible lost revenue costs and to justify a larger power driver that is a minimum of +10% above rated power.

Lessons Learned

The majority of mechanical drive gas turbine installations suffer revenue opportunity losses from insufficient site power.

Insufficient driver power will restrict maximum possible flow rates or generated power. The associated revenue losses can exceed over $100 MM for the life of the plant.

Benchmarks

This best practice has been used since 1990 to ensure that sufficient gas turbine power is available at site conditions. Project management resistance will be initially encountered and will usually require a life cycle cost analysis to justify the additional revenue opportunity as compared to the additional capital and installation costs for a larger gas turbine.

B.P. 6.3. Supporting Material

Please refer to material in B.P. 6.2.

Best Practice 6.4

Screen for proven model experience (including experience for all support systems) during the pre-FEED phase of the project to determine the acceptable vendors list.

Do not assume that a proven vendor's gas turbine model will have sufficient field operating experience.

Model generations (higher operating temperatures, new control systems and modified support systems) can and will change.

Require vendor experience and references for review of the exact turbine model quoted and all support systems during the pre-FEED phase of the project.

Lessons Learned

Failure to determine experience of gas turbine models and their support systems has led to significant start-up delays and reliability issues that can last for the life of the operating plant.

Start-up delays can result in significant revenue losses.

In addition, reliability issues resulting from unproven gas turbine components and/or their support systems can add to revenue opportunity losses for the life of the process unit.

These issues can expose the end user to revenue losses in the hundreds of millions of USD for the life of the process unit.

Benchmarks

This is a new best practice, added in 2010 as a result of the selection of a proven gas turbine from a vendor that had recently combined forces with a larger company. While the gas turbine design was maintained during the takeover, the proven control system was replaced by a 'new generation' system that was used for the first time on our project. The result was a start-up delay of three months to correct the problems.

B.P. 6.4. Supporting Material

In this section, we will present and discuss the various gas turbine support systems. The gas turbine, at first glance, appears to be a very complicated piece of rotating equipment. Part of the reason for this perception is due to the complexity and number of the various support systems involved.

The objective of this section will be to classify each type of support system and present its particular function. It is felt that this approach will make the reader realize that the support systems used for a gas turbine are very similar, in most cases, to those used by turbo-compressors and steam turbines.

After each major type of support system is defined and discussed, the accessory gearbox will be presented. The accessory gearbox is a very critical piece of equipment, since it provides power take-offs to the majority of support system pumps, starters and blowers. Accessory gearboxes will be discussed for both industrial and aero-derivative type gas turbines.

Figure 6.4.1 shows a sketch of the location and types of gas turbine support systems along with the definition of a support system.

As can be seen in the figure 6.4.2, the availability of the gas turbine is a direct function of the support systems. Particular attention must be paid to the preventive maintenance (PM) and

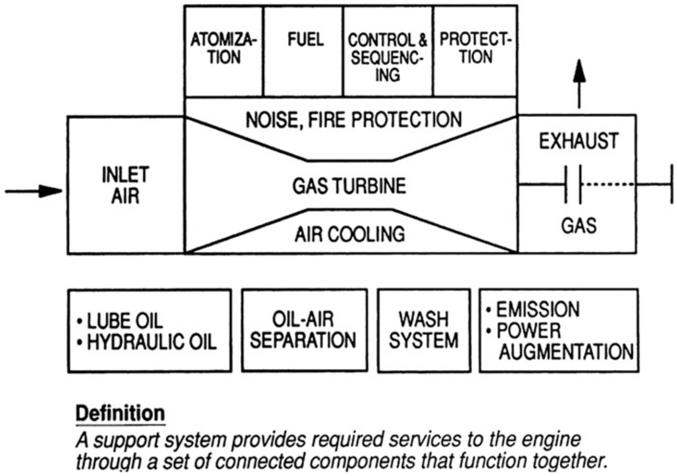


Fig 6.4.1 • Gas turbine support systems

- Most unscheduled engine shutdowns are the result of a support system malfunction.
- The availability (reliability) of an engine is only as high as the support system component availability.
- There are typically 8–10 support systems per engine.
- There are approximately 1,000 support system components in a typical engine.

Fig 6.4.2 • Gas turbine support system facts

Air/gas handling
- Inlet
- Filter
- Silencer
- Duct
- Exhaust
- Duct
- Silencer
- Heat recovery (HRSG)

Lube/control oil
- Engine lube
- Auxiliary lube
- Driven equipment lube
- Engine control actuation
- Guide vane
- Stator vane
- Trip
- Control valve
- Air/oil separation
- Scavenging

Control
- Engine start/stop sequencing
- Condition monitoring
- Governor system
- Protection system
- Fuel systems
- Gas
- Liquid
- Dual
- Atomizing air

Cooling
- Engine internal
- Industrial
- Aero-derivative
- Engine external
- Industrial
- Auxiliary cooling
- Lube oil coolers(etc.)
- Engine cleaning
- Liquid
- Crank
- On-line

Injection
- Emission control
- H_2O
- Steam
- Power augmentation
- H_2O
- Steam
- Engine enclosures
- Noise
- Pressurization
- Fire protection
- Halon
- CO_2

Solid
- On-line (industrial)

Fig 6.4.3 • Types of gas turbine support systems

predictive maintenance (PDM) requirements of all support systems to achieve optimum gas turbine reliability.

The various types of gas turbine support systems are shown in Figure 6.4.3. The actual systems present in a specific gas turbine design are a function of vendor design preferences, customer requirements and local environmental requirements.

Types of support systems by classification

Following are the function definition and details concerning each type of gas turbine support system.

Inlet and exhaust system

The inlet and exhaust systems provide the engine with an acceptable level of inlet air filtration, moisture removal and noise reduction. Figure 6.4.4 shows a typical arrangement for a simple cycle installation.

There are two basic types of gas turbine air filters in current use:

- Pulse air type
- Conventional staged type

Figure 6.4.5 presents the function definition of gas turbine inlet filters and a picture of each type.

Pulse type air filters have gained wide acceptance in regions of excessive dust (desert regions) and in regions of very low temperature conditions. They are highly efficient and can be changed on line.

Regardless of the type of air filter (pulse or conventional); filters are often 'staged' to meet local conditions. Figures 6.4.6 and 6.4.7 contain details concerning this application.

Engine noise abatement

The inherent result of energy input and extraction from a gas along with gas velocities (in excess of 960 kilometers per hour [600 miles per hour]) result in high engine noise levels. A highly sophisticated noise abatement system, along with an engine enclosure, is required to reduce the generated noise to acceptable levels. Figure 6.4.8 describes the function, major components and design features of this system.

Control and protection

The control system is the heart of the gas turbine, and is responsible for safe and reliable start-up, at-speed operation, shutdown, monitoring and protection. Figure 6.4.9 defines the design objectives of the control system.

The major functions of the control system are shown in Figure 6.4.10.

Lube and hydraulic systems

The lubrication and hydraulic systems continuously provide clean, cool lubrication and hydraulic fluid to the components at the proper pressure, temperature and flow. The lubrication

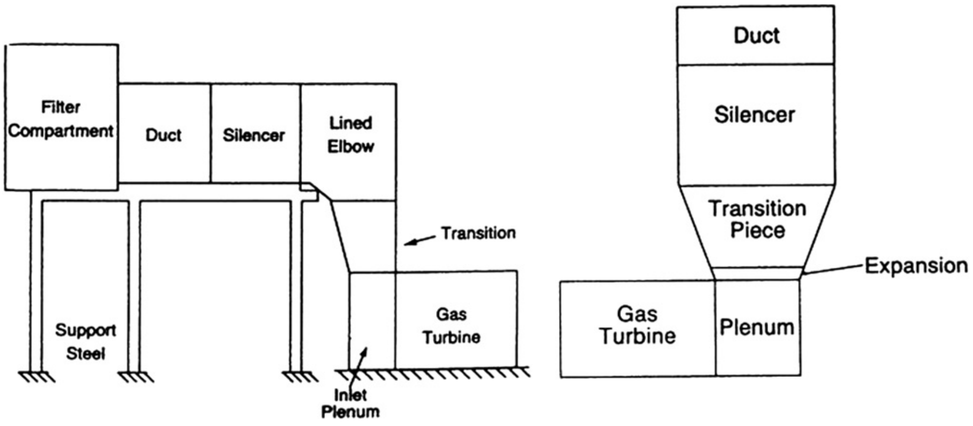


Fig 6.4.4 • Gas turbine inlet (left) and exhaust (right) systems: major components

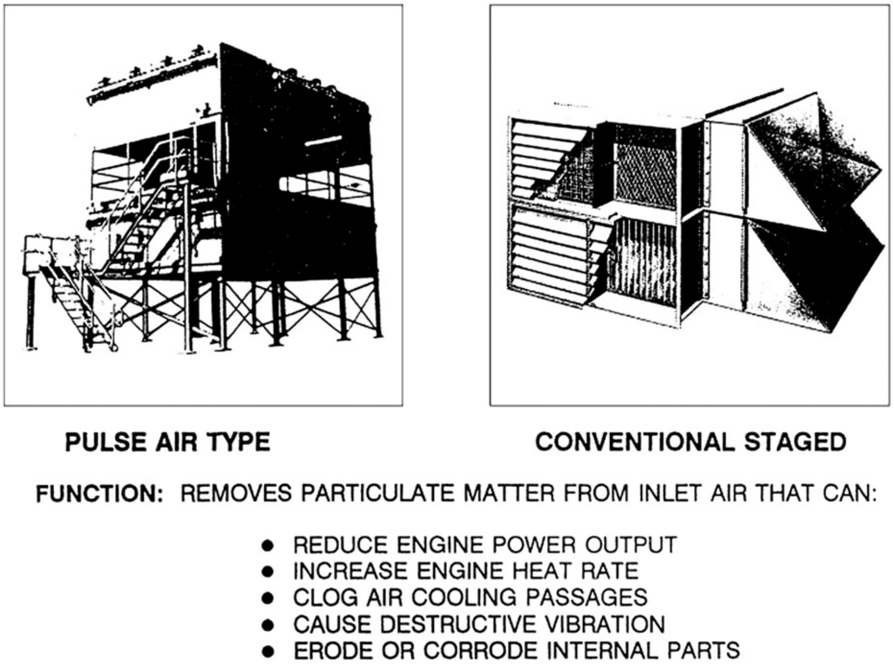


Fig 6.4.5 • Gas turbine inlet air filtration (Courtesy of American Air Filter)

system used for aero-derivative type gas turbines is different from that in an industrial gas turbine, in that a scavenge (vacuum) system is added in the latter, to return lube oil to the sump under flight conditions. This system is retained on mechanical drive applications of gas turbines. Industrial gas turbines use gravity drain methods for returning lube oil to the reservoir. Details concerning aero-derivative gas turbine lube systems are presented in Figure 6.4.11.

Figure 6.4.12 contains the functional definition of the lube and hydraulic (control) system, and shows differences between aero-derivative and industrial systems.

Cooling (engine external, internal and auxiliary systems)

Because of the high temperatures generated within the engine (in excess of 1,100°C [2,000°F]), cooling plays a very important role in engine reliability. Figure 6.4.13 presents the types of cooling required and the components serviced.

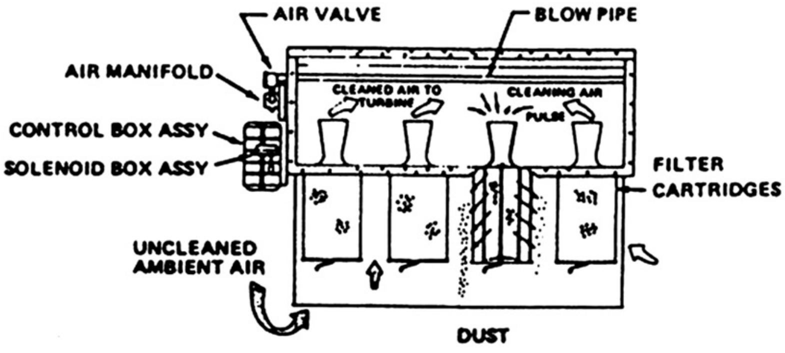
Injection

Injection systems are required for pollution control and/or additional power (power augmentation). Figure 6.4.14 defines the function and features of these systems.

Fire protection

High engine temperatures, and the close proximity of fuel and potential ignition sources within an enclosure, provide a potentially hazardous environment for the engine. As a result, a fire protection system is required in the engine enclosure. Figure 6.4.15 presents the function and facts concerning the system features.

CLEANING ACTION



FEATURES:

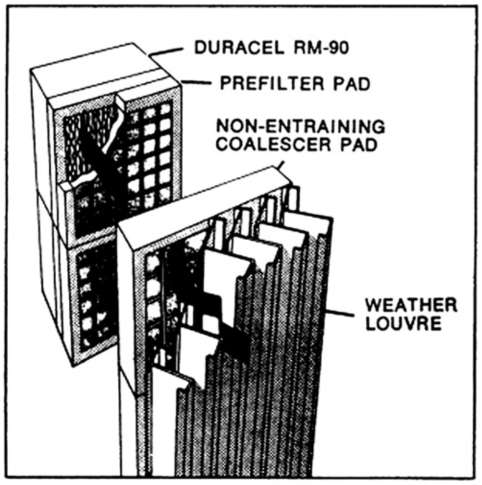
- HIGH EFFICIENCY
- LOW PRESSURE DROP
- AUTOMATED SELF CLEANING
- ELEMENTS CHANGEABLE ON LINE

SINGLE STAGE MULTI-ELEMENT FILTER THAT UTILIZES REVERSE PULSE JET AIR TO CLEAN ON LINE

APPLICATIONS:

- DESERT REGIONS
- PROCESS OR BASE LOAD
- COLD CLIMATES

Fig 6.4.6 ● Pulse air filters



SPE-3 MULTI-STAGE LOW VELOCITY WET AND DRY SALT REMOVAL SYSTEM

TYPICAL FILTERS UTILIZED

- PRE FILTER
- INERTIAL FILTER
- HIGH EFFICIENCY
  - STATIONARY
  - MOVABLE (ROLL "O" MATIC)

INSTALLED ON MOST FIRST GENERATION ENGINES - MULTI-ELEMENT FILTER NOT SELF CLEANING.

APPLICATIONS:

- ALL

Fig 6.4.7 ● Multi-stage filters

Function:	Reduce noise to specified level at specified distances from engine. Typical levels are: ■ 90 DBA overall – 1 meter (3.3 ft) ■ 55–65 DBA overall – 130 meters (400 ft)
Components:	■ Inlet silencer ■ Engine compartment ■ Exhaust silencer
Comments:	■ Rigid design ■ Corrosion proof elements

Fig 6.4.8 • Engine noise abatement

- Like steam turbines, gas turbines have similar control and protection systems
- In fact, third generation control systems are identical for steam and gas turbines
- However, due to high engine temperatures, gas turbine start-up (heat-up) and shutdown (cool down) time must be accurately controlled to prevent engine damage caused by:
  - Rubs
  - Rotor bow

Fig 6.4.9 • Gas turbine control systems

Sequencing system	Control system	Protection
■ Automated ■ Start cycle (starter) ■ Stop cycle	■ Governor ■ Gas generator ■ Power turbine ■ Fuel control system ■ Liquid ■ Atomizing air ■ Gas ■ Dual fuel	■ Overspeed protection ■ Power turbine ■ Gas generator ■ Train parameter protection ■ Lube oil ■ Seal oil ■ Vibration ■ Engine temperature (etc.)

Fig 6.4.10 • Gas turbine control

In addition to the normal components, aero-lube systems utilize:
- Scavenge pumps
- Air/oil separators

These components are required because:
- Anti-friction bearings are used to minimize system flight weight
- Conventional atmospheric drains are not possible

Oil/air mist from bearings must be:
- Scavenged (drawn) back to reservoir
- Separated prior to return to reservoir

Both scavenge pumps and separators are engine driven via the auxiliary gear box

Fig 6.4.11 • Aero-derivative gas turbine lube oil systems

- Function: identical to steam turbine/turbo-compressor – continuously provide cool, clean oil to bearings and control components at the proper pressure, temperature and flow rate.
- However, due to the high temperatures experienced in gas turbines and engine design features, there are differences.

Industrial types	Aero-derivative types
■ Main pump engine driven (thru accessory gear box) ■ Emergency cool down pump (D.C.) ■ Compact design ■ Guarded pipe (supply pipe within drain pipe) ■ Possible use of: Synthetic oil (high flash point)	■ Main pump engine driven (thru accessory gear box) ■ Smaller, very compact design ■ Use of synthetic oil ■ Requirement of: Air/oil separator Scavenge return oil pumps

Fig 6.4.12 • Gas turbine lubrication and control systems

External engine cooling (air)	External engine cooling (H_2O or air)	Auxiliary cooling (H_2O)
■ Combustor ■ H.P. turbine ■ Nozzles ■ Blades ■ Rotor/discs ■ L.P. turbine ■ Nozzles ■ Blades ■ Discs ■ Rotors ■ Bearing housings ■ Struts	■ HP turbine case (jacket) ■ Atomization air	■ Lube oil coolers

Fig 6.4.13 • Gas turbine cooling systems

- Function: Inject H_2O or steam into engine for the purpose of:
- NOx emission reduction (25–42 ppm)
- Increased engine power (power augmentation)

Features:

NOx reduction	Power augmentation
■ Injected directly into combustor ■ Self contained skids ■ Stainless steel pipe ■ High pressure centrifugal (sundyne or equal) for H_2O injection ■ Amount approximately 2% mass	■ Injected down stream of compressor ■ Superheat requirements for steam +30°C (50°F) ■ Maximum 3–5% by mass flow

Considerations:
- Hot path maintenance requirements

Fig 6.4.14 • Gas turbine injection systems

Function: Quickly and effectively extinguish and confine engine fires.

Protection mediums:
- Carbon dioxide
- Halon (limited use in future – impact on ozone layer)

System features:
- Multiple fire detectors
- Medium control system
- Dispenses medium and trips engine on confirmation of fire

Fig 6.4.15 • Gas turbine fire protection systems

Function: To remove air compressor blade deposits and to restore power output and efficiency in a safe and reliable manner.

Options:
- Crank – performed at low (crank) speeds
- H_2O/glycol/detergent
- H_2O/glycol[1]
- On-line – performed at usually idle speed
- H_2O/glycol[2]
- H_2O/glycol/detergent
- Solid particle (walnut shells, catalyst) heavy duty only

Notes:[1] Manufacturer must be consulted regarding acceptable cleaning procedures.

[2]Must be used where operation below approximately 40° is possible.

Fig 6.4.16 • Gas turbine internal component cleaning systems

Internal component cleaning

Available power can be significantly reduced by engine fouling (accumulation of dirt on air compressor blades and stators). Most gas turbines incorporate some type of crank and/or on-line cleaning system. Figure 6.4.16 shows the function and options for these systems.

Figure 6.4.17 contains details concerning the cleaning system.

Accessory gear box

A picture of an accessory gear box used on a Dresser Rand DR 990 gas turbine is shown in Figure 6.4.18. The typical accessory gear box connections for both industrial and aero-derivative gas turbines are shown in Figure 6.4.19.

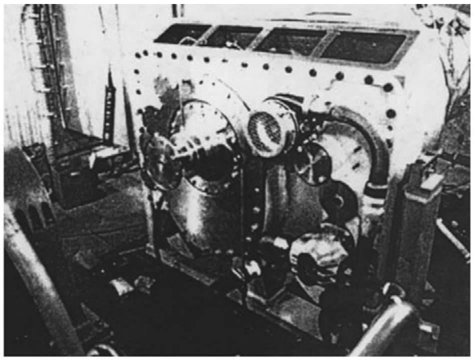
Remember, the engine reliability is a direct function of the reliability of each individual system component! Required PM and an effective PDM program is a must!

On-line cleaning should be performed periodically and frequently because:
- All particles must be removed immediately in a uniform manner to avoid destructive vibration.

Cleaner injection rates and injection periods should be minimized to prevent:
- Erosion
- Corrosion

Cleaner composition must be confirmed to be satisfactory by manufacturer to ensure:
- Material and coating capability
- Acceptable hot path component capability
- Combustors
- Nozzle, blade cooling passages
- Entire cleaning procedure must be reviewed with engine manufacturer.

Fig 6.4.17 • Gas turbine cleaning systems on-line cleaning precautions



FUNCTION: PROVIDE VIBRATION FREE POWER TO ACCESSORIES AT REQUIRED SPEED AND MINIMUM TRANSMISSION LOSSES

Fig 6.4.18 • The gas turbine accessory gear box DR 990 drive gear module (Courtesy of Dresser Rand)

Industrial	Aero-derivative
■ Engine starter	■ Engine starter
■ Main lube oil pump	■ Main lube oil pump
■ Hydraulic (control oil pump)	■ Hydraulic (control oil pump)
■ Atomization air compressor (liquid fuels)	■ Scavenge pump(s)
■ Main fuel pump (liquid fuel)	■ Air/oil separator
■ Cooling water pump (optional)	

Fig 6.4.19 • Type accessory gear box connections

Best Practice 6.5

Use tilting pad radial bearings and not 'lemon or offset sleeve anti-whirl type bearings' to positively eliminate vibration instabilities.

Tilting pad radial bearings provide vibration stability at any load angle.

Lemon bore (elliptical) or offset sleeve (to achieve an elliptical arrangement) bearings do not eliminate vibration instabilities if the load angle lies in the major axis of the ellipse, since the oil film stiffness in this region may not be sufficient to prevent vibration instabilities.

Lessons Learned

Lemon bore or offset sleeve bearings have caused extended FAT periods necessary to modify bearing split line orientation or changes to three or four lobe or tilt pad bearings.

If bearing instabilities are experienced, a possible modification is to rotate the major axis of the ellipse to provide sufficient oil film stiffness at the load angle. This procedure takes time and must be repeated in the field each time the machine is disassembled.

If the above procedure is not successful, installation of multi lobe or tilting pad bearings will be required. This procedure will be time consuming and can delay tests for months.

Benchmarks

This best practice has been used since the early 1980s when offset bearings were required to be changed to multi lobe bearings during the FAT. Delivery delay was three months. It should be noted that the vendor made the modified multi lobe bearing standard on all subsequent turbines. Use of this best practice has resulted in trouble free turbine operation and reliabilities 99.5% and higher.

B.P. 6.5. Supporting Material

Hydrodynamic bearings

Hydrodynamic bearings support the rotor using a liquid wedge formed by the motion of the shaft (see Figure 6.5.1).

Oil enters the bearing at supply pressure values of typically 103-138kPa (15-20 psig). The shaft acts like a pump which increases the support pressure to form a wedge. The pressure of the support liquid (usually mineral oil) is determined by the area of the bearing by the relationship:

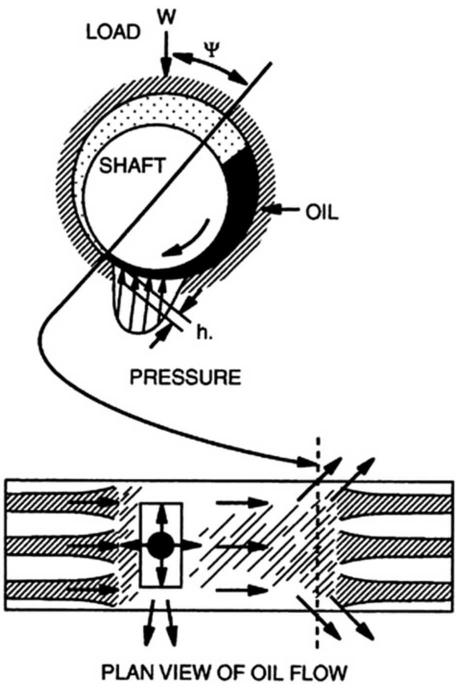


Fig 6.5.1 • Hydrodynamic lubrication (Courtesy of Bently Nevada Corp.)

$$P = \frac{F}{A}$$

Where: P = Wedge support pressure (P.S.I.)
F = Total bearing loads (static and dynamic)
A = Projected bearing area ($A_{PROJECTED}$)
$A_{PROJECTED} = L \times d$
Where: L = Bearing axial length
d = Bearing diameter

As an example, a 4" diameter bearing with an axial length of 2" ($L/d = 0.5$) would have $A_{PROJECTED} = 8\ in^2$.

If the total of static and dynamic forces acting on the bearing are 1600 lbs force, the pressure of the support wedge is:

$$P = \frac{1600\ Lb_{FORCE}}{8\ in^2} = 1,380\ kPa\ (200\ PSI).$$

The maximum desired design wedge pressure for oil is approximately 3,450kPa (500 psi). However, it has been common practice to limit hydrodynamic bearing loads to approximately 1,725kPa (250 psi) in compressor applications. Figure 6.5.2 is a side view of a simple hydrodynamic bearing showing the dynamic load forces.

The primary force is the load, which acts in the vertical direction for horizontal bearings. However, the fluid tangential force can become large at high shaft speeds. The bearing load vector then is the resultant of the load force and fluid tangential force. The fluid radial force opposes the load vector and thus supports the shaft. It has been demonstrated that the average velocity of the oil flow is approximately 47-52% of the shaft velocity. The fluid tangential force is proportional to the journal oil flow velocity. If the fluid tangential force exceeds the load force, the shaft will become unstable and will be moved around the bearing shell. This phenomenon is known as oil whirl.

Hydrodynamic bearing types

Regardless of the type of hydrodynamic bearing, all its surfaces are lined with a soft, surface material made of a composition of

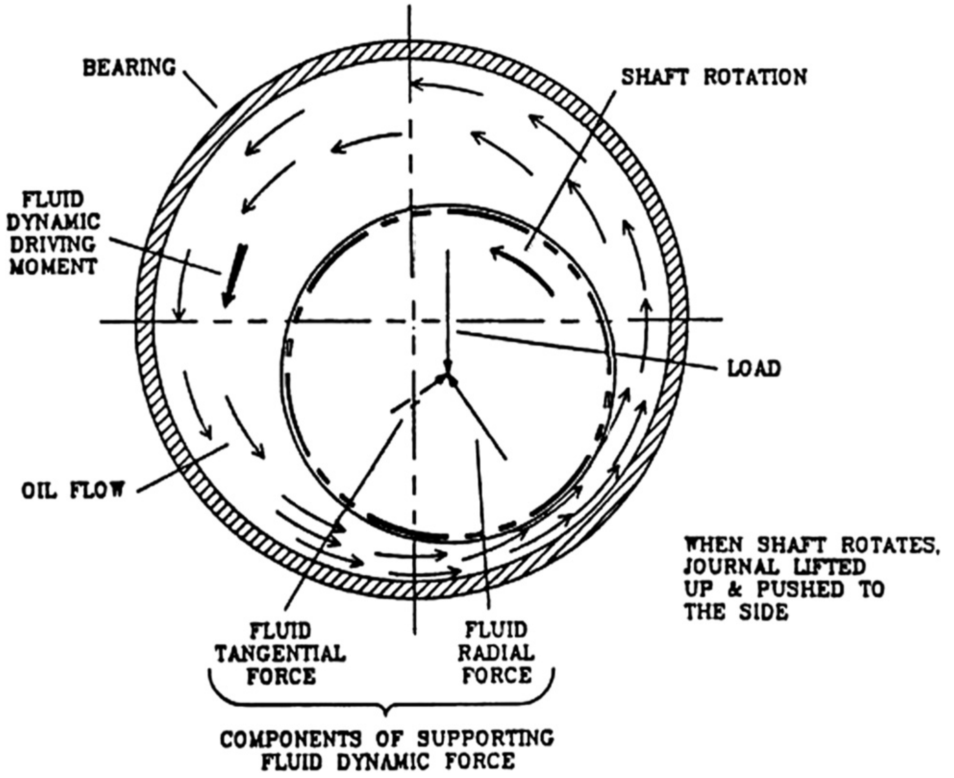


Fig 6.5.2 • Shaft bearing dynamics (Courtesy of Bently Nevada Corp.)

tin and lead. This material is known as Babbitt. Its melting temperature is above 204°C (400°F), but under load will begin to deform at approximately 160°C (320°F). Typical thickness of Babbitt over steel is 1.5mm (0.060"). Bearing embedded temperature probes are a most effective means of measuring bearing load point temperature and are inserted just below the Babbitt surface. RTDs or thermocouples can be used. There are many modifications available to increase the load effectiveness of hydrodynamic bearings. Among the methods available are:

- Copper backed Babbitt or 'Trimetal' – to aid in heat removal
- Back pad cooling – used on tilt pad bearings to remove heat
- Direct cooling – directing cool oil to maximum load points

A typical straight sleeve hydrodynamic journal bearing is shown in Figure 6.5.3.

Straight sleeve bearings are used for low shaft speeds (less than 5,000 rpm) or for older turbo-compressor designs. Frequently, they are modified to incorporate a pressure dam, in the direction of rotation. The pressure dam must be positioned in the top half of the bearing to increase the load vector (see Figure 6.5.2). This action ensures that the tangential force vector will be small relative to the load vector, thus preventing shaft instability. It should be noted that incorrectly assembling the pressure dam in the lower half of the bearing would render this type of bearing unstable. When shaft speed is high, other alternatives to prevent rotor instabilities are noted in Figure 6.5.4.

Shown are examples of anti-whirl bearings. The most common types of these bearings are the three and four lobe design. Elliptical and offset bearing designs do prevent instabilities but tend to increase shaft vibration if the load vector

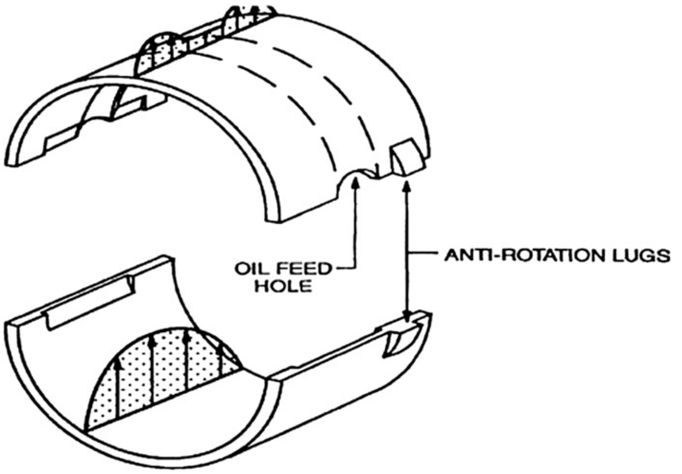


Fig 6.5.3 • Straight sleeve bearing liner (Courtesy of Elliott Co.)

passes through the major axis of the bearing. These types of bearings may have to be rotated in the bearing brackets to prevent this occurrence.

The most common hydrodynamic bearing for higher speed applications is the tilt pad journal bearing shown in Figure 6.5.5.

A tilting pad bearing offers the advantage of increased contact area since the individual pads conform to the shaft orbit. In addition, this type is also a highly effective anti-whirl bearing since the spaces between the pads prevent oil whirl.

Most end users specify tilt pad radial and thrust bearings for turbo-compressor applications.

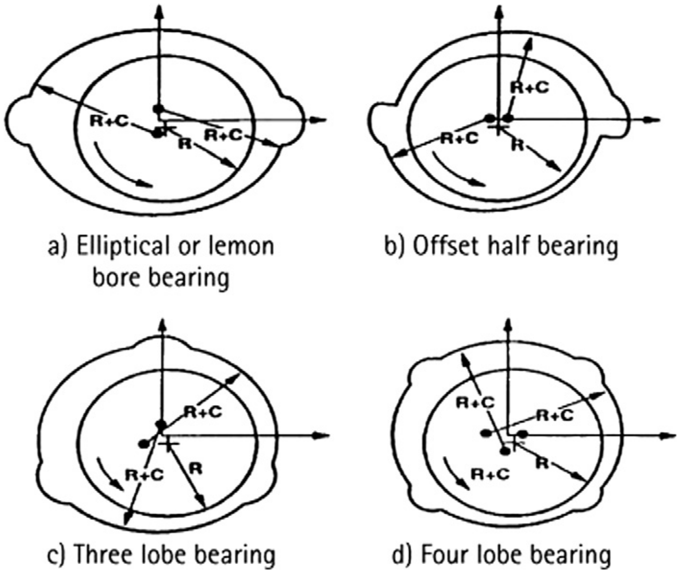


Fig 6.5.4 • Prevention of rotor instabilities

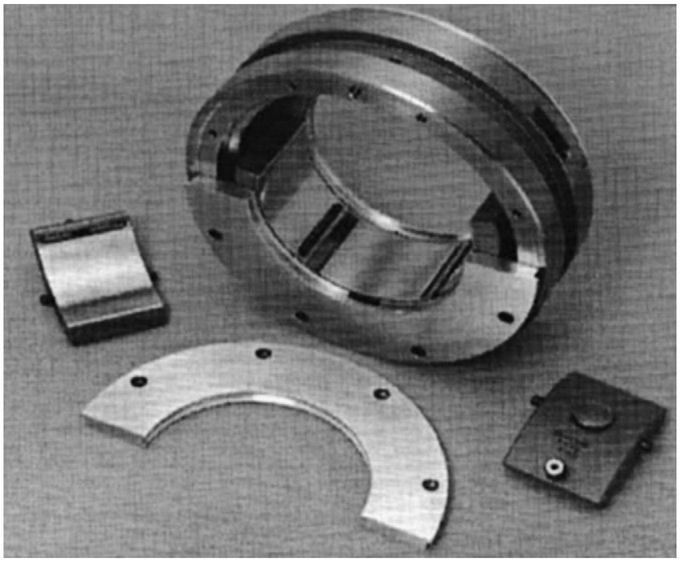


Fig 6.5.5 • Tilting pad journal bearing assembly (Courtesy of Kingsbury, Inc.)

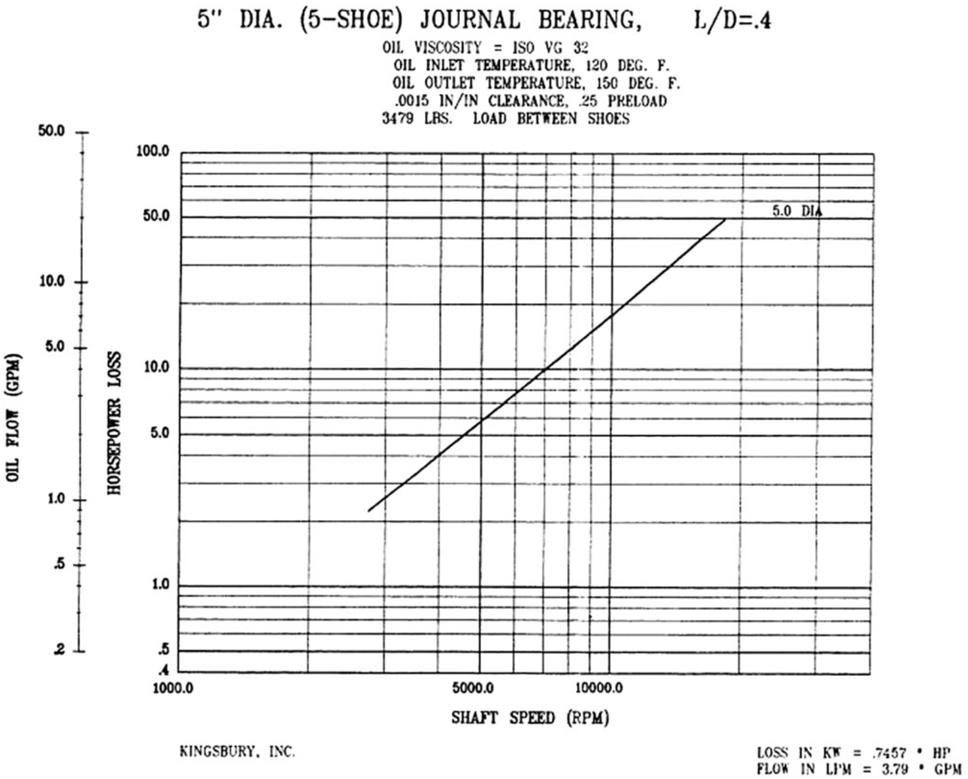


Fig 6.5.6 • Typical journal bearing selection curve (Courtesy of Kingsbury, Inc.)

Figure 6.5.6 shows the mechanical frictional losses and oil flow requirements for a tilt pad journal bearing as a function of shaft speed.

Note that the basis for horsepower loss and oil flow is an oil temperature rise of 30°F. This is the normal design ΔT for all hydrodynamic bearings. Also given in this figure is the data necessary to calculate bearing pressure at the load point.

As an exercise calculate the following for this bearing:

- Projected Area
  $A_{PROJECTED}$ = 5" × 2"
  = 10 square inches
- Pressure
  = 3479 lb force ÷ 10 square inches
  = 347.9 psi on the oil film at load point

Best Practice 6.6

Set up a modular change out agreement for both industrial and aero-derivative gas turbines to minimize downtime and optimize MTTR.

Currently (2010), most gas turbine models have an available modular change-out agreement that will greatly minimize downtime for repair.

Perform a life cycle analysis to compare the cost of the offered agreement to the savings in downtime and corresponding revenue gains from increased operation time.

Aero-derivative complete change outs (gas generator and power turbine) can be accomplished in as little as 36 hours, depending on the location and facilities available.

Hybrid change outs (aero-derivative gas generator and industrial power turbine) can also be accomplished in 36 hours or less, and do not require alignment, since the power turbine rarely requires maintenance, and the gas generator is aero-dynamically coupled (connection duct between gas generator outlet and power turbine inlet).

Industrial change-outs can also be accomplished in much less time than in the past, and most vendors offer a 'modular' approach to gas generator and power turbine change outs.

Lessons Learned

Failure to justify and execute modular change-out agreements have cost end users significant downtime for repair, with corresponding revenue losses during the extended maintenance periods.

Benchmarks

This best practice has been used since 1988, when a modular change-out agreement for an aero-derivative generator set was purchased, which guaranteed a 36 hour complete turbine change-out. The agreement paid off in 1989 when the turbine suffered a complete failure resulting from lack of anti-freeze being added to the water wash system during the winter months.

B.P. 6.6. Supporting Material

See B.P. 6.1 for supporting material.

Best Practice 6.7

Utilize a self-cleaning turbine inlet air filter (pulse type) sized for 1" H_2O clean pressure drop to minimize the effect of fouling on turbine power output.

Review proposed vendor experience and design during the bidding stage of the project to ensure proper filter size and clean filter differential pressure.

Lessons Learned

Failure to select a pulse type inlet air filter of adequate size has resulted in reduced turbine power output, and corresponding loss of production revenue. It has also exposed the turbine to frequent clean or wash cycles, which will eventually require an additional shutdown for maintenance.

Benchmarks

This best practice has been used since 1987, when project requirements for a turbine generator set were defined and included a pulse type inlet air filter sized for 1" clean pressure drop. Since that time, this practice has been followed on all gas turbine projects and has resulted in maximum reliability and has minimized the necessity to clean the turbines.

B.P. 6.7. Supporting Material

Inlet and exhaust system

The inlet and exhaust systems provide the engine with an acceptable level of inlet air filtration, moisture removal and noise reduction. Figure 6.7.1 shows a typical arrangement for a simple cycle installation.

There are two basic types of gas turbine air filters in use today:

- Pulse air type
- Conventional staged type

Figure 6.7.2 presents the function definition of gas turbine inlet filters and a picture of each type.

Pulse type air filters have gained wide acceptance in regions of excessive dust (desert regions) and in regions of very low temperature conditions. They are highly efficient and can be changed on-line. Figure 6.7.3 presents facts concerning this type of filter.

Regardless of the type of air filter (pulse or conventional), filters are often 'staged' to meet local conditions. Figure 6.7.4 contains details concerning this application.

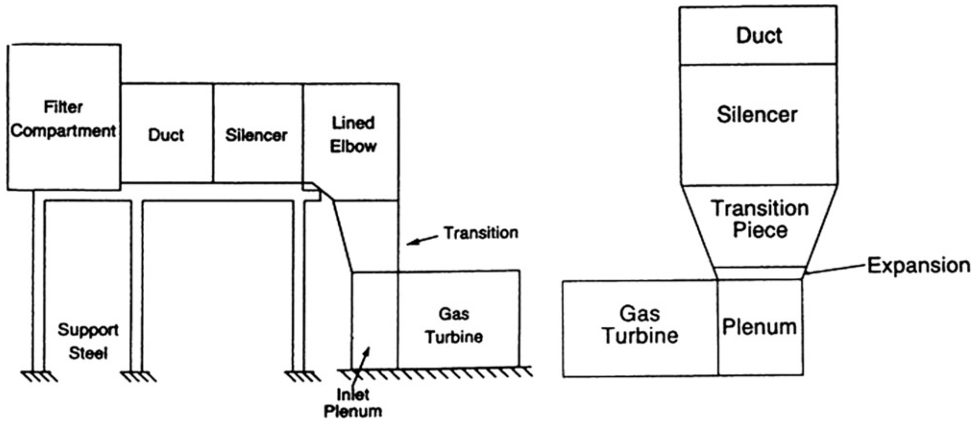
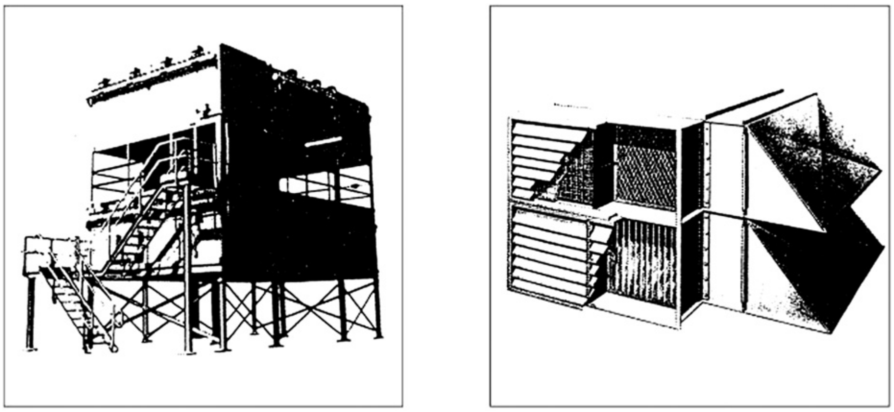


Fig 6.7.1 • Gas turbine inlet (left) and exhaust (right) systems: major components



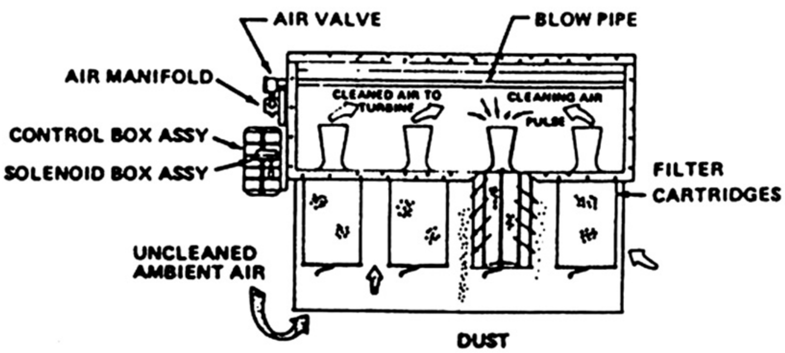
PULSE AIR TYPE   CONVENTIONAL STAGED

FUNCTION: REMOVES PARTICULATE MATTER FROM INLET AIR THAT CAN:

- REDUCE ENGINE POWER OUTPUT
- INCREASE ENGINE HEAT RATE
- CLOG AIR COOLING PASSAGES
- CAUSE DESTRUCTIVE VIBRATION
- ERODE OR CORRODE INTERNAL PARTS

Fig 6.7.2 • Gas turbine inlet air filtration (Courtesy of American Air Filter)

CLEANING ACTION



FEATURES:

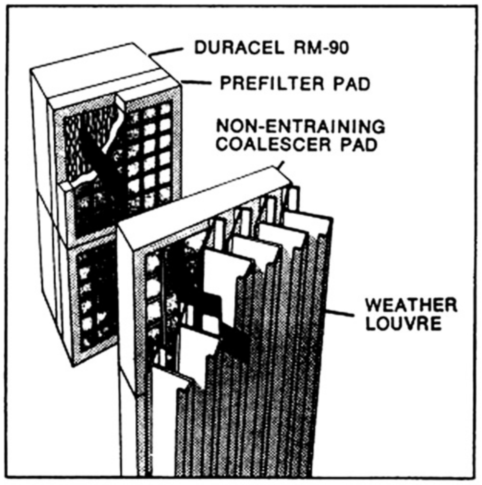
- HIGH EFFICIENCY
- LOW PRESSURE DROP
- AUTOMATED SELF CLEANING
- ELEMENTS CHANGEABLE ON LINE

SINGLE STAGE MULTI-ELEMENT FILTER THAT UTILIZES REVERSE PULSE JET AIR TO CLEAN ON LINE

APPLICATIONS:

- DESERT REGIONS
- PROCESS OR BASE LOAD
- COLD CLIMATES

Fig 6.7.3 • Pulse air filters



SPE-3 MULTI-STAGE LOW VELOCITY WET AND DRY SALT REMOVAL SYSTEM

TYPICAL FILTERS UTILIZED

- PRE FILTER
- INERTIAL FILTER
- HIGH EFFICIENCY
  - STATIONARY
  - MOVABLE (ROLL "O" MATIC)

INSTALLED ON MOST FIRST GENERATION ENGINES - MULTI-ELEMENT FILTER NOT SELF CLEANING.

APPLICATIONS:

- ALL

Fig 6.7.4 • Multi-stage filters

Best Practice 6.8

Require a stainless steel oil reservoir and oil piping for optimum turbine unit reliability.

Using a stainless steel oil system will minimize start-up oil flushing time and ensure a clean, iron-sulfide free oil supply to the turbine for its whole working life.

Most gas turbine vendors currently (2010) offer stainless steel oil reservoirs and piping as an option.

Justify the additional cost during the pre-FEED phase of the project by the additional time for oil flushing (chemical treating) a carbon steel system, and supply case histories describing unscheduled downtimes and maintenance for carbon steel systems.

Lessons Learned

Industrial case histories are full of incidents involving gas turbines that have used carbon steel systems. These have needed extended oil flush times during commissioning, and have experienced bearing failures which have resulted in unscheduled shutdown (usually one week).

Benchmarks

This best practice has been used for all gas turbine oil systems since the late 1980s, resulting in minimum oil flushing times and zero bearing failures due to iron sulfide oil contamination.

B.P. 6.8. Supporting Material

Lube and hydraulic systems

The lubrication and hydraulic systems continuously provide clean, cool lubrication and hydraulic fluid to the components at the proper pressure, temperature and flow. The lubrication system used for aero-derivative type gas turbines is different from that in an industrial gas turbine, in that a scavenge (vacuum) system is added in the latter, to return lube oil to the sump under flight conditions. This system is retained on mechanical drive applications of gas turbines. Industrial gas turbines use gravity drain methods for returning lube oil to the reservoir. Details concerning aero-derivative gas turbine lube systems are presented in Figure 6.8.1.

Figure 6.8.2 contains the function definition of the lube and hydraulic (control) system and differences between aero-derivative and industrial systems.

In addition to the normal components, aero-lube systems utilize:
- Scavenge pumps
- Air/oil separators

These components are required because:
- Anti-friction bearings are used to minimize system flight weight
- Conventional atmospheric drains are not possible

Oil/air mist from bearings must be:
- Scavenged (drawn) back to reservoir
- Separated prior to return to reservoir

Both scavenge pumps and separators are engine driven via the auxiliary gear box

Fig 6.8.1 • Aero-derivative gas turbine lube oil systems

- Function: identical to steam turbine/turbo-compressor – continuously provide cool, clean oil to bearings and control components at the proper pressure, temperature and flow rate.
- However, due to the high temperatures experienced in gas turbines and engine design features, there are differences.

Industrial types	Aero-derivative types
■ Main pump engine driven (thru accessory gear box)	■ Main pump engine driven (thru accessory gear box)
■ Emergency cool down pump (D.C.)	■ Smaller, very compact design
■ Compact design	■ Use of synthetic oil
■ Guarded pipe (supply pipe within drain pipe)	■ Requirement of: Air/oil separator Scavenge return oil pumps
■ Possible use of: Synthetic oil (high flash point)	

Fig 6.8.2 • Gas turbine lubrication and control systems

Best Practice 6.9

Require condition monitoring features of the control system to include facilities for trending of air compressor head and efficiency, gas generator and power turbine efficiency.

Complete predictive maintenance monitoring of gas turbines requires trends of compressor, gas generator and power turbine performance to optimize run times between maintenance cycles.

The end users of 'best of the best' turbines optimize their gas turbine performance monitoring methodology to extend their run times significantly beyond vendor's recommendations (by as much as 60,000 hours).

Review vendor's performance monitoring capabilities during the bid stage of the project and require complete performance monitoring capabilities.

Lessons Learned

Failure to trend compressor, gas generator and power turbine performance results in the practice of completely following vendor maintenance cycles without the benefit of extending these cycle times by the use of effective predictive maintenance techniques.

Vendors now provide diagnostic services that will evaluate turbine condition and recommend when maintenance is to be performed.

Benchmarks

This practice has been used since the 1990s, in which trending of air compressor performance and engine efficiency is highly recommended, when not provided by the vendor. Note: Compressor discharge pressure and temperature must be available for air compressor performance trending.

B.P. 6.9. Supporting Material

FAI has developed and used Excel spreadsheets that receive, calculate and trend gas turbine performance (air compressor, gas generator turbine and power turbine) as well as mechanical and support system condition. We have included this material for reference.

Table 6.9.1 Spreadsheet for trending gas turbine performance

	Client:						
	Equip #:						
Date							
Compressor Performance							
P atmosphere (bara)							
T ambient (°C)							
P Disch. (bara)							
T Disch. (°C)							
Speed (RPM)							
K							
(K-1)/K	–	–	–	–	–	–	–
(n-1)/n	–	–	–	–	–	–	–
Polytropic Efficiency	–	–	–	–	–	–	–
What Action is Required Based on % Difference from Design Efficiency?							

Table 6.9.1 Spreadsheet for trending gas turbine performance—Cont'd

Gas Turbine Performance							
Fuel LHV (KJ/Kg)							
Flow (Kg/sec)							
Output KW							
Site Heat Rate (KJ/KW-hr)	–	–	–	–	–	–	–
GT Efficiency	–	–	–	–	–	–	–
What Action is Required Based on Gas Turbine Performance info. Above?							
Bearing Condition							
# 1 Journal Brg. Vib X							
# 1 Journal Brg. Vib Y							
Major Frequency Observed							
Radial Shaft Position							
# 1 Brg. Pad Temp. 1 (°C)							
# 1 Brg. Pad Temp. 2 (°C)							
# 1 Brg. Drain Temp. (°C)							
Thrust Brg. Displacement							
Direction of Thrust							
Pad Temp. Inlet End 1 (°C)							
Pad Temp. Inlet End 2 (°C)							
Pad Temp. Exhaust End 1 (°C)							
Pad Temp. Exhaust End 2 (°C)							
Thrust Brg. Drain Temp. (°C)							
# 2 Journal Brg. Vib X							
# 2 Journal Brg. Vib Y							
Major Frequency Observed							
Radial Shaft Position							
# 2 Brg. Pad Temp. 1 (°C)							
# 2 Brg. Pad Temp. 2 (°C)							
# 2 Brg. Drain Temp. (°C)							
Brg. Inlet Oil Pressure (bara)							
Brg. Inlet Oil Temperature (°C)							
Viscosity (cst)							
% Water in Oil							
Lube Oil Flashpoint (°C)							

(Continued)

Table 6.9.1 Spreadsheet for trending gas turbine performance—Cont'd

What Action is Required Based on Gas Turbine Bearing info. Above?						
Air Filtration Sys. Condition						
Filter DP (bard)						
Action of Self Cleaning Air System (Active or Inactive)?						
Action?						
Variable Inlet Guide Vanes						
Hyd. Supply Differential Press. (bard)						
Guide Vane Position						
Air Flow Set Point						
Measured Air Flow						
IGV Exhaust Temp. Reference						
Measured Exhaust Temp. (°C)						
Action?						
Cooling and Sealing Air Sys.						
Cooling Air Pressure (bara)						
Other Cooling Sys. Observations						
Air Sealing System Supply Pressure (bara)						
Fuel Gas System						
Ambient Temp. (°C)	–	–	–	–	–	–
Relative Humidity						
Fuel Supply Pressure (bara)						
Fuel Flow Rate (kg/hr)						
Fuel Pressure Upstream of Shut Off Valve (bara)						
Fuel Shut Off Valve Position						
Fuel Pressure Upstream of Fuel Control Valve (bara)						
Fuel Control Valve Position						
Fuel Supply Pressure to Combustors (bara)						
Action?						

Table 6.9.1 Spreadsheet for trending gas turbine performance—Cont'd

Combustion Monitoring System						
Ambient Temp. (°C)	–	–	–	–	–	–
Relative Humidity						
Exhaust Temp. 1 (°C)						
Exhaust Temp. 2 (°C)						
Exhaust Temp. 3 (°C)						
Exhaust Temp. 4 (°C)						
Exhaust Temp. 5 (°C)						
Exhaust Temp. 6 (°C)						
Exhaust Temp. 7 (°C)						
Exhaust Temp. 8 (°C)						
Exhaust Temp. 9 (°C)						
Exhaust Temp. 10 (°C)						
Exhaust Temp. 11 (°C)						
Exhaust Temp. 12 (°C)						
Allowable Spread (°C)						
Top Spread 1 (°C)						
Top Spread 2 (°C)						
Top Spread 3 (°C)						
Exhaust Thermocouple Alarm on?						
Combustion Alarm on?						
High Exhaust Temp. Spread Trip?						
Monitor Enable Activated?						
Action?						
Exhaust Temperature Control System						
Compressor Disch. P (bara)						
Fuel Stroke Reference						
Ambient Temperature (°C)						
Relative Humidity						
Calculated Firing Temp. (°C)						
Action?						

Best Practice 6.10

Require the installation of a torquemeter to allow accurate calculation of gas turbine efficiency.

Calculation of gas turbine efficiency requires the calculation of the driven equipment power.

This calculation can be inaccurate, due to the many variables involved in determining the driven equipment power.

Requiring a torquemeter during the project phase will ensure accurate gas turbine efficiency calculations.

Torquemeters can be installed in existing trains but will require:

- Rotor response study
- Torsional study
- Coupling and coupling guard modifications

Lessons Learned

Gas turbine efficiency calculations will be erroneous and inconclusive in determining maintenance cycles when torquemeters are not installed.

Benchmarks

The installation of torquemeters has been recommended as part of project specifications since 1995. This action has resulted in the accurate determination of critical (un-spared) gas turbine maintenance cycles and significant extension of these cycles.

B.P. 6.10 Supporting Material

Please refer to material in B.P: 6.2.

7

Lube, Seal and Control Oil System Best Practices

It has been my experience that 80% of the root causes of machinery failure lie in the process system and/or the machinery supporting systems. Considering that any unscheduled shutdown for critical (un-spared) equipment can result in revenue loss of millions of USD, ensuring that best practices are used in all aspects of oil system specification, design, test, operation and condition monitoring is essential to maintaining optimum machinery train reliability. This chapter will therefore present the best practices for lube, control and oil seal systems.

Best Practice 7.1

Specify all ambient conditions and desired oil type on the oil system data sheet to ensure proper component selection.

Specifying the incorrect ambient temperature limits will affect oil viscosity and therefore can impact the following components:

- Reservoir heater sizing (lower than specified temperature)
- Rotary pump wear (low viscosity – higher than specified temperature)
- Driver power overload (high viscosity – lower than specified temperature)
- Cooler heat load (higher than specified temperature)
- Filter pressure drop (lower than specified temperature)

The use of a different viscosity grade oil than specified on the data sheet will also affect all of the components mentioned above in the manner noted.

Lessons Learned

Different ambient conditions and oil grades than specified on the data sheet and used in component selection will affect the reliability of the oil system and impact the machinery serviced by the system.

Case histories are full of incidents where oil systems that are not designed for the actual ambient conditions, or the oil type in use, have caused unscheduled shutdowns and loss of large amounts of revenue.

Benchmarks

This best practice has been used since the mid-1970s, when an oil system design audit approach was formulated that has since been used for all new oil systems and 'bad actor' oil systems that had caused more than one shutdown per year. The use of this best practice has saved countless field shutdowns and maximized process unit revenue.

B.P. 7.1. Supporting Material

Critical equipment vendor data

This data must be furnished by each critical equipment vendor, and must contain information as shown in Figure 7.1.1. It is important to note that different vendors furnish different pieces of critical equipment in the same unit. In this case, all vendors should agree to a common lube oil type and common value of oil supply conditions, if possible. Failure to do so only complicates system design, and requires additional components which can reduce system reliability.

- Oil flow rate for each bearing or component
- Bearing or component friction loss (heat load – kJ/hr [BTU/hr])
- Required lube oil type
- Required oil supply pressure and temperature ranges (minimum and maximum) to each bearing or component
- Equipment coast down time
- Any special requirements (equipment cool off time, etc.)

Fig 7.1.1 • Critical equipment vendor data

Site conditions

This information is required for the proper design of the system and should be accurately stated. As a minimum, the data noted in Figure 7.1.2 should be included. Frequently, this information is not known until well into the project (if at all), and leads to cost adders, delivery delays and unreliable systems. End user input in the pre-purchase order phase of the project will eliminate these problems. In addition, determination of auxiliary system arrangements and module location at this time will usually result in simpler, more practical designs that can increase system reliability. A typical auxiliary system vendor data sheet is included at the end of this chapter. It has been completed to include both equipment vendor site data and end user required data for the present example.

- Site environmental conditions
- All utility data
- Location of system modules (consoles) relative to critical equipment – distance and elevation
- Area electrical classification
- Information or sketch detailing system arrangement (location of oil supply and drain connections, component location on modules, required space for maintenance and minimum size of modules)

Fig 7.1.2 • Site condition data

Best Practice 7.2

Require that a stainless steel reservoir, vessels and piping be used to ensure minimum oil flushing time, optimum machinery component life and machinery reliability.

Oil flushing time can significantly extend commissioning and turnaround time, if any oil system components or piping are subject to corrosion.

Lube and seal oil overhead tanks that are not stainless steel will reduce bearing, oil seal and or driver control and protection MTBF, since there cannot be a filter between these tanks and these components.

Replacement of existing non-stainless oil system piping, components or the entire system can usually be justified in un-spared critical machinery trains, if the train has experienced one or more shutdowns, which have required oil flushing prior to restart.

Lessons Learned

Critical (un-spared) equipment trains that do not use all stainless steel piping and components have suffered un-scheduled shutdowns that resulted in long periods of outage for oil flushing, and substantial loss of revenue.

Systems that use overhead tanks that are either not stainless steel, or have carbon steel components, have reduced the machine component MTBF due to iron sulfide building up in the small clearances of the machinery components, which has resulted in premature failure.

Benchmarks

This best practice has been used for projects and retrofits since 1990, producing oil unit trains of the highest reliability and serviced critical machinery.

This best practice has optimized centrifugal compressor train reliability (above 99.7%) and machinery component MTBFs (greater than 100 months).

B.P. 7.2. Supporting Material

Reservoir, vessel, piping and component material preferences

Any reservoir, overhead tank design or material preference should be stated. It is recommended that reservoirs and overhead tanks be constructed of Austenitic stainless steel, to ensure minimal penetration of excessive debris into the auxiliary system. In addition, any preference for piping and synthetic materials should be stated. Recent practice has been to require stainless steel piping, as well as reservoir and overhead tank material, while carbon steel slip-on flanges have been acceptable for lube oil service piping. Experience has shown that in systems containing water, such as water seal systems, stainless steel flanges as well as pipes are required, since considerable amount of rust scale emanates from the flange pipe interface area below the flange gaskets. The subject of the material of the main components (filters, coolers, valves, etc.), is a purchaser's preference. When one considers the potential damage resulting from excessive debris in a system, the additional cost for non-corrosive components can usually be justified. This issue should be thoroughly investigated prior to auxiliary system purchase.

Best Practice 7.3

Specify a minimum of one meter distance between all oil system components to allow operator access, and to optimize system reliability.

Access to all major components is essential to allow operator intervention to switch over to spared system components without the risk of unscheduled trips of critical equipment (un-spared).

Lessons Learned

Consoles that are crowded and do not allow easy access are often ignored by operators and not fully understood in terms of system function.

I have experienced critical unit shutdowns of a steam turbine driven pump, when the local trip lever was accidentally hit due to limited space being available when the personnel were climbing on the console in the course of normal maintenance activities.

Benchmarks

This best practice has been used since my design days for a major machinery vendor (late 1960s) to ensure optimum oil system and corresponding critical machinery train reliability. This best practice approach continues from that time to ensure + 99.7% reliability on all critical equipment trains.

B.P. 7.3. Supporting Material

Console layout and component arrangement

Having confirmed an acceptable component sizing and selection, the console layout and arrangement of components must be reviewed. Methods of review incorporate either a review of outline drawings of the proposed arrangement, or a model review, or a CAD 3D drawing review. Many vendors and users have found that models or CAD 3D drawings aid greatly in the understanding and reviewing of maintenance accessibility and layout considerations.

Console construction

Auxiliary equipment consoles or modules house most of the components present in the auxiliary systems. Their construction should be reviewed to ensure proper stiffness and facilities for installation on site. Many horizontal consoles are constructed in a flexible manner that can result in bending or excessive pipe strains being introduced into during shipment or at installation. It is suggested that full length cross members be positioned as a minimum under pumps, coolers and filters on the equipment baseplate (see Figure 7.3.1). If the baseplate is to be grouted in the field, grout and vent holes should be specified, and reviewed for accessibility to pore grout when equipment is installed on the baseplate.

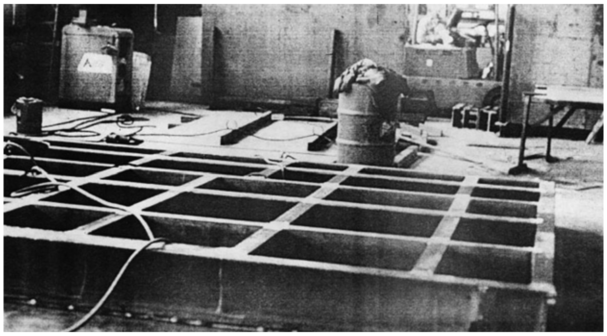


Fig 7.3.1 • Console baseplate construction (Courtesy of Fluid Systems)

Maintenance accessibility

Since equipment must be maintained and calibrated while the auxiliary system is in operation, it is important to provide ample personnel space such that this equipment can be maintained safely and reliably without damage to surrounding components. A rule of thumb is to provide approximately one meter of space around components for accessibility. Note that this is with the utility lines installed. The review of equipment on a model, CAD 3D drawing or an outline should be made considering installation of all utility lines that will be installed in the field.

On-line testing and calibration accessibility

Considering that many components (pumps, drivers, coolers, filters, control valves, instrumentation) will be tested and calibrated with equipment in operation, accessibility for this operation must be considered.

In addition to reviewing the vendor manufactured skids, the placement of all skids in the field must be reviewed for accessibility. Consideration of the skid arrangement only to be complicated by installation against a column or wall in the field will not obtain the objectives of total accessibility.

Utility supply arrangement

Care should be given to the routing of all utility (conduits, steam lines, water lines) supply lines in order to maximize accessibility to the critical equipment auxiliary systems.

Considerations for component disassembly

All components must be able to be dissembled quickly, easily and safely while the unit is operating in the field. To meet this requirement, sufficient space around the auxiliary console must be available for such exercises as cooler bundle removal, filter cartridge removal and auxiliary or main driver removal. In addition, consoles are frequently installed in congested areas, and lifting arrangements should be reviewed beforehand to confirm that components can be removed in a safe and easy manner.

Best Practice 7.4

Fill oil console baseplates with cement for mass after checking for pipe stress and soft foot on pumps and drivers, to ensure maximum pump, driver and instrument reliability.

Horizontal oil console baseplates (the oil console design preference) do not completely support all components, and do not prevent support member vibration, which affects component and instrument reliability.

Filling the console support frame completely with concrete will provide a firm, vibration-free support, and will optimize the MTBF of all components and instruments. Ensure that all piping alignment and soft foot checks, and any corrective action to them, are complete before filling the console baseplate.

If the grout holes provided for perimeter grouting of the baseplate are not sufficient for filling all compartments, burn or cut additional holes in the cover plate for access to each compartment of the console baseplate.

Lessons Learned

Horizontal oil console support member vibration, when the baseplate has not been filled with cement, will affect pump, driver, other component and instrument reliability and can cause unscheduled shutdowns of critical equipment.

Low pump bearing, driver bearing and pump seal MTBFs can result from baseplate vibration. In addition, critical equipment console trips have been caused by auxiliary pump start switch damage, resulting from baseplate vibration, that prevented the auxiliary pump from starting when required.

Benchmarks

This best practice has been used since the early 1980s, when I was involved with a large petrochemical construction project. Its use has resulted in trouble-free oil console start-up and operation and oil systems of the highest reliability.

B.P. 7.4. Supporting Material

Console construction

Auxiliary equipment consoles or modules house most of the components present in the auxiliary systems. Their construction should be reviewed to ensure proper stiffness and facilities for installation on site. Many horizontal consoles are constructed in a flexible manner that can result in bending or excessive pipe strains introduced into components during shipment and at installation. It is suggested that full length cross members be positioned as a minimum under pumps, coolers and filters on the equipment baseplate (see Figure 7.4.1).

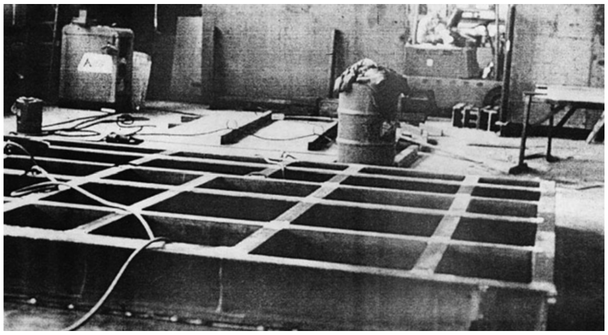


Fig 7.4.1 • Console baseplate construction (Courtesy of Fluid Systems)

If the baseplate is to be grouted in the field, grout and vent holes should be specified and reviewed for accessibility to pore grout when equipment is installed on the baseplate. In addition, it is strongly recommended that after perimeter and support grouting, the entire baseplate be filled with cement for added support rigidity.

Best Practice 7.5

Always perform a design audit of system and component sizing early in the engineering phase for 'bad actor' consoles (more than one shutdown per year), to ensure the highest possible level of critical machinery reliability.

Two factors which play a great part in attaining a reliable auxiliary system are communication and revenue. Frequently, auxiliary systems are sub-contracted by the original equipment manufacturer to a specialty auxiliary system facility. The communication between the original equipment manufacturer (OEM) and the sub-vendor is not always efficient. After receipt of an order to manufacture an auxiliary system, confirmation of scope of supply should be obtained with the end manufacturer of the system involved.

The other factor is the cost of the critical equipment and the price charged for it. In the real world of competition, original equipment vendors often sell the equipment for less than originally anticipated. Therefore, costs are high and profit decreases. In an effort to minimize cost, sub-systems can suffer in terms of design and quality. OEMs will competitively select sub-vendors for auxiliary systems. Care must be taken to ensure the selected sub-vendor is an experienced, quality shop.

Lessons Learned

Failure to design audit new and 'bad actor' oil system and component design has caused many start-up delays and trips of critical (un-spared) compressor trains.

Even at the present time (2010), a design audit of oil systems is not common. Failure to conduct an audit can result in shop testing, start-up delays and troublesome low reliability oil consoles.

Benchmarks

This best practice has been used since the early 1980s to ensure optimum oil console design and component selection, and has resulted in critical machine reliability exceeding 99.7%.

B.P. 7.5. Supporting Material

Design audit agenda

In this section, we will deal with specific areas important to the confirmation of good auxiliary system design and manufacture. To ensure maximum effectiveness of these reviews, it is recommended that a prior agenda, mutually agreed upon between OEM and user, be generated and supplied to both parties well in advance of any meetings. In addition to detailing the subjects of discussion, the agenda should also define the attendees of the meeting. A well defined meeting is still ineffective if the participants are not familiar with the subject or have a minimum amount of experience.

Confirmation of scope

For the reasons mentioned above, scope review approximately one to two months after auxiliary system order placement is recommended. The major areas of scope review are:

- Schematic review
- Data sheet review
- Exceptions to specification

Schematic (P&ID) review

The original system schematic (P&ID) (console and unit) as contained in the equipment specification should be reviewed at this point, to confirm all system logic and instrumentation is as specified. That is, the schematic should be reviewed in the framework of a P&ID (process and instrument diagram). All comments should be noted and the system schematics corrected.

Data sheet review

The system data sheet should be complete and be thoroughly reviewed at this point, including specific component details and desired manufacturers of major components. This review goes both ways; that is, vendor required information and user information must be detailed and correct on the data sheets. Frequently, utility and site information is not complete. This absence can only lead to reliability and communication problems in the field. As with any meeting, detailed minutes should be kept and every effort should be expended to resolve all open items prior to its conclusion. Postponing decisions only creates inefficiencies.

Exceptions to specifications

All vendor exceptions to specifications must be reviewed and either accepted or rejected. The final, mutually agreed to list of vendor exceptions should become part of the job specification.

Component sizing audit

A typical component sizing audit form is included at the end of this B.P. We will now review the major areas of this audit form.

System requirements

The first subject of discussion concerns confirmation of auxiliary system flow rates, pressures and heat loads required. This information determines the size of all major system components. It must be correct and not modified during the design of the equipment. The component sizing agenda should emphasize the need to have all required information supplied and confirmed by each critical equipment vendor. Attention is drawn to comparing values noted. If significant discrepancies appear, question them! Remember all critical equipment components are equivalent orifices, and at a specified pressure will only pass a given flow. If the component oil flow specified is greater than

the amount the components will actually pass, the excess oil will be bypassed back to the oil reservoir and could create overheating problems in the system. Conversely, if too low a value of component oil flow is specified, a system may continuously operate with both main and stand-by pump in operation since the capacity of the main pump will have been sized too small for the system.

Reservoir sizing, construction, and sub-component details

Refer to Figure 7.5.1, which is a schematic representation of an auxiliary system reservoir. Reservoir size and levels, as noted in Figure 7.5.1, must be determined at this time. Size will be a function of system flow, which previously will have been defined. The height of the reservoir should be such that, in its final field location, it will provide adequate gravity return from the main equipment.

The construction of the reservoir should be checked at this time. The original equipment vendor should have a reservoir drawing, detailing the reservoir internals, available for review. Attention is drawn to the requirement that auxiliary return fluid should not be allowed to free fall to the surface of the liquid. All returns should be through stilling tubes or sloped troughs. It is also wise to confirm that the internal design is proven, and that the manufacturer has successfully designed similar such reservoirs in the past. Accessibility for cleaning should be confirmed, and the location of return connections and pump supply nozzles should be such that maximum residence time of system fluid is ensured. Material of construction should be confirmed at this point and all details of the following reservoir sub-components reviewed:

- Reservoir heater sizing calculations
- Level control alarm
- Connection locations and size
- Additional instrumentation

Pump and driver sizing

Pump performance

Regardless of the types of pumps used, centrifugal or positive displacement, the performance curves should be reviewed at this point.

A. *Positive displacement* – Positive displacement pumps, furnished without external timing gears, are mechanically sensitive to fluid viscosity. The performance curve should be checked at all operating points to confirm that adequate rotor separation is present at low fluid viscosities. If an operating point is at the end of within 20% of a pressure vs. flow curve at a low operating viscosity (7.4–10 centistokes [50–60 SSU]), the pump vendor should be contacted to confirm a correct selection has been made. Refer to Figure 7.5.2 for an example of this case.

B. *Centrifugal pumps* – Since centrifugal pump performance must be corrected for viscous fluid operation, pump sizing must confirm that the actual operating points are not close to the operating extremities of the corrected curve. That is, any operating point should not be less than 20% of pump best efficiency point, nor be more than 110% of best efficiency point of the operating curve corrected for viscosity. Refer to Figure 7.5.3 for an example. Operation outside the stated boundaries, in addition to causing high revenue costs due to lower efficiency, can jeopardize the reliable operation of the pump.

Pump mechanical requirements

Pump data sheets must be checked at this point to confirm that proper pump case material design, bearings, seals and pump flushing arrangements are provided as specified. In addition, it is recommended that all pumps be factory tested prior to the auxiliary system test to confirm acceptable operation.

Pump unit couplings

Couplings should be selected for the maximum driver horsepower and include a sizing safety factor (usually 20–25%) above this maximum. Spacer couplings are recommended, in order to

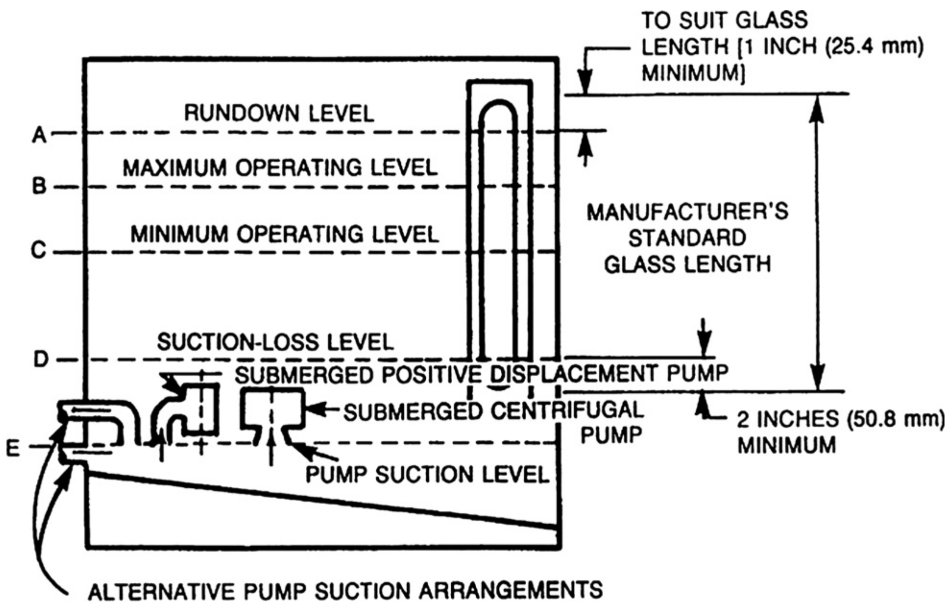


Fig 7.5.1 • Schematic representation of an auxiliary system reservoir

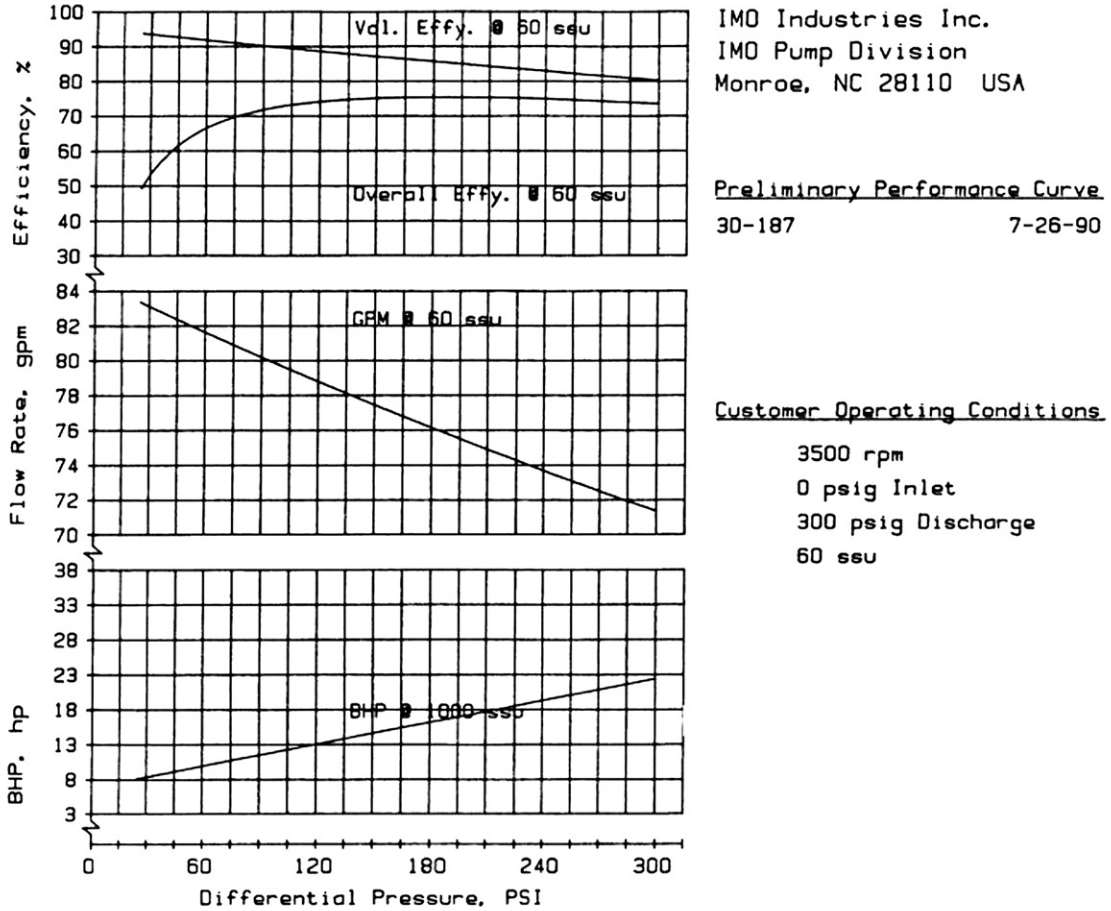


Fig 7.5.2 • Screw pump performance (Courtesy of IMO Ind.)

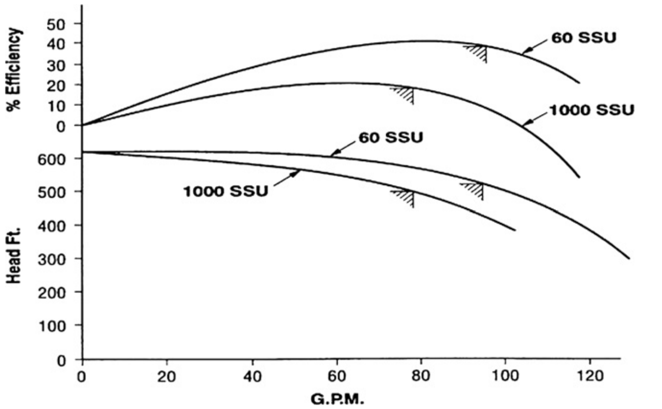


Fig 7.5.3 • The effect of oil viscosity on centrifugal pump performance

provide ease of maintenance, and minimize the necessity to remove a pump or driver while the critical equipment unit is operating. The type of coupling selected should be of high quality and reliability, and provide a minimum of three years continuous operation. While either batch lube gear type couplings or dry flexible element types can be used, the latter are preferred for their low maintenance requirements. Coupling material should be steel, as opposed to cast iron, to prevent breakage during removal or during extreme temperature changes (as during a fire). Flexible elements should be stainless steel.

The coupling shaft fit configuration and amount of shrink fit should be checked to confirm correct values.

Driver sizing

Driver sizing must be confirmed to ensure adequate delivered horsepower during all operating conditions. Utility conditions to the drivers should be re-checked at this point to ensure that values are as stated on data sheets. As an example, steam turbine data (inlet pressure and temperature, and exhaust temperature) should be checked so that all conditions will exist on site as stated. Similarly, the minimum starting voltage for motor drivers should be confirmed. Lower minimum starting voltage values than stated on data sheets will cause stand-by pump start time to be less than anticipated, shorten motor life, and could result in serious transient auxiliary pump start problems that could cause critical equipment shutdown.

Driver sizing must be confirmed with specification requirements, such that driver horsepower equals pump horsepower times a specified service factor. Selection charts for expansion turbines should be checked, and it should be confirmed that proper, standard size, electrical drivers have been selected. In applications where viscous fluids are used, pump calculations for horsepower corrections at maximum fluid viscosity must be checked. Attention is drawn to realistic sizing of

pumps and drivers concerning viscosity. If minimum site ambient is below 40°F for example, and a properly sized reservoir heater is furnished, there will not be a requirement for high viscosity operation, if it is accepted that the reservoir heater will bring the auxiliary fluid to a minimum pump starting temperature prior to pump operation. A permissive temperature switch could be installed to preclude the possibility of equipment start prior to acceptable temperature conditions.

Driver mechanical requirements

Data sheets for both main and auxiliary drivers should be checked to confirm proper mechanical design.

Motor drivers should be designed as specified, with attention being paid to bearing and motor housing design. Many smaller auxiliary systems have utilized aluminum frame motors in the past. Due to the high coefficient of thermal expansion of aluminum (double that of steel), these motors are subject to significant alignment changes with operating temperatures, which could cause coupling misalignment problems.

Expansion turbine mechanical review should include governor and safety system confirmation. Some safety valves furnished with small expansion turbines are not designed for positive shut off. This can result in operation of the turbine at lower speed once the equipment has been tripped. Most steam turbines currently operating in auxiliary systems do not have speed indicators. To ensure correct operating speed, a stroboscope or hand-held tachometer, both of which can give inaccurate readings, are used. Particularly in the case of dynamic pumps, turbine speed setting is important to ensure proper flow to the system. Therefore, any new installation incorporating expansion turbines should be equipped with speed indication.

Relief valve selection

Relief valve selection should be confirmed to qualify proper size, minimum accumulation (the pressure required over the valve setting to provide full flow) and chatter free operation. Relief valves should be located as close as possible to the pump discharge line to minimize the possibility of air entrainment in the line to the relief valve, which can result in a delayed pump flow to the unit. This would be the case if the RVs were mounted on the reservoir a significant distance from the pump discharge line.

Control valve selection

Control valve data sheets for each control valve in the system should be available for review. Information furnished on these data sheets should be complete in terms of valve sizing, actuator selection and valve controller (if present).

Valve C_v – All operating valve coefficients (C_vs) should be stated on the control valve data sheet, including the normal C_v, maximum C_v and minimum C_v. These values should be compared with the selected valve internals to ensure that all operating conditions fall within 10% to 90% of the maximum valve coefficient. Failure to confirm this can lead to valve instabilities. When reviewing valve coefficients, the system design must also be reviewed (system schematic), since certain changes in the system could render the valve unstable.

Bypass valve

For this application, the valve back pressure is atmospheric and the control valve differential depends on the condition of the auxiliary system cleanliness and any additional control valve setting (refer to the typical system schematic Figure 7.5.4).

- C_v *Minimum* – The minimum valve coefficient in this application would be with a dirty filter (filter ΔP) and one pump operating.
- C_v *Maximum normal* – The normal valve coefficient in this application occurs with a clean system (minimum filter ΔP) and one pump operating.
- C_v *Maximum* – The maximum valve coefficient would be with the main and auxiliary pumps operating, and the minimum pressure drop across the valve (clean filter).

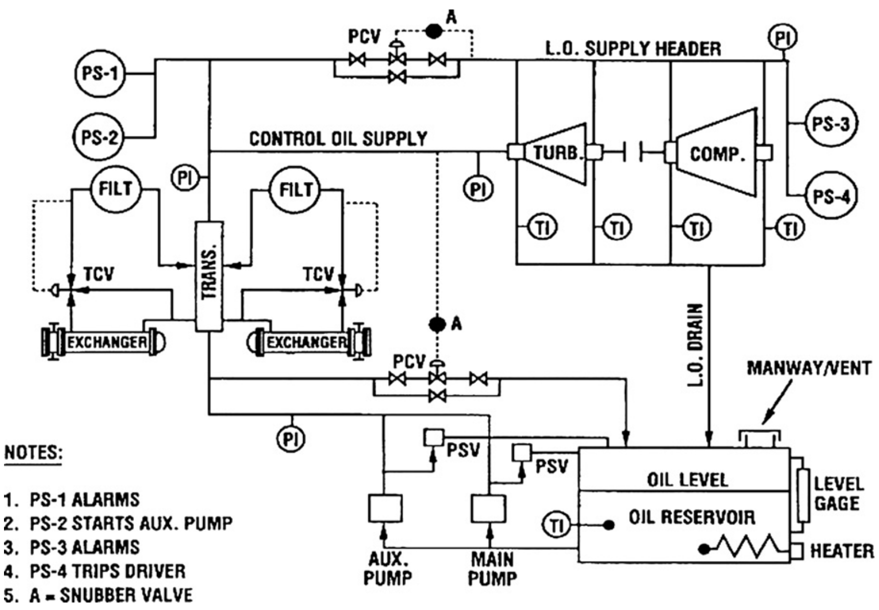


Fig 7.5.4 • Typical lube oil supply system
(Courtesy of M.E. Crane, Consultant)

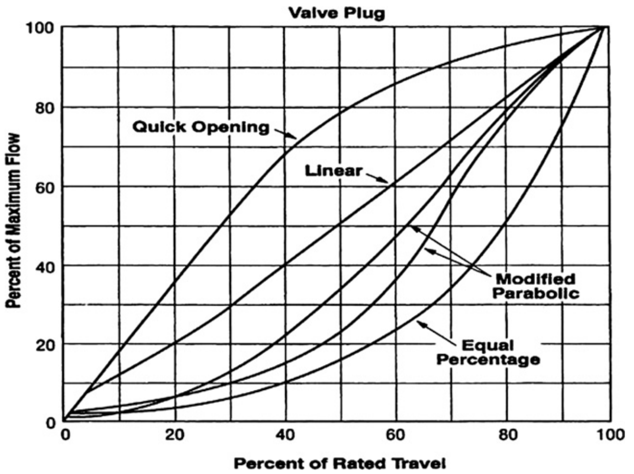


Fig 7.5.5 • Control valve flow characteristics (Courtesy of Fisher Controls, Inc.)

The maximum flow for this condition would be the normal bypass flow of the main pump plus the total flow of the auxiliary pump.

Attention is drawn to the characteristic of the valve curve for this application. The normal operating point would be approximately the minimum C_v, therefore a valve characteristic that results in a fairly significant (15–25%) valve travel for this small C_v would be desired (quick opening). Two pump operation (maximum C_v) is an abnormal case. Therefore the valve should be designed merely to pass this flow (refer to Figure 7.5.5) at 90% or less of the valve catalogue C_v.

Pressure reducing valve

In a centrifugal pump application, pressure reducing valves would experience minimum, normal and maximum C_vs similar to bypass valves, with the exception that downstream valve pressure will change with increasing flow.

When pressure reducing valves are used to reduce pressure levels (control oil pressure to lube oil pressure, seal oil pressure to lube oil pressure, etc.) the valve C_v should be selected for all possible operating cases, as mentioned above for bypass valves. Care should be taken to ensure *all possible operating cases* are considered.

Temperature control valves

The temperature control valve C_v will remain relatively constant under all auxiliary system conditions. Two way valves, however, must be sized such that the full flow pressure drop across the valve is less than the clean pressure drop across the cooler in parallel with this valve.

Differential pressure control valves and level control valves are sized and examined in the manner described above for bypass and pressure reducing valves. Details will be discussed in subsequent chapters. Viscosity corrections are required for all control valve sizing when operating viscosities exceed 7.4 centistokes (50 Sabolt Universal Seconds [50 SSU]). Significant size increases are required for high viscosity operation – i.e. values approaching 220 centistokes (1,000 SSU) – on the order of $1^1/_2$ to 2 times the selected valve coefficient without viscosity considerations.

Control valve sensing line snubber devices (dampers)

If these devices are included, a review of device design and confirmation of proper installation should be confirmed. Such devices provide unrestricted flow in one direction and restricted flow in another. The total auxiliary system operation must be reviewed in this light to confirm proper installation and orientation.

Supply pipe velocity checks

The pump header, interconnecting console pipe, and piping to the unit should all be checked for proper fluid velocity. Typical velocity values in auxiliary system supply pipes are on the order of four to six feet per second velocity. Velocity is derived from the following equation for incompressible flow:

$$Q \text{ ft/min} = A \times V$$

Where: A = Internal pipe area (m^2 or ft^2)
V = Fluid velocity (m/sec or ft/sec)

Charts for standard pipe sizes and schedules are available to determine velocities (see Figure 7.5.6).

Note that schedule 80 is usually used for carbon steel pipe below 2". Schedule 40 is used above 2". For stainless steel pipe, schedules 10 and 20 are used respectively.

Typical drain line velocities are 0.15–0.08 m/sec ($^1/_2$ to $^1/_4$ feet/sec). Attention is drawn to the need to properly size drain pipes for installations, where critical equipment is significantly elevated above the reservoir. All drain pipes should be sized with adequate area, to preclude excessive air being entrained with the oil to promote drainage back to the reservoir. An additional consideration for supply headers at the unit is that supply headers are frequently sized for one standard pipe dimension. In the case of large, critical, equipment units (two or three bodies and driver), the amount of oil from the entrance to the header to the last component decreases significantly. In an effort to minimize pipe size, many vendors specify small size headers, so pressure drop here is excessive and requires a higher supply header pressure than anticipated in the unit design. Improper sizing of critical equipment supply headers could cause excessive flow across equivalent orifices (bearings), thereby requiring all flow of the main pump and necessitating the operation of the auxiliary pump.

Transfer valve sizing

Transfer valve configuration and materials of construction should be confirmed at this point. Transfer valve design should be checked to confirm tight shutoff.

Cooler sizing

The cooler data sheet should be reviewed to confirm correct duty, cooling media details, fouling factors and materials of construction.

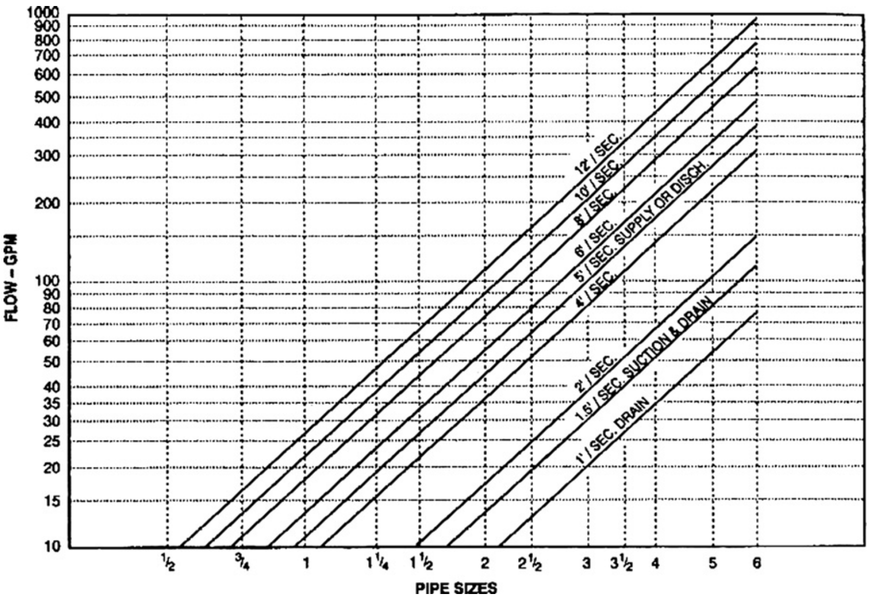


Fig 7.5.6 • Typical pipe sizing chart

Filter sizing

Filter information should be reviewed to confirm correct filter sizing for the normal and the maximum viscosity case (in the case of viscous fluids). Additionally, maximum filter collapse pressure, internal filter cartridge design and cartridge sealing design should also be reviewed at this time.

Instrumentation

All instrumentation should be reviewed to confirm proper selection, materials of construction and proposed installation locations. All instrumentation loops should be reviewed to ensure that critical instrumentation can be calibrated and maintained while unit is in operation.

Console layout and component arrangement

Having confirmed that component sizing and selection are acceptable, the console layout and arrangement of components must be reviewed. Methods of review use either the review of outline drawings of the proposed arrangement, or a model review, or a CAD 3D drawing review. Many vendors and users have found that models or CAD 3D drawings aid greatly in understanding and reviewing maintenance accessibility and layout considerations.

Console construction

Auxiliary equipment consoles or modules house most of the components that are present in the auxiliary systems. Their construction should be reviewed to ensure proper stiffness and facilities for installation on site. Many horizontal consoles are constructed in a flexible manner that can result in bending or excessive pipe strains introduced into components during shipment and at installation. It is suggested that full length cross members be positioned as a minimum under pumps, coolers and filters on the equipment baseplate (see Figure 7.5.7). If the baseplate is to be grouted in the field, grout and vent holes should be specified and reviewed for accessibility to pore grout when equipment is installed on the baseplate.

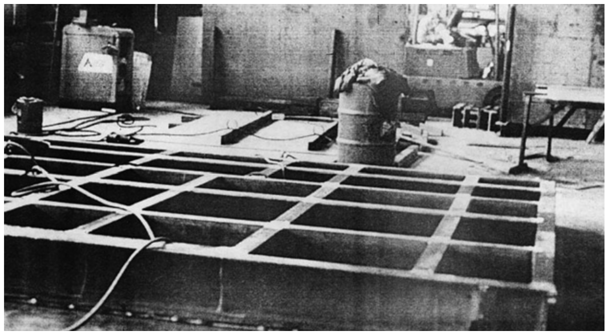


Fig 7.5.7 • Console baseplate construction (Courtesy of Fluid Systems)

Maintenance accessibility

Since equipment must be maintained and calibrated while the auxiliary system is in operation, it is important to provide ample personnel space such that equipment can be maintained safely and reliably without damage to surrounding components. A rule of thumb is to provide approximately one meter of space around components for accessibility. Note that this is with the utility lines installed. The review of equipment on a model, CAD 3D drawing or an outline should be made considering installation of all utility lines that will be installed in the field.

Accessibility for on-line testing and calibration

Considering that many components (pumps, drivers, coolers, filters, control valves, instrumentation) will be tested and calibrated with equipment still being in operation, accessibility for this operation must be considered.

In addition to reviewing the vendor manufactured skids, the placement of all skids in the field must be reviewed for

AUXILIARY SYSTEM COMPONENT SIZING AUDIT FORM

APPLICABLE SPECIFICATIONS ______________________________

CRITICAL EQUIPMENT DESCRIPTION ______________________________

SCHEMATIC DRAWING NUMBERS ______________________________

DATA SHEET ______________________________

1. SYSTEM REQUIREMENTS

A. OIL REQUIREMENTS

APPARATUS		FLOW RATE				GAGE PRESSURE (At Equipment)		ΔP (At Equipment)		HEAT LOAD	
		GPM	L/min.	GPM	L/min.	PSI	bar	PSI	bar	BTU/HR	kw
COMPRESSOR	THRUST BEARING										
	SUCTION END JOURNAL										
S.O.	DISCHARGE END JOURNAL										
COMPRESSOR SEALS (TOTAL)	NORMAL										
	MINIMUM										
	MAXIMUM										
Leakage ____ GPD (___ L/day)											
COMPRESSOR	THRUST BEARING										
	SUCTION END JOURNAL										
S.O.	DISCHARGE END JOURNAL										
COMPRESSOR SEALS (TOTAL)	NORMAL										
	MINIMUM										
	MAXIMUM										
Leakage ____ GPD (___ L/day)											
GEAR											
S.O.											
TURBINE	THRUST BEARING										
	STEAM END JOURNAL										
S.O./P.R.	EXHAUST END JOURNAL										
GOVERNOR											
SERVO MOTOR	NORMAL										
SERVO MOTOR	MAXIMUM										
TRIP & THROTTLE VALVE											
TURNING GEAR											
MOTOR	OUTBOARD END BEARING										
S.O.	COUPLING END BEARING										
NON RETURN VALVE											
CONTINUOUS LUBE COUPLINGS(S) TOTAL											
TOTALS											

Fig 7.5.8 • Auxiliary system component sizing audit form

accessibility. Consideration of the skid arrangement only to be complicated by installation against a column or wall in the field will not obtain the objectives of total accessibility.

Utility supply arrangement

Care should be given to the routing of all utility (conduits, steam lines, water lines) supply lines in order to maximize accessibility to the critical equipment auxiliary systems.

Considerations for component disassembly

All components must be capable of being dissembled quickly, easily and safely while the unit is operating in the field. To meet this requirement, sufficient space around the auxiliary console must be available for such exercises as cooler bundle removal, filter cartridge removal and auxiliary or main driver removal. In addition, consoles are frequently installed in congested areas, and so lifting arrangements should be reviewed beforehand to confirm components can be removed in a safe and easy manner.

This completes comments concerning the component sizing audit. All changes made during this meeting should be documented and followed up to guarantee that final component design and arrangements are as specified and agreed to in this meeting.

Factory testing and inspection

Having properly specified, designed and manufactured the unit, it remains to confirm proper arrangement and operation. This is accomplished during factory testing and inspection. The

objectives, then, of this phase are to confirm the proper arrangement details and functional operation of the equipment.

Test agenda

To meet the objectives of this phase the equipment must be thoroughly tested prior to field installation. The test should confirm the functional operation of all components as they will operate in the field. In order to ensure a valid factory test, a test agenda should be prepared approximately two months before the test date, and be reviewed by the equipment purchaser. Specific areas of concern are:

Flushing

Component system flushing is required as an inspection point and should be accepted prior to the initiation of the test. Additionally, all test agendas should always be structured such that a limit for each item to be tested is specifically defined in the test agenda. The flushing acceptance criteria must be mutually agreed upon and be adhered to during the review of flushing the operation. A typical field flushing procedure is given in B.P. 7.18.

Confirm arrangement details

Prior to commencement of test, it should be confirmed that the arrangement of all components, controls and instruments agree with design requirements. Any discrepancies should be corrected prior to functional test.

Confirm proper test fluid in sufficient capacity is present to prior to the initiation of test.

Temporary test setup

If it is necessary to import utilities or switches for the test that are not normally present, as in the case of steam turbine steam generator and control switches, these items must be confirmed prior to test initiation. In addition, a means of confirming proper flow rates, temperature and pressure during the test must be provided. Supply lines to the unit must be provided with properly size orifices to duplicate unit flow requirements.

Functional testing

Having confirmed proper test setup, calibration of all instrumentation, proper fluid and test instrumentation, the functional test is now ready to be performed. As a minimum the following tests should be performed on any auxiliary system console:

- Relief valve test (if supplied)
- Transfer valve test
- Auto start test of auxiliary pump with the following conditions:
- Main pump tripped
- Two pump operation (main pump in operation, standby pump started)

During all functional testing, any system pulsations or pressure drops above specified values are reason for the non-acceptance of the test. All components which do not meet requirements must be corrected and units must be completely retested.

The value of testing the auxiliary console with the unit should be seriously considered. Since the console and the unit piping form the specific auxiliary system, there is a significant benefit to testing both together. The additional costs of such a test should be evaluated against the potential reduction of reliability and loss of operation time in the field if any malfunctions exist that were not determined by test of the console alone.

Testing the console without the unit assumes that a model of the actual equivalent critical equipment orifices has been properly installed during the test. There is no assurance that the actual manufacturer of the equipment has not incorporated changes in equivalent orifices sizes. This would result in different flows to the unit and different console responses for various transient operational modes. Daily revenue should be considered in evaluating the extra cost involved for a full console unit test in a manufacturer's works. If a full unit test is to be performed, it is still recommended that the console be tested at the point of manufacture prior to the unit test in the critical equipment vendors shop. This action will reveal console design problems prior to shipment to the vendor's plant.

I. System requirements – see chart (previous page) for input data

1. Total pump flow (Positive displacement pump)	= ______ × equipment flow (l_a)
Total pump flow	= ______
(Centrifugal pump)	= ______ (total flow in l_a)
2. Bypass flow	= (1) – l_a total flow
(Positive displacement pump)	= ______–_____
	= ____________
3. Total heat load (from la)	= ______ BTU/HR or kW
4. Pump discharge pressure	______60 _____1000
Viscosity (SSU) or centistokes	________ ________
A. Lube oil pressure	________ ________
(at equipment)	________ ________
B. Elevation ΔP	________ ________
C. Pipe ΔP	________ ________
D. Valve ΔP	________ ________
E. Cooler ΔP	________ ________
F. Filter (clean) ΔP	________ ________
G. Miscellaneous ΔP	________ ________
Pump discharge press (Add A through G)	Pressures in kPa or PSI

Note: If system is combined with seal and/or control oil, add highest value to determine pump discharge pressure.

II. Component requirements

Confirm sizing as stated below and check data sheet and specific requirements for each component.

A. Pump selection

	Positive displacement Main/aux	Centrifugal main/aux
1. Pump type	______	______
2. Make	______	______
3. Model	______	______
4. *Speed	______	______
5. Disch. press @ 10 centistokes (60 SSU [rated])	______	______
6. Disch. press @ max. centistokes (SSU)	______	______
7. Rated flow @ 10 centistokes (60 SSU)	______	______
8. Flow @ max. SSU	______	______
9. Flow @ relief valve press (positive displacement pump only)	______	______
10. Rated kW (BHP)	______	______
11. Max. kW (BHP [@R.V. and max. viscosity])	______	______
12. NPSH available-m (ft)	______	______
13. NPSH required-m (ft)	______	______
14. Suction lift (if pumps are mounted above fluid level-m [ft])	______	______

*If steam turbine driver is used, it is recommended that speed should be 2 pole (3600/3000 RPM) motor speed to minimize steam rate.

B. Coupling selection

	Main	Auxiliary
1. Pump	______	______
2. Coupling model	______	______
3. Size	______	______
4. Driver max. power-kW (HP)	______	______
5. kW (HP)/100 RPM	______	______
6. Coupling kW (HP)/100 RPM	______	______

Note: Confirm appropriate coupling service factor is used. Rotary (PD) pumps require a higher service factor.

C. Driver selection

	Main	Auxiliary
1. Service	______	______
2. Type	______	______
3. Speed	______	______
4. Pump max. power kW (HP)	______	______
5. Driver rated power kW (HP) (= 1.1 × pump max. kW or HP)	______	______
6. Driver normal power (@ 10 centistokes [60 SSU])	______	______
7. Turbine steam rate kg/kW-HR (lb/HP-HR [max/rated])	______	______
8. Steam quantity @ minimum steam energy condition kg/HR (lb/HR)	______	______
*9. Driver starting time		
10. RPM-rated speed	______	______

Note: Confirm sufficient steam is available at minimum energy conditions. *Calculated for minimum energy conditions (minimum steam energy or motor minimum starting voltage) and pump rated conditions. If greater than three (3) seconds, accumulator(s) should be used.

D. Relief valve selection (Positive displacement pumps only)

1. Pump max. discharge pressure at max. viscosity = ______
2. Relief valve pressure = 1.1 × D1 or 172 kPa (25 PSI) greater, whichever is higher. ______

Relief valve type (modulating preferred) = ______
Model = ______
Set pressure = ______
Overpressure (pressure to pass full flow)-kPA (PSI) = ______
Normal leakage (valve closed) = ______

E. Reservoir sizing (based on rectangular tank) Per API 614

1. Normal flow liters/min (GPM) = ______
2. Retention time (minutes) = ______
   2A Capacity = (1) × (2) (Refer to Figure 7.5.9) ______

Confirm size

3. Reservoir length × width mm (inches) = ______
4. Capacity cm (inch) of height $= \frac{(3)}{231 \text{ in.}^3/\text{gal}}$

   = ______ liter/cm (gal/inch)
5. Level E (pump suction level) = ______ cm (in) above grade
6. Level D (suction loss level) = (5) + level required to
   = maintain prime
   = ______
7. Level C = (6) + 5 (minutes) × (1)
   Note: 5 minutes = Working time
   Working capacity (volume between levels C & D) or level C

$$= \frac{8(\text{min}) \times (1)}{(4)} + \text{tank bottom height above grade}$$

   Note: 8 minutes = retention time
   Retention capacity = volume between bottom of tank and level C.
8. Level B = Highest level of oil during operation (approximately 1 minute retention time)
9. Level A = Highest level oil can reach
   = Level B + capacity contained in all components that drain back to the reservoir ÷ (4)
   Note: This quantity should also include allowance for interconnecting piping and any overhead tanks.

10. Minimum reservoir free surface area:

$= 232\ cm^2$/LPM of normal flow ($0.25\ ft.^2$/GPM)
$= 0.25 \times (1)$
$= ft.^2$

Confirm reservoir internals, material, etc. meet data sheet and specifications required. Review reservoir internal drawing.

F. Reservoir heating requirements

Type	Electric	Steam
Time to head oil from ________	°C (°F) to ________	°C (°F) = ________ Hours
1. Calculated heat load (minus reservoir heat loss)	= ________kJ (BTU)	
2. Heater size kJ/HR (BTU/HR)	$= \dfrac{(F1)}{\text{Total time allowed (hours)}}$	
3. Electric heater max watt density	= ____________	

Note: Confirm if heaters can be removed without draining reservoir.

G. Supply pipe velocity

Maximum velocity 1.2–1.8 m/sec (4–6 ft/sec)
Maximum console supply pipe velocity = _________m/sec (ft/sec)
Maximum unit supply pipe velocity = _________ m/sec (ft/sec)

H. Control valve sizing

H1. Bypass (back pressure) valve

1.1 Type: self acting, pneumatic or electric controller ____________
1.2 Make ____________
1.3 Model ____________
1.4 Action – direct or reverse ____________
1.5 Valve plug type ____________
1.6 Failure mode ____________
1.7 Actuator size ____________
1.8 Actuator force available/force required ____________
1.9 Maximum valve C_v = ____________
1.10 Operating C_v min. (one pump dirty system) = ____________
1.11 Operating C_v max. (two pumps clean system) = ____________

Note: Operating C_vs should be between 10% – 90% of valve max. C_v.

Sensing line pulsation snubber required? If so, confirm proper orientation. Confirm fast response is to *open or close* valve.

H2. Transfer valve(s)

2.1 Make ____________
2.2 Model ____________
2.3 Size ____________
2.4 Plus type – *Taper, Straight, Globe* ____________
2.5 Lifting jack required? ____________
2.6 Tight shut-off required? ____________
2.7 Max ΔP on changeover ____________

H3. Temperature control valve(s)

3.1 Make ____________
3.2 Model ____________
3.3 Size ____________
3.4 Normal flow liters/min (GPM) ____________
3.5 Temperature range °C (°F) ____________
3.6 Valve max operating C_v ____________
(If 2-way valve, C_v must be based on clean cooler)

Note: Butterfly type valve often used for 2-way applications.

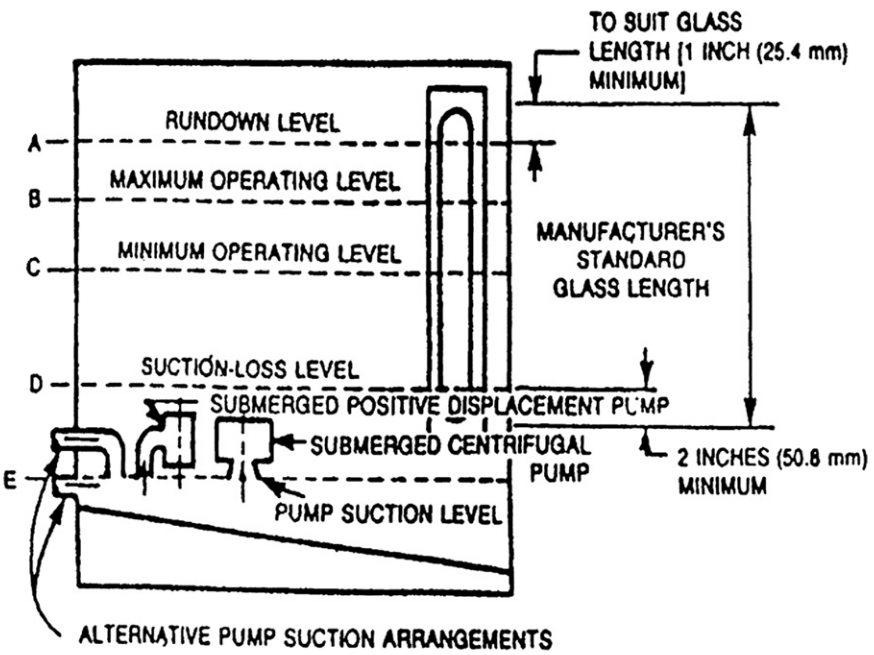


Fig 7.5.9 • Reservoir levels and oil level glass details

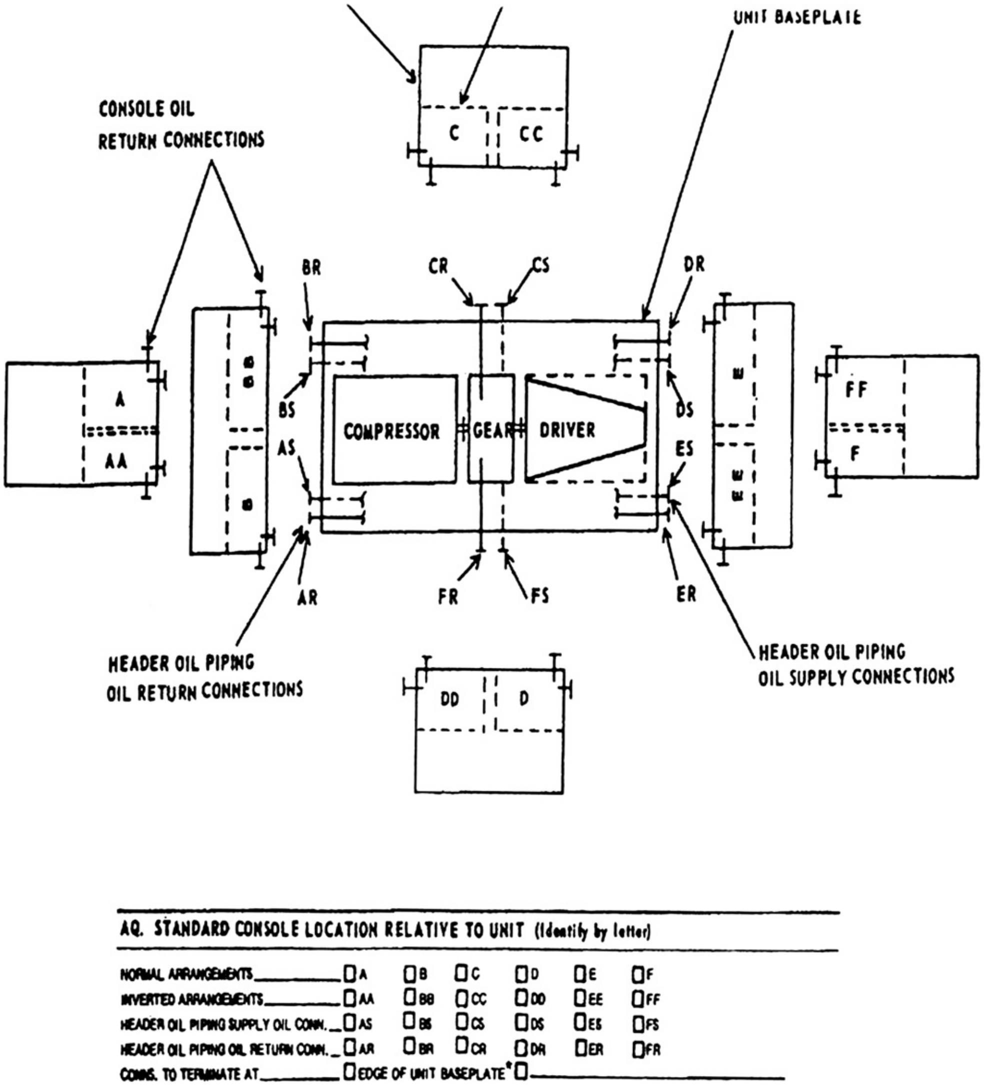


Fig 7.5.10 • Connection orientation drawing (Courtesy of Elliott Co.)

H4. Pressure reducing valve

4.1 Type: self acting, pneumatic or electric	__________
4.2 Make	__________
4.3 Model	__________
4.4 Action – direct or reverse	__________
4.5 Valve plug type	__________
4.6 Failure mode	__________
4.7 Actuator size	__________
4.8 Actuator force	__________
4.9 Maximum valve C_v =	__________
4.10 Normal valve operating C_v = (Unit at operating speed)	__________
4.11 Minimum valve operating C_v = (Unit at rest – oil system on)	__________

I. Cooler sizing

Type	Shell and Tube	Air (Fin Fan)
1. Twin or single	__________	__________
2. Make	__________	__________
3. Model	__________	__________
4. Size	__________	__________
5. Heat load kJ/HP (BTU/HR)	__________	__________
6. Oil side ΔP clean kPa (PSI)	__________	__________
7. Fouling factor (total)	__________	__________
8. Oil flow LPM (GPM)	__________	__________
9. Water quantity LPM (GPM)	__________	__________

J. Filter sizing

1. Make	__________
2. Model	__________
3. Type – surface depth	__________
4. Normal flow LPM (GPM)	__________
5. Max. flow LPM (GPM)	__________
6. Filtration (microns)	__________

7. Clear filter ΔP max-kPa (PSI) ____________
8. Cartridge material ____________
9. Type end seals ____________
10. Cartridge – single or multiple ____________
11. Cartridge center tube material ____________
12. ΔP at max viscosity-kPA (PSI) ____________
13. Collapse pressure-kPA (PSI) ____________
14. Number of cartridges ____________
15. LPM (GPM) per cartridge ____________

K. Switches or transmitters

Confirm proper range, type, materials and maximum dead-band (change in actuation point) of each switch. Confirm proper selection of transmitters.

L. Gauges

Confirm proper range, type, material of each pressure, differential pressure, temperature and level gauges.

M. Accumulator sizing

1. Type: bladder ________ or direct acting ________
2. System flow KPM (GPM) $=$ ________
3. System transient time (sec.) $=$ ________
4. Capacity of fluid required $= \frac{(2)\times(3)}{60} =$ ________ liters (gallons)
5. System pressure below which accumulator begins to drain kPA (PSIA) $=$ ________
6. Precharge pressure kPa (PSIA) $=$ ________
7. Proposed accumulator internal volume (approximately 90% of normal size) $=$ ________
8. Actual fluid capacity per accumulator $=$ ________
   $(7)\times\left[1-\left(\frac{(6)}{(5)}\right)\right] =$ ________
9. Precharge type: manual, self contained, automatic

N. Additional tank sizing and construction confirmation

- Overhead rundown (lube)
- Overhead (seal)
- Degassing tank(s)

These tanks should be checked against specifications data sheets for proper capacity, construction and ancillaries.

O. Piping, vessel, flange and component material

Confirm that all specified materials are supplied.

P. Console and unit connection orientation

Refer to Figure 7.5.10 and finalize all connection locations.

Best Practice 7.6

Monitor lube/seal oil reservoir level on all refrigeration applications to ensure that seal oil does not enter the process loop and foul the chillers.

The entrance of oil through the oil seals into refrigeration systems will reduce the capacity of the system and impact on process unit revenue.

Careful monitoring of oil reservoir level will signal oil seal system problems and lead to their resolution before the refrigeration system, plant capacity and revenue are affected.

Lessons Learned

Failure to monitor and correct infiltration of oil into refrigeration systems has resulted in large decreases in plant capacity totaling millions of USD in revenue.

This is a common problem with plants that have centrifugal compressors using seal oil systems.

It is sometimes possible to correct this problem by modification to the seal oil system design (see B.P. 7.42).

Contamination of refrigeration systems by centrifugal compressor seal oil has been one of the major reasons for the use of dry gas seals and/or field modifications to dry gas seal systems.

Benchmarks

This best practice has been used since the mid-1980s, when multiple issues had reduced the capacity of refrigeration processes and affected plant revenue. Since that time this best practice has been used to optimize oil seal system reliability and refrigeration process plant revenue.

B.P. 7.6. Supporting Material

Sweet hydrocarbon or inert gas service

For sweet or inert gas service, the seal oil drain can be returned directly to the reservoir, provided the drainers are sized for adequate residence time and the seal oil leakage is reasonable (less than one gallon per hour per seal). A sweet hydrocarbon gas is defined as a gas that does not contain hydrogen sulfide (H_2S). The vent line on top of the drainer can be routed to a lower pressure source, to the atmosphere or back to the compressor suction. If routed back to the compressor suction, a demister should be installed to

prevent oil from entering the compressor case. The sizing of the orifice in the vent line of each drainer is critical, since it ensures that all contaminated oil flow will enter the drainer. Too low a velocity will allow contaminated oil to enter the compressor, but too high a velocity could cause oil to enter the compressor via the vent or reference line. Typical velocities in this line should be 4.6–6 m/sec (15–20 ft/sec).

Best Practice 7.7

Use centrifugal single stage pumps instead of screw pumps whenever possible, to increase the reliability of lube oil systems.

The use of centrifugal pumps eliminates the need for relief and backpressure (bypass) control valves.

Malfunction of relief valves and/or backpressure control can cause an unscheduled shutdown of unit and hence result in significant revenue loss.

Single stage centrifugal pumps can be used whenever the ambient temperature along with the use of thermostatically controlled reservoir heaters maintain an oil viscosity that allows the use of a centrifugal pump (oil viscosity is low enough to minimize the effect of viscosity on centrifugal pump power – low viscosity correction factors).

Lessons Learned

The most common cause of oil system induced unit trips is the failure of the backpressure control valve to respond to transient system changes (trip or slow speed reduction of the main turbine driven oil pump). The use of centrifugal pumps eliminates the need for a backpressure control valve.

Benchmarks

This best practice has been used since the mid-1980s to optimize the reliability of oil systems, and to achieve compressor train reliabilities exceeding 99.7%

B.P. 7.7. Supporting Material

The pumps

Auxiliary systems that contain liquids use positive displacement or centrifugal pumps depending on the application – typical examples are shown in Figures 7.7.1, 7.7.2 and 7.7.3. The screw pump (Figure 7.7.1) and gear pump (Figure 7.7.2) are used in systems containing oil. The centrifugal pump (Figure 7.7.3) is used primarily for non-viscous duty, but can be used for oil systems if properly sized, and the efficiency and horsepower penalties are acceptable.

Regardless of the type of pump used, the function of all pumps in auxiliary system service is '*to continuously supply the*

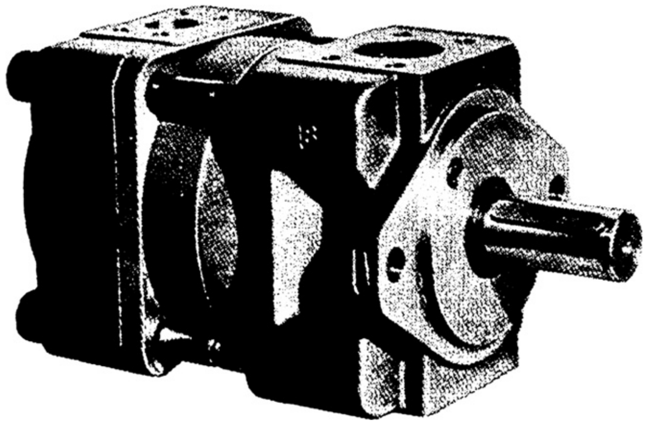


Fig 7.7.2 • Gear pump (Courtesy of IMO Industries)

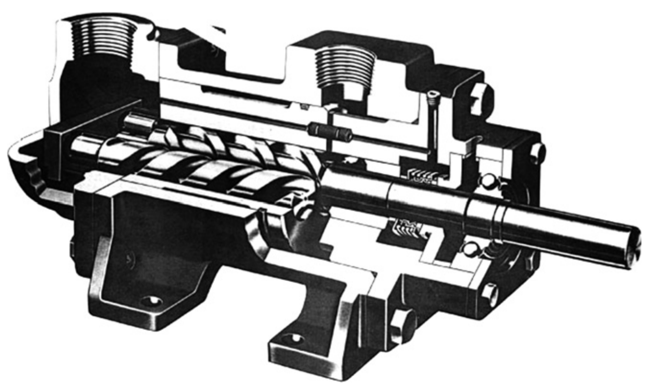


Fig 7.7.1 • Screw pump (Courtesy of IMO Industries)

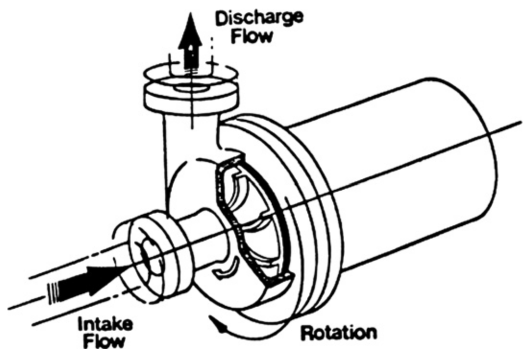


Fig 7.7.3 • Centrifugal pump

ROTARY PUMP DATA SHEET

Page 1 of 2

ISSUE	APPROVED	ISSUE	APPROVED	JOB NO.	ITEM NO.
1		3		P.O. NO.	REQ NO.
2		4			

APPLICABLE TO: ○ PROPOSAL ○ PURCHASE ○ AS BUILT DATE ______ REVISION ______
FOR ______ UNIT ______
SITE ______ SERIAL NO. ______
SERVICE ______ NO. PUMPS REQUIRED ______
MANUFACTURER ______ NO. MOTORS REQUIRED ______
PROVIDED BY ______ SIZE AND TYPE ______
INQUIRY NO. ______ MODEL ______

NOTE: ○ INDICATES INFORMATION TO BE COMPLETED BY PURCHASER ☐ BY MANUFACTURER
API STANDARD 676 GOVERNS UNLESS OTHERWISE NOTED.

OPERATING CONDITIONS (TO BE COMPLETED BY PURCHASER)

Liquid ______
Pumping Temperature, °F:
Normal ______ Maximum ______ Minimum ______
Specific Gravity ● PT ______
Vapor Pressure ● PT, psia ______
Viscosity ● PT (SSU)(cp):
Maximum ______ Minimum ______
Electrical Area Hazard:
Class ______ Group ______ Division ______
Site Temperature, °F:
Normal ______ Maximum ______ Minimum ______
Corrosion/Erosion Caused by ______

Rated Capacity ● PT, US gpm:
● Maximum Viscosity ______ ● Minimum Viscosity ______
Discharge Pressure, psig:
Maximum ______ Minimum ______ Rated ______
Suction Pressure, psig:
Maximum ______ Minimum ______ Rated ______
Differential Pressure, psi:
Maximum ______ Minimum ______ Rated ______
NPSH Available, psi ______
Hydraulic hp ______
Location: ○ Indoor ○ Heated ○ Outdoor ○ Unheated
REMARKS: ______

PERFORMANCE (TO BE COMPLETED BY MANUFACTURER)

AT RATED CONDITIONS:
NPSH Required, psi ______
Rated Speed, rpm ______
Displacement, US gpm ______
REMARKS: ______

AT RATED CONDITIONS:
Volumetric Efficiency, % ______
Mechanical Efficiency, % ______
bhp ● Maximum Viscosity ______
bhp ● Relief Valve Setting ______
Maximum Allowable Speed, rpm ______
Minimum Allowable Speed, rpm ______

CONSTRUCTION (TO BE COMPLETED BY PURCHASER AND MANUFACTURER)

Pump Type:
☐ Spur Gear ☐ Twin-Screw ☐ Vane
☐ Helical Gear ☐ Three-Screw ☐ Progressing Cavity
☐ Other
Casing:
Maximum Allowable Pressure: ______ psig ● ______ °F
Hydrostatic Test Pressure: ______ psig
Jacket Pressure: ______ psig ● ______ °F
Rotor Mount: ☐ Between Bearings ☐ Overhung

Nozzle	Size	Rating	Facing	Location
Suction				
Discharge				
Gland Flush				
Drains				
Vents				
Jacket Connection				

Packing:
☐ Manufacturer and Type ______ ☐ No. of Rings ______

Fig 7.7.4a • Rotary pump data sheet (Courtesy of The American Petroleum Institute)

system fluid at the required pressure and flow rate'. Let's examine this functional definition in detail.

'To continuously supply' means that the pump must be capable of uninterrupted operation for the same period of time as the critical equipment it is servicing. Critical equipment is designed for a minimum of three years' operation between scheduled shutdowns. In order to attain this reliability, the pump and its unit components (coupling and driver) must be properly specified, selected and designed.

As a first step, the equipment must be properly specified. The American Petroleum Institute specifications (API-614) provide a good basis for specifying highly reliable pumps and steam turbine drivers. As previously stated, all equipment requirements and site data should be entered on a data sheet to ensure a correct selection of the pump. Again, each major component of the system must be treated the same as the critical equipment to ensure maximum system reliability. A typical rotary and centrifugal pump data sheet are included in Figures 7.7.4a to 7.7.5b for reference (please note that only 2 of 5 pages of Figure 7.7.5a and 7.7.5b are shown).

In order to ensure reliable, trouble free operation, pump mechanical seals are recommended instead of shaft packing. A properly selected and installed pump mechanical seal in auxiliary system service can operate continuously for a three year period.

ROTARY PUMP DATA SHEET

Page 2 of 2

CONSTRUCTION (Contd)

Timing Gears? ☐ Yes ☐ No
Bearing Type: ☐ Radial ______ ☐ Thrust ______
Lubrication Type:
☐ Pumped Fluid ☐ Ring Oil ☐ Oil Mist
☐ External ☐ Oil Flood ☐ Grease
☐ Lubricant Type ______
REMARKS: ______

○ Mechanical Seals:
☐ Manufacturer and Model ______
☐ Manufacturer Code ______
○ API 610 Seal Flush Plan ______
☐ API 610 Code ______
Piping for Seal Flush Furnished by:
○ Pump Vendor ○ Others
Piping for Cooling/Heating Furnished by:
○ Pump Vendor ○ Others

MATERIALS (TO BE COMPLETED BY MANUFACTURER)

Casing ______
Stator ______
End Plates ______
Rotor(s) ______
Vanes ______
Shaft ______
Sleeve(s) ______

Gland(s) ______
Bearing Housing ______
Timing Gears ______
Baseplate ______
REMARKS: ______

SHOP TESTS (TO BE COMPLETED BY PURCHASER)

Test	Nonwitnessed	Witnessed
Hydrostatic	○	○
Mechanical Run	○	○
Performance	○	○
NPSH	○	○

○ Shop Inspection
○ Dismantle and Inspect After Test
○ Other ______

REMARKS: ______

DRIVER (TO BE COMPLETED BY PURCHASER AND MANUFACTURER)

○ Motor ○ Turbine ○ Other ______
○ Driver Data Sheet ______
☐ ______ hp @ ______ rpm

REMARKS: ______

DRIVE MECHANISM (TO BE COMPLETED BY PURCHASER AND MANUFACTURER)

○ Direct Coupled ○ Toothed Belt Drive ○ Variable Speed
☐ Coupling Manufacturer ______

REMARKS: ______

BASEPLATE (TO BE COMPLETED BY PURCHASER)

○ By Pump Manufacturer ______
REMARKS: ______

○ Decking ______
○ Open Beam Support ______
○ Fully Grouted ○ Ungrouted

ADDITIONAL INFORMATION

______

Fig 7.7.4b • Rotary pump data sheet (Courtesy of The American Petroleum Institute)

All specified pump operating conditions should be confirmed so that they reflect the actual system requirements. A check list is presented in Figure 7.7.6. This step ensures that the second part of the pump function definition will be met: '*to supply the system fluid at the required pressure and flow rate*'.

As previously mentioned, there are two major classifications of pumps, and either can be used for auxiliary system duty:

- Positive displacement pumps
- Dynamic pumps

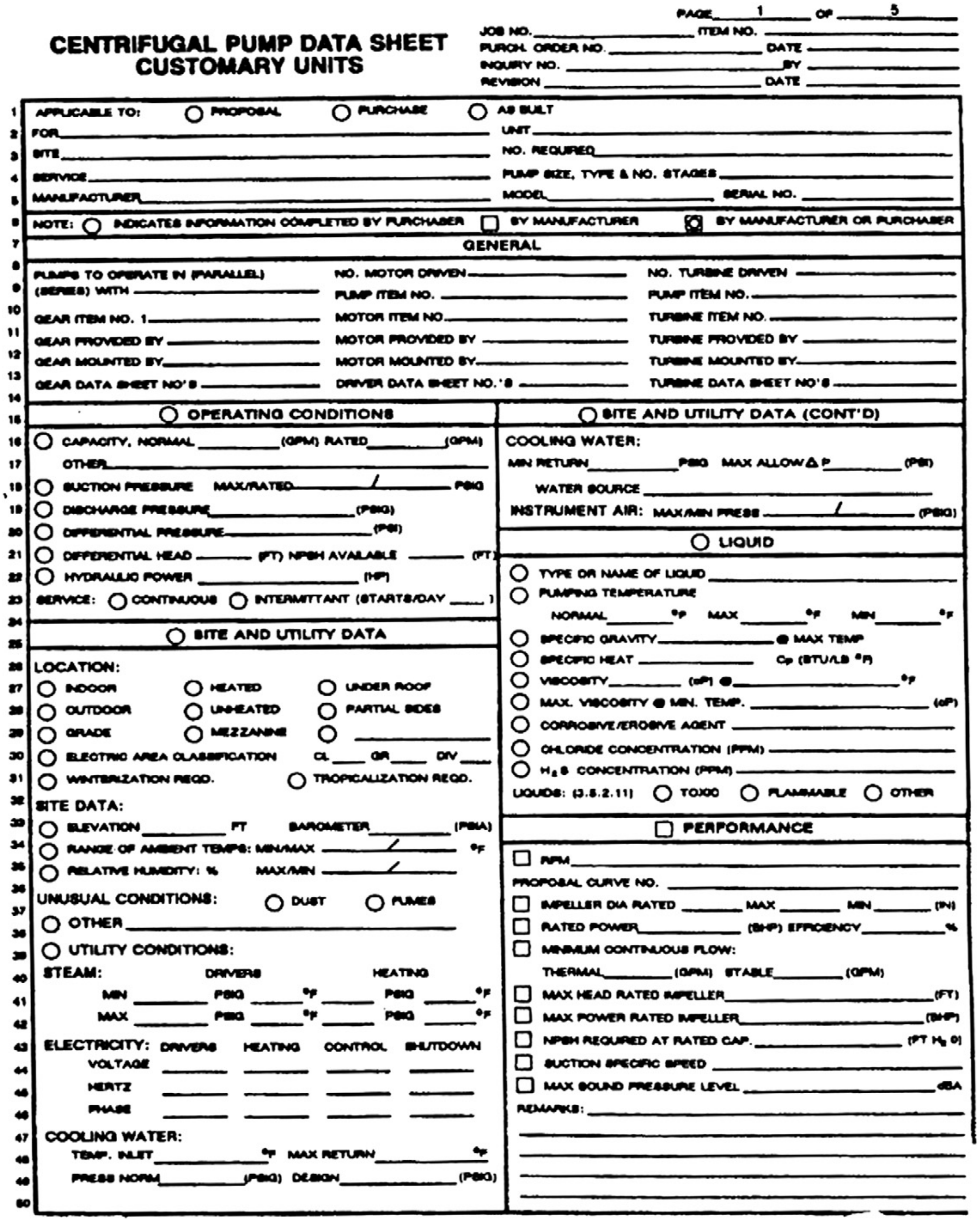
The definitions and characteristics of both are shown in Figure 7.7.7.

Pump differential head in Figure 7.7.7 is expressed in feet of liquid and is determined by:

Differential pump head (ft)

$$= \frac{2.311\left(\frac{\text{ft of H}_20}{\text{psi}}\right) \times \text{pump differential pressure (psi)}}{\text{specific gravity}}$$

With this relationship it is very important to understand that 'head' is energy. Referring back to Figure 7.7.7, we can see that a positive displacement pump is a variable energy (head) device as opposed to a dynamic pump which is a fixed energy (head) device. As the specific gravity of a liquid (the ratio of the weight of a given



PAGE 1 OF 5

CENTRIFUGAL PUMP DATA SHEET
CUSTOMARY UNITS

JOB NO. ____ ITEM NO. ____
PURCH. ORDER NO. ____ DATE ____
INQUIRY NO. ____ BY ____
REVISION ____ DATE ____

APPLICABLE TO: ○ PROPOSAL ○ PURCHASE ○ AS BUILT
FOR ____ UNIT ____
SITE ____ NO. REQUIRED ____
SERVICE ____ PUMP SIZE, TYPE & NO. STAGES ____
MANUFACTURER ____ MODEL ____ SERIAL NO. ____
NOTE: ○ INDICATES INFORMATION COMPLETED BY PURCHASER □ BY MANUFACTURER ⊠ BY MANUFACTURER OR PURCHASER

GENERAL

PUMPS TO OPERATE IN (PARALLEL) (SERIES) WITH ____ | NO. MOTOR DRIVEN ____ | NO. TURBINE DRIVEN ____
PUMP ITEM NO. ____ | PUMP ITEM NO. ____
GEAR ITEM NO. 1 ____ | MOTOR ITEM NO. ____ | TURBINE ITEM NO. ____
GEAR PROVIDED BY ____ | MOTOR PROVIDED BY ____ | TURBINE PROVIDED BY ____
GEAR MOUNTED BY ____ | MOTOR MOUNTED BY ____ | TURBINE MOUNTED BY ____
GEAR DATA SHEET NO'S ____ | DRIVER DATA SHEET NO.'S ____ | TURBINE DATA SHEET NO'S ____

○ OPERATING CONDITIONS

○ CAPACITY, NORMAL ____ (GPM) RATED ____ (GPM)
OTHER ____
○ SUCTION PRESSURE MAX/RATED ____ / ____ PSIG
○ DISCHARGE PRESSURE ____ (PSIG)
○ DIFFERENTIAL PRESSURE ____ (PSI)
○ DIFFERENTIAL HEAD ____ (FT) NPSH AVAILABLE ____ (FT)
○ HYDRAULIC POWER ____ (HP)
SERVICE: ○ CONTINUOUS ○ INTERMITTANT (STARTS/DAY ____)

○ SITE AND UTILITY DATA

LOCATION:
○ INDOOR ○ HEATED ○ UNDER ROOF
○ OUTDOOR ○ UNHEATED ○ PARTIAL SIDES
○ GRADE ○ MEZZANINE ○ ____
○ ELECTRIC AREA CLASSIFICATION CL ____ GR ____ DIV ____
○ WINTERIZATION REQD. ○ TROPICALIZATION REQD.
SITE DATA:
○ ELEVATION ____ FT BAROMETER ____ (PSIA)
○ RANGE OF AMBIENT TEMPS: MIN/MAX ____ / ____ °F
○ RELATIVE HUMIDITY: % MAX/MIN ____ / ____
UNUSUAL CONDITIONS: ○ DUST ○ FUMES
○ OTHER ____
○ UTILITY CONDITIONS:
STEAM: DRIVERS HEATING
MIN ____ PSIG ____ °F ____ PSIG ____ °F
MAX ____ PSIG ____ °F ____ PSIG ____ °F
ELECTRICITY: DRIVERS HEATING CONTROL SHUTDOWN
VOLTAGE
HERTZ
PHASE
COOLING WATER:
TEMP. INLET ____ °F MAX RETURN ____ °F
PRESS NORM ____ (PSIG) DESIGN ____ (PSIG)

○ SITE AND UTILITY DATA (CONT'D)

COOLING WATER:
MIN RETURN ____ PSIG MAX ALLOW Δ P ____ (PSI)
WATER SOURCE ____
INSTRUMENT AIR: MAX/MIN PRESS ____ / ____ (PSIG)

○ LIQUID

○ TYPE OR NAME OF LIQUID ____
○ PUMPING TEMPERATURE
NORMAL ____ °F MAX ____ °F MIN ____ °F
○ SPECIFIC GRAVITY ____ @ MAX TEMP
○ SPECIFIC HEAT ____ Cp (BTU/LB °F)
○ VISCOSITY ____ (cP) @ ____ °F
○ MAX. VISCOSITY @ MIN. TEMP. ____ (cP)
○ CORROSIVE/EROSIVE AGENT ____
○ CHLORIDE CONCENTRATION (PPM) ____
○ H_2S CONCENTRATION (PPM) ____
LIQUIDS: (3.5.2.11) ○ TOXIC ○ FLAMMABLE ○ OTHER

□ PERFORMANCE

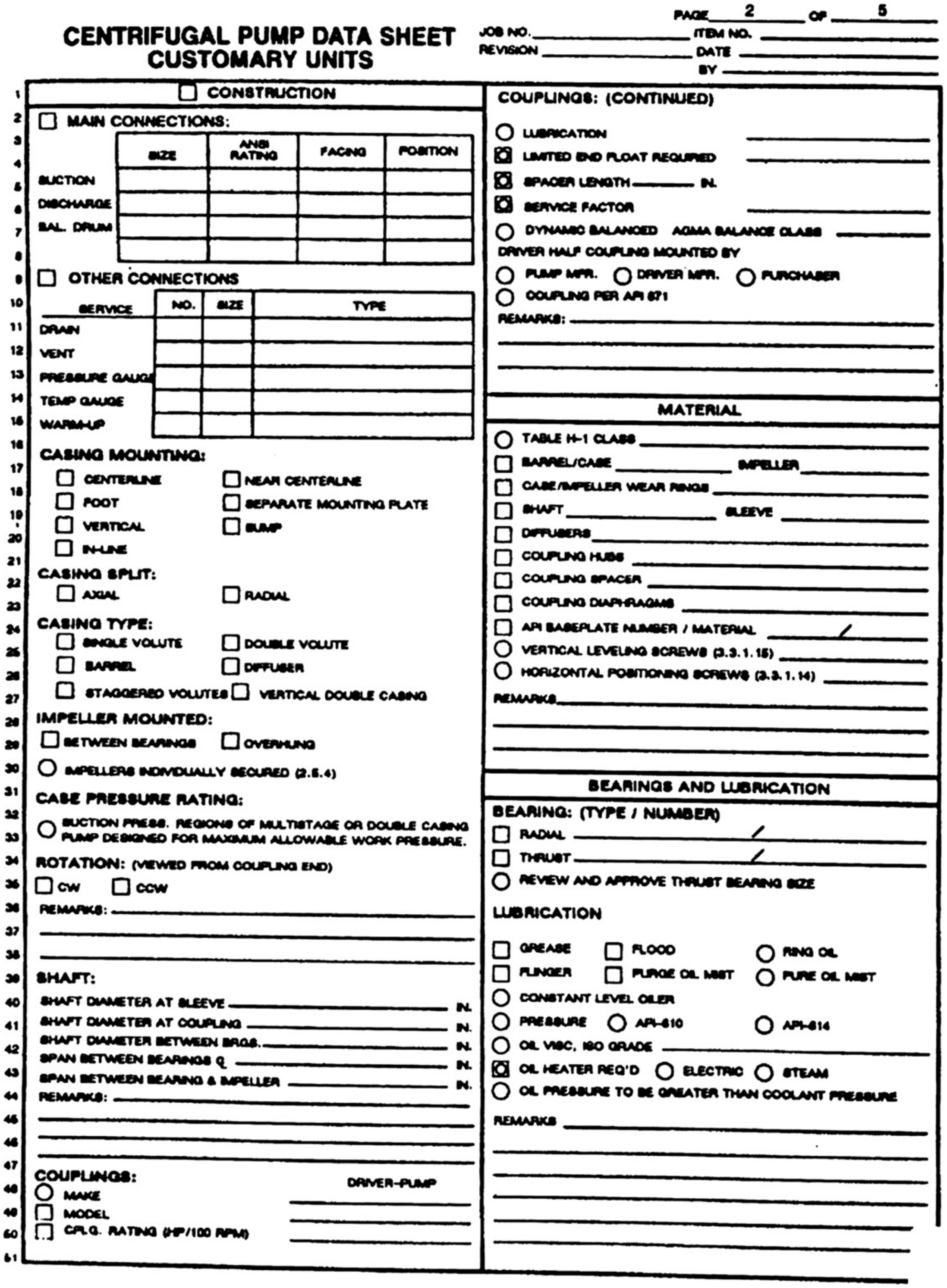
□ RPM ____
PROPOSAL CURVE NO. ____
□ IMPELLER DIA RATED ____ MAX ____ MIN ____ (IN)
□ RATED POWER ____ (BHP) EFFICIENCY ____ %
□ MINIMUM CONTINUOUS FLOW:
THERMAL ____ (GPM) STABLE ____ (GPM)
□ MAX HEAD RATED IMPELLER ____ (FT)
□ MAX POWER RATED IMPELLER ____ (BHP)
□ NPSH REQUIRED AT RATED CAP. ____ (FT H_2O)
□ SUCTION SPECIFIC SPEED ____
□ MAX SOUND PRESSURE LEVEL ____ dBA
REMARKS: ____

Fig 7.7.5a • Centrifugal pump data sheet (Courtesy of The American Petroleum Institute)

volume of a liquid to an equal volume of water at standard conditions) decreases, the energy (head) required to produce the same differential pressure increases proportionally. Therefore a positive displacement pump can meet the required energy increase at essentially the same flow, whereas a dynamic pump cannot. The only way for a dynamic pump to produce additional energy is to lower the flow rate (assuming the pump operates at a constant speed).

Therefore, increased system energy (head) requirements will force a centrifugal pump to a lower flow, or decreased system energy (head) requirements will cause it to deliver additional flow. In the case of bearing wear, a larger 'equivalent orifice' would reduce the pump's discharge system resistance, and result in increased flow rate. On the other hand, a positive displacement pump will essentially deliver the same flow in the above case. As we will see in the next chapter, both classifications of pumps present problems in terms of meeting auxiliary system objectives that will have to be solved using a reliable control scheme.

Auxiliary system pump applications involve both viscous (lubricating and sealing oils, etc.) and non-viscous fluids (water). The viscosity of a fluid is its tendency to resist a shearing force. It can be thought of as the internal friction that results when one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CENTRIFUGAL PUMP DATA SHEET
CUSTOMARY UNITS

JOB NO. ________ ITEM NO. ________
REVISION ________ DATE ________
BY ________

☐ CONSTRUCTION

☐ MAIN CONNECTIONS:

	SIZE	ANSI RATING	FACING	POSITION
SUCTION				
DISCHARGE				
BAL. DRUM				

☐ OTHER CONNECTIONS

SERVICE	NO.	SIZE	TYPE
DRAIN			
VENT			
PRESSURE GAUGE			
TEMP GAUGE			
WARM-UP			

CASING MOUNTING:
☐ CENTERLINE ☐ NEAR CENTERLINE
☐ FOOT ☐ SEPARATE MOUNTING PLATE
☐ VERTICAL ☐ SUMP
☐ IN-LINE

CASING SPLIT:
☐ AXIAL ☐ RADIAL

CASING TYPE:
☐ SINGLE VOLUTE ☐ DOUBLE VOLUTE
☐ BARREL ☐ DIFFUSER
☐ STAGGERED VOLUTES ☐ VERTICAL DOUBLE CASING

IMPELLER MOUNTED:
☐ BETWEEN BEARINGS ☐ OVERHUNG
○ IMPELLERS INDIVIDUALLY SECURED (2.5.4)

CASE PRESSURE RATING:
○ SUCTION PRESS. REGIONS OF MULTISTAGE OR DOUBLE CASING PUMP DESIGNED FOR MAXIMUM ALLOWABLE WORK PRESSURE.

ROTATION: (VIEWED FROM COUPLING END)
☐ CW ☐ CCW

REMARKS: ________

SHAFT:
SHAFT DIAMETER AT SLEEVE ________ IN.
SHAFT DIAMETER AT COUPLING ________ IN.
SHAFT DIAMETER BETWEEN BRGS. ________ IN.
SPAN BETWEEN BEARINGS ℄ ________ IN.
SPAN BETWEEN BEARING & IMPELLER ________ IN.
REMARKS: ________

COUPLINGS: DRIVER–PUMP
○ MAKE ________
☐ MODEL ________
☐ CPLG. RATING (HP/100 RPM) ________

COUPLINGS: (CONTINUED)
○ LUBRICATION ________
☐ LIMITED END FLOAT REQUIRED ________
☐ SPACER LENGTH ________ IN.
☐ SERVICE FACTOR ________
○ DYNAMIC BALANCED AGMA BALANCE CLASS ________
DRIVER HALF COUPLING MOUNTED BY
○ PUMP MFR. ○ DRIVER MFR. ○ PURCHASER
○ COUPLING PER API 671
REMARKS: ________

MATERIAL
○ TABLE H-1 CLASS ________
☐ BARREL/CASE ________ IMPELLER ________
☐ CASE/IMPELLER WEAR RINGS ________
☐ SHAFT ________ SLEEVE ________
☐ DIFFUSERS ________
☐ COUPLING HUBS ________
☐ COUPLING SPACER ________
☐ COUPLING DIAPHRAGMS ________
☐ API BASEPLATE NUMBER / MATERIAL ________ / ________
○ VERTICAL LEVELING SCREWS (3.3.1.15) ________
○ HORIZONTAL POSITIONING SCREWS (3.3.1.14) ________
REMARKS ________

BEARINGS AND LUBRICATION
BEARING: (TYPE / NUMBER)
☐ RADIAL ________ / ________
☐ THRUST ________ / ________
○ REVIEW AND APPROVE THRUST BEARING SIZE

LUBRICATION
☐ GREASE ☐ FLOOD ○ RING OIL
☐ FLINGER ☐ PURGE OIL MIST ○ PURE OIL MIST
○ CONSTANT LEVEL OILER
○ PRESSURE ○ API-610 ○ API-614
○ OIL VISC. ISO GRADE ________
☐ OIL HEATER REQ'D ○ ELECTRIC ○ STEAM
○ OIL PRESSURE TO BE GREATER THAN COOLANT PRESSURE
REMARKS ________

Fig 7.7.5b • Centrifugal pump data sheet (Courtesy of The American Petroleum Institute)

layer of fluid is made to move in relation to another layer. There are two basic viscosity parameters – absolute viscosity and kinematic viscosity.

Absolute viscosities are given in terms of force required to move a unit area a unit distance:

$$\frac{\text{lb force} - \text{seconds}}{\text{ft}^2}$$

The absolute viscosity metric system equivalent is a 'poise':

$$\frac{\text{dyne-seconds}}{\text{centimeter}^2}$$

Absolute viscosities are usually expressed in centipoises; 1/100 of a poise.

A. Pumped liquid data
- Type
- Pressure and temperature – normal, maximum, minimum
- Vapor pressure – normal, maximum, minimum
- Viscosity – normal, maximum, minimum
- Specific gravity

B. Pump suction and discharge pressure
- Suction pressure – normal, maximum, minimum (high viscosity condition)
- Discharge pressure
- Normal – clean system condition
- Maximum – dirty system condition

C. Pump flow data
- Pump rated flow ¼ critical equipment required flow plus 10–20% allowance for pump and critical equipment component wear.
- Pump normal flow ¼ critical equipment required flow

D. Pump net positive suction head available ($NPSH_A$)
- Confirm pump location relative to minimum operating level or reservoir (Note: If pump is above liquid level state the distance on the data sheet)
- Confirm $NPSH_A$ noted on data sheet is for the rated flow case and maximum suction pressure drop case (maximum fluid viscosity, i.e., minimum liquid temperature)

Fig 7.7.6 • Operating data checklist — pump data sheet

Most pipe friction and pump correction charts use kinematic viscosity.

Kinematic viscosities are expressed in:

$$\text{centistokes} = \frac{1}{100} \times \text{stoke} = \frac{\text{centimeter}^2}{\text{second}}$$

The other kinematic viscosity term frequently used is SSU or Saybolt Universal Seconds. 'SSU' are related to centistokes by:

$$\text{SSU} = \text{centistokes} \times 4.635$$

Kinematic viscosity is related to dynamic viscosity by:

$$\text{Kinematic viscosity (centistokes)} = \frac{\text{dynamic viscosity (centipoise)}}{\text{specific gravity}}$$

Applying the definition of viscosity to the definition of positive displacement and dynamic pumps, it can be seen that viscosity has little effect on positive displacement pump performance, since pressure is increased by operating on a fluid in a fixed space. However, dynamic pump performance is significantly affected by viscosity, since this type of pump increases the pressure by using rotary blades to increase fluid velocity. The shearing force required to move a viscous fluid will reduce the fluid velocity produced by the blades and will reduce head produced, flow and efficiency and result in a significant pump horsepower increase. Pump horsepower is determined by the relationship:

$$\text{Power} = \frac{\text{Flow} \times \text{Head} \times \text{Specific Gravity}}{\text{Constant} \times \text{Pump Efficiency}}$$

where:

Metric units
Power = KW
Flow = M^3/HR
Head = Meters
S.G = dimensionless
Constant = 3600
Pump Eff'y = decimal

Positive displacement – dynamic pump comparison

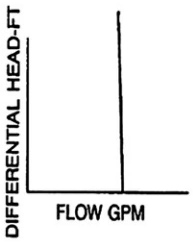
	Positive displacement	Dynamic
Definition	Increases pressure by operating on fixed volume in a confined space	Increases pressure by using rotary blades to increase fluid velocity
Types	Screw, gear, reciprocating	Centrifugal, axial
Characteristics	■ Constant volume ■ Variable differential head ■ Relatively insensitive to liquid properties ■ Relatively insensitive to system changes ■ Not self-limiting	■ Variable volume ■ Constant differential head ■ Sensitive to liquid properties ■ Sensitive to system changes ■ Self-limiting
Characteristic flow vs. differential head curves		

Fig 7.7.7 • Positive displacement — dynamic pump comparison

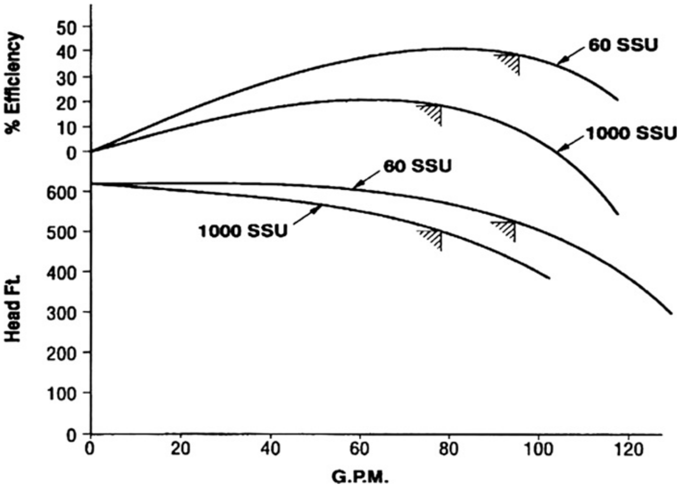


Fig 7.7.8 • The effect of oil viscosity on centrifugal pump performance

Customery units
Power = Horse Power
Flow = GPM
Head = FT
SG = dimensionless
Constant = 3960
Pump Eff'y = decimal

Figure 7.7.8 is a sample performance curve for a centrifugal pump showing the effect of operating the same pump on water and 220 centistokes (1,000 SSU) oil.

The performance for oil was calculated using the hydraulic institute viscosity correction procedure. As a result, positive displacement pumps are usually used for auxiliary systems containing oil, while centrifugal pumps are always used for large auxiliary systems containing non-viscous (water-based) liquids.

When determining if a centrifugal or positive displacement pump should be used in an auxiliary system containing oil, the factors noted in Figure 7.7.9 should be considered.

- Compare pump types of equal reliability.
- Compare yearly operating costs based on normal (operating) liquid viscosity case.
- Compare driver and coupling size and costs for maximum liquid viscosity case.
- Determine critical equipment minimum and maximum flow requirements and compare control system complexity and cost.

Fig 7.7.9 • Auxiliary system pump type determination factors positive displacement vs. dynamic

Sizing examples

The next step is to properly select a pump for the auxiliary system. To demonstrate this procedure, both a positive displacement and centrifugal pump will be selected and compared.

The following data is available:

Pumped fluid data

Type – light mineral oil – 150 SSU @ 100°F

Conditions	Minimum	Normal	Maximum
Pressure (PSIG)	0.63	0.75	1.22
Temperature (°F)	40°F	150°	160°F
Specific gravity	0.85	0.85	0.85
Viscosity (SSU)	1,000	60	57
Vapor pressure (PSIA)	*0.001*	*0.001*	*0.001*

Pump suction and discharge pressures

	Min	Norm	Max
Suction pressure (PSIG)	0.63	0.75	1.22
Discharge pressure (PSIG)	160	160	200*

*Maximum discharge pressure for a positive displacement type pump = maximum system discharge pressure plus relief valve accumulation.

Pump flow data

- Pump rated flow = 75 GPM
- Pump normal flow = 60 GPM

$NPSH_A$

Location of pump relative to liquid level = below level

$NPSH_A$ at minimum oil temperature and reservoir level at minimum level = 41.695 feet (15.33 PSIA)

$$NPSH_A(\text{ft}) = \frac{2.311 \times 14.7 + 2\ \text{ft}}{.85}$$

(minimum oil height above – pump suction)

$$\frac{2.311 \times .001}{.85}(\text{vapor pressure})$$

$$-\frac{2.311 \times .1}{.85}(\text{pipe pressure drop})$$

$$= 41.695\ \text{feet}$$

Having obtained the above data, pump selections can now be made. It can be seen from the above example that it is very unusual for an oil system to have insufficient NPSH available. Therefore, when cavitation noise is detected it is usually the result of air entrainment.

Positive displacement pump sizing example

A screw-type pump similar to Figure 7.7.1 will be used. Pump speeds are usually based on site electrical frequency (50 Hz or 60 Hz) and are selected for a two-pole motor nominal speed for reasons of pump driver efficiency.

Assuming a 60 Hz electrical supply, a nominal speed of 3,600 RPM will be selected. Since positive displacement pumps are constant flow devices, a pump capacity equal to or greater than the rated flow will be selected. As will be explained later, selecting too large a pump can reduce system reliability. A typical screw pump selection chart is shown in Figure 7.7.10

3D Rotor Size 187

Speed 3500 RPM

	Viscosity SSU	Differential Pressure—PSI 25	75	150	200	350	500	Net Inlet Pressure Required PSIA
GPM	33	82.3	77.3	72.7	–	–	–	
	40	82.9	78.4	74.1	71.7	–	–	
	60	84.0	80.4	76.9	74.9	70.3	66.8	
	100	85.2	82.3	79.6	78.1	74.5	71.7	
	150	85.9	83.6	81.4	80.1	77.2	74.9	
	500	87.3	86.1	84.8	84.1	82.6	81.8	
	1,000	87.8	86.9	86.1	85.6	84.5	83.6	
	2,000	88.2	87.6	86.9	86.6	85.8	85.2	
BHP	150	3.4	6.0	9.9	12.5	20.3	28.1	8.7
	500	5.7	8.3	12.2	14.8	22.6	30.4	9.8
	1,000	8.1	10.7	14.6	17.2	25.0	32.8	11.6
	2,000	11.8	14.4	18.3	20.9	28.7	36.4	17.1

Speed 1750 RPM

	Viscosity SSU	Differential Pressure—PSI 25	75	150	200	350	500	Net Inlet Pressure Required PSIA
GPM	33	37.7	32.8	28.2	–	–	–	
	40	38.4	33.9	29.6	27.2	–	–	
	60	39.5	35.9	32.3	30.4	25.8	21.2	
	100	40.6	37.8	35.1	33.6	30.0	27.2	
	150	41.4	39.0	36.8	35.6	32.7	30.4	
	500	42.8	41.5	40.3	39.6	38.0	36.8	
	1,000	43.8	42.4	41.6	41.1	39.9	39.0	
	20,000	44.3	44.1	43.9	43.8	43.5	43.3	
BHP	150	1.2	2.5	4.4	5.7	9.6	13.5	4.8
	500	1.8	3.1	5.0	6.3	10.2	14.1	4.9
	1,000	2.4	3.7	5.6	6.9	10.8	14.7	5.1
	2,000	3.3	4.6	6.7	7.9	11.8	15.7	5.5
	5,000	5.3	6.6	8.6	9.9	13.8	17.7	6.6
	7,000	6.4	7.7	9.7	11.0	14.9	18.8	7.4
	10,000	7.9	9.2	11.1	12.4	16.3	20.2	8.5
	20,000	11.7	13.0	15.0	18.3	20.2	24.1	15.6

3D Rotor Size 218

Speed 3500 RPM

	Viscosity SSU	Differential Pressure—PSI 25	75	150	200	350	500	Net Inlet Pressure Required PSIA
GPM	33	132	126	117	–	–	–	
	40	133	127	120	117	–	–	
	60	134	129	125	122	115	111	
	100	136	132	128	126	121	117	
	150	137	134	130	129	125	122	
	500	139	137	135	134	132	130	
	1,000	139	138	137	136	135	134	
BHP	150	5.4	9.5	15.7	19.8	32.2	44.6	12.0
	500	9.1	13.2	19.5	23.6	35.9	48.3	14.7

Speed 1750 RPM

	Viscosity SSU	Differential Pressure—PSI 25	75	150	200	350	500	Net Inlet Pressure Required PSIA
GPM	33	61.5	54.7	47.6	–	–	–	
	40	62.4	56.3	50.4	47.1	–	–	
	60	63.9	58.9	54.1	51.4	45.2	39.0	
	100	65.4	61.6	57.9	55.8	51.0	47.1	
	150	66.4	63.2	60.2	58.5	54.6	51.4	
	500	68.3	66.6	64.9	64.0	61.9	60.2	
	1,000	69.0	67.8	66.7	66.0	64.5	63.2	
	10,000	70.2	69.8	69.4	69.2	68.7	68.3	
BHP	150	1.9	3.9	7.0	9.1	15.3	21.5	5.2
	500	2.8	4.9	8.0	10.0	16.2	22.4	5.4
	1,000	3.8	5.9	8.9	11.0	17.2	23.4	5.7
	2,000	5.3	7.3	10.4	12.5	18.7	24.9	6.2
	5,000	8.5	10.6	13.6	15.7	21.9	28.1	7.9
	7,000	10.2	12.3	15.3	17.4	23.6	29.8	9.4
	10,000	12.5	14.5	17.6	19.7	25.9	32.1	11.6

Fig 7.7.10 • Screw pump sizing data (Courtesy of IMO Industries)

and the corresponding pump performance curve is shown in Figure 7.7.11.

From this data, the following values of pump performance are obtained:

Item	Temperature		
Condition	**Min (40°F)**	**Normal (150°F)**	**Maximum (160°F)**
Corresponding viscosity (SSU)	1,000	60	57
Actual pump flow (GPM) @ normal pressure of 160 PSIG	86.3	77	77
Actual pump flow @ maximum pressure of 250 PSIG (pump R.V. setting + accumulation)	83.3	73	73
Pump horsepower Normal pressure (160 PSIG) Normal viscosity (60 SSU)	15.0	9	9
Maximum pressure 255 PSIG Maximum viscosity	19.66	14.8	14.8
NPSH required (PSIA)	11.6	8.5	8.5

The following facts can be observed from the above selection:

- Since the positive displacement pump (screw pump) is essentially a constant volume device, the pump is selected for the rated flow case (maximum flow). Normal flow operation will be attained by bypassing excess flow.
- Maximum pump horsepower occurs at the 'cold' condition, where oil viscosity and system pressure drop will be maximum. Therefore the pump relief valve setting plus relief valve accumulation is used.
- The NPSH required is relatively constant but varies with viscosity and is less than the NPSH available in the system.

Before proceeding to a centrifugal pump selection, the following comments are made concerning positive displacement pump reliability.

- *Rotor Contact* – The majority of screw and gear pumps used in lubrication service rely on the pumped fluid to separate the pump rotating elements and lubricate the pump bearings. As a result, attention must be paid to selecting these pumps within the specified pump vendor limits of pumped fluid pressure and temperature, and to designing a system that ensures these conditions are maintained. Some users use a safety factor of 80–90% from the vendors maximum pressure for a given pump and viscosity (240–270 PSI in Figure 7.7.10) to ensure reliable operation. Pumps equipped with external timing gears will not require this consideration since the timing gears prevent the rotors from coming into contact.

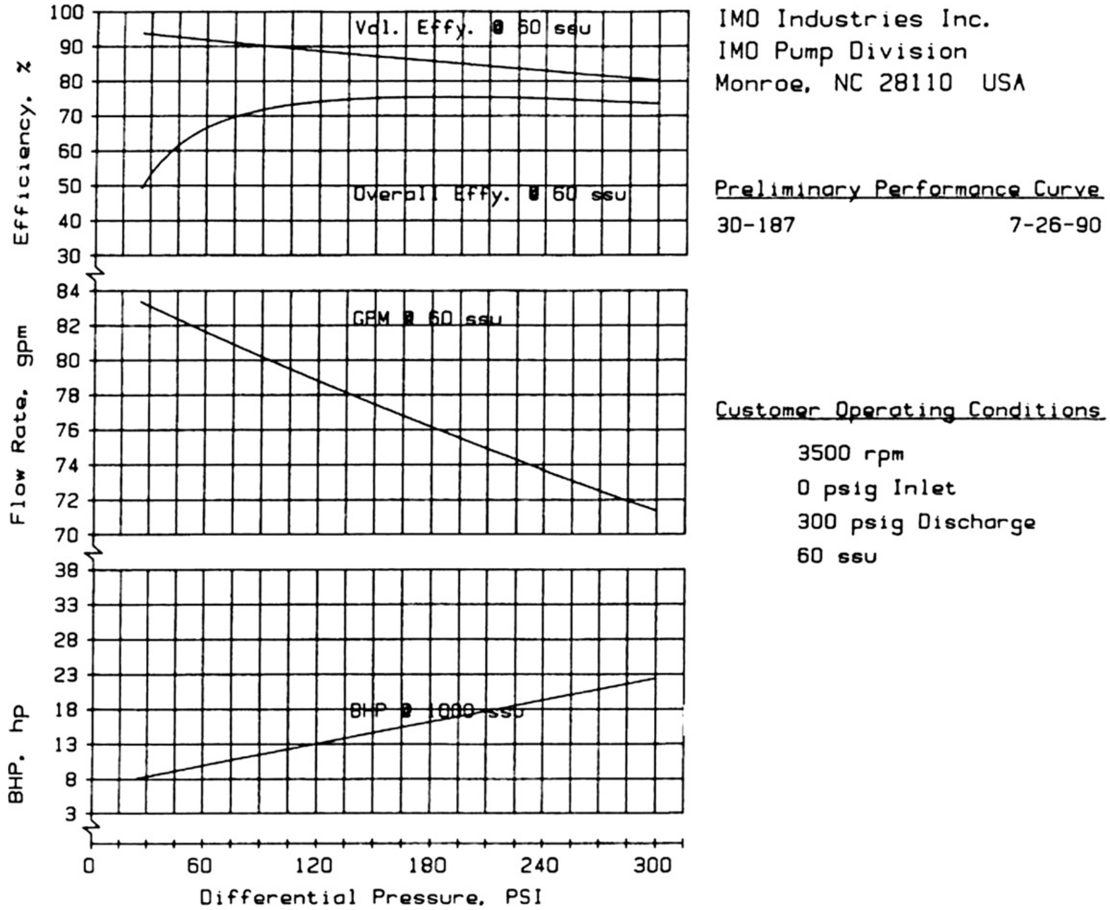


Fig 7.7.11 • Screw pump performance (Courtesy of IMO Industries)

- *Suction Conditions* – The pump must be assured of liquid flow at minimum acceptable conditions during operation. If strainers are installed in the pump suction a low suction pressure alarm switch is recommended downstream of the strainer.

The pump must always move the fluid in its liquid state. To ensure this, the net positive suction system head available ($NPSH_A$) must exceed the net positive suction head required ($NPSH_R$) by the pump. The $NPSH_R$ of any pump is simply the internal pump pressure drop from the pump suction flange to the pumping element (screw, gear impeller).

The net positive suction head available ($NPSH_A$) is the absolute pressure at the specified suction fluid conditions above the liquid vapor pressure. Any liquid will change to a vapor when its surrounding pressure is less than its vapor pressure. Therefore, if the pressure drop internal to a pump ($NPSH_R$) equals or exceeds the available pressure surrounding the liquid, the liquid will become a vapor, so causing pump cavitation. Continued operation under these conditions will cause loss of pump lubrication, pump damage and possible failure, since the original incompressible fluid (liquid) supporting the rotors is now compressible, and cannot support the load required to separate the rotors.

The same damage can occur from entrained gas (process gas in seal oil systems or air) in the pumped liquid. Sources are:

- Insufficient degassing in the reservoir
- Air carried into the system by pump suction system leaks and pump seal leaks, since most auxiliary system pumps operate under a slight vacuum on the suction side.

The effect will be the same as cavitation, since the suction end of the pump will be operating on a compressible fluid before it is compressed by the energy available in the pump. In both cases, potential damage can best be detected by the high-pitched sound or whine emitted. Recommended action concerning pump cavitation and aeration is noted in Figure 7.7.12.

Fluid cleanliness

Pump life is definitely related to pumped liquid contamination level. Simply put, the cleaner the system, the longer the pumps will last. Attention is drawn to the practice of using pump strainers. It is our opinion that an effective auxiliary system and component ensurance program as detailed in this section proves to be a more reliable alternative to ensure pump reliability than the installation of pump suction strainers or filters.

Symptom	Action
1. Pump noisy, excessive wear (since first day of operation)	**A.** Confirm $NPSH_A > NPSH_R$ **B.** Calculate $NPSH_A$ **C.** Obtain $NPSH_R$ from pump vendor for actual operating conditions If $NPSH_A > NPSH_R$, proceed to Step 2 If $NPSH_R > NPSH_A$: ■ Raise liquid level to correct problem Or ■ Lower pump relative to liquid level Or ■ Change to pump with acceptable $NPSH_R$
2. Pump noisy, possible pump wear (problem occurs intermittently or has recently occurred)	**A.** Tape or coat pump suction piping with heavy oil if sound stops, correct suction piping leak. **B.** Inspect pump seal at first opportunity for signs of wear in both primary and secondary (shaft sleeve 'O' ring or packing) seal **C.** Check reservoir for entrained air or gasses Caution – obtain work permit prior to obtaining sample. If gasses or air are in liquid (cloudy sample – not clear), correct problem by proper degassing, reservoir modification or drain pipe modification. NOTE: This is a difficult problem and can require extensive modifications.

Fig 7.7.12 • Pump cavitation and aeration checklist

Centrifugal pump sizing example

A centrifugal type pump similar to Figure 7.7.3 will be used. Data available is the same as the previous pump application as noted on previous page. The same comments apply as previously stated concerning pump speed. Therefore a nominal pump speed of 3,600 RPM has been selected.

Since a dynamic pump is a variable capacity device, a selection curve, as opposed to a chart (utilized for a positive displacement pump) is used. In addition, the pump must be oversized to account for the effects of oil viscosity as previously explained (see Figure 7.7.8). Therefore, the pump must be selected to produce the performance requirements noted on previous page at the 'worst case' – highest viscosity conditions. In order to select the proper pump the following steps are required:

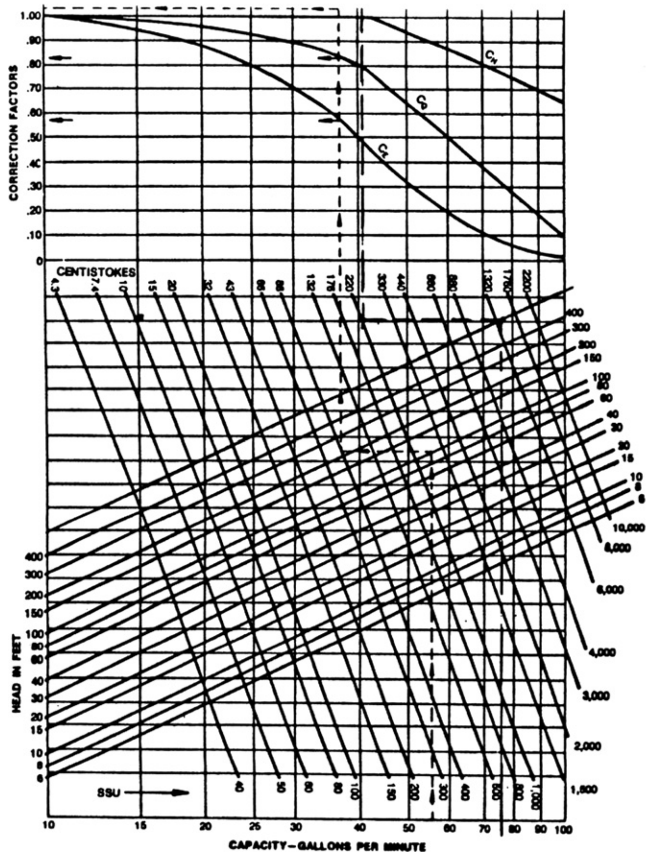


Fig 7.7.13 • Hydraulic institute viscosity correction factors

1. *Determine 'uncorrected' rated point*

$$\text{T.D.H. (Total differential head)} = \frac{2.311 \times (P_2 - P_1)}{\text{Specific gravity}}$$

$$\text{From data on previous page} = \frac{2.311 \times 198.8^*}{.85}$$

$$= 540.5 \text{ feet}$$

$$\text{Q rated} = 75 \text{ GPM}$$

*A discharge pressure of 200 PSIG was used since a relief valve is not required with a dynamic pump because it is self-limiting.

2. *Determine the viscosity corrected head and rated flow required for the maximum viscosity to be pumped*

Given:

Q rated uncorrected	=	75 GPM
Head rated uncorrected	=	540.5 feet
Maximum pumped viscosity	=	1,000 SSU

Referring to viscosity correction factors as published by the Hydraulic Institute standards – 12th edition, copyright 1969 (see Figure 7.7.13).

Q correction factor	=	0.80
Head correction factor	=	1.00
Efficiency correction factor	=	0.5

For the rated point.

As a first move, for selecting a properly sized pump, one can use the corrected valves of Q and head above to determine the approximate size pump impeller required.

$$Q_{selection} = \frac{\text{Qrated}}{\text{Q corr. factor}}$$

$$Q_{selection} = \frac{75 \text{ GPM}}{0.8} = 93.75 \text{ GPM}$$

$$HD_{selection} = \frac{\text{HD rated}}{\text{HD. corr factor}}$$

$$HD_{selection} = \frac{540.5}{1.0} = 540.5 \text{ feet}$$

A selection for a single stage centrifugal pump operating at 3,550 RPM with a 2" inlet, 1" discharge and 11" diameter impeller is made. The curve is noted in Figure 7.7.8 and shows the following pump operating points.

- Rated flow @ maximum viscosity (1,000 SSU)
- Normal flow @ maximum viscosity (1,000 SSU)
- Rated flow @ normal viscosity (60 SSU)
- Normal flow @ normal viscosity (60 SSU)

The following facts can be observed from Figure 7.7.10:

A. All operating points other than the rated point at 1,000 SSU viscosity exceed the head required of 540.0 feet.
B. A specific differential head value exists for each pump flow and liquid viscosity.
C. Pump efficiency is low for this relatively small centrifugal pump and is reduced approximately 50% for the high viscosity case.

The pump horsepower is calculated by (if a viscosity correction is not required):

$$\text{BHP} = \frac{\text{Head(ft)} \times \text{flow(GPM)} \times \text{specific gravity}}{3960 \times \text{Pump effciency(\%)}}$$

If a viscosity correction is required:

$$\text{BHP} = \frac{\text{(head)} \times \text{(head viscosity corr.)} \times \text{(flow)} \times \text{(flow viscosity corr.)} \times \text{specific gravity}}{(3960) \times \text{(Pump efficiency)} \times \text{(Pump efficiency vicosity corr.)}}$$

For the maximum viscosity condition:

$$\text{BHP} = \frac{(540.5) \times (93.75) \times (0.8) \times (.85)}{(3960)(.38) \times (.5)}$$
$$= 46.0 \text{ BHP}$$

For the normal viscosity condition:

$$\text{BHP} = \frac{(540.5) \times (75)\ (.85)}{(3960)\ (0.38)}$$
$$= 23.0 \text{ BHP}$$

The pump NPSH required is obtained from the pump curve.

	Positive displacement	Dynamic
Rated flow		
60 SSU (Normal)	77 GPM	93.3
1000 SSU (Max.)	83.3 GPM	75 GPM
Discharge pressure		
60 SSU (Normal)	160	200
1000 SSU (Max.)	250	200
Pump horsepower		
60 SSU (Normal)	14.8 BHP	23 BHP
1000 SSU (Max.)	19.66 BHP	46 BHP

Fig 7.7.14 • Pump performance positive displacement vs. dynamic lube oil application

A comparison of positive displacement vs. dynamic pump performance for this application is noted in Figure 7.7.14 for normal and rated points at normal and maximum viscosities.

The following facts should be noted:

A. The positive displacement pump meets the required discharge pressure at greater flows than specified.
B. The dynamic pump meets the required flow at greater discharge pressures than specified.
C. The efficiency for a centrifugal pump at high viscosities and this relatively low flow rate is significantly lower than the positive displacement pump. Note that for high flow rate applications the efficiency of a dynamic pump can actually exceed a P.D. pump.
D. The required horsepower of a centrifugal pump for this application is far greater than the positive displacement pump.

The preceding example demonstrates why positive displacement pumps are usually favored over dynamic pumps in relatively low flow systems using viscous liquids. However, if the liquid viscosity does not vary significantly, and the system flow requirements are large, centrifugal pumps provide efficient, low maintenance service and are usually used.

It must be remembered that dynamic pumps, like positive displacement pumps, must be provided with the correct suction conditions to ensure reliable operation. Cavitation, entrained gases and low flow rates can cause impeller damage and wear that will reduce pressure output and result in lower system flow rates.

The types of pumps used for our example are only two of the types used for auxiliary system service. Other types of pumps that can be used for auxiliary systems are:

- Gear type (external or internal gear)
- Twin screw
- Lobe type
- Sliding vane

Before concluding this section on auxiliary system pumps, the subject of component redundancy must be discussed. In order to satisfy the design objective of '*continuously supplying…*' the pumps, as well as all the major components of the system

subject to wear should be redundant. Critical equipment systems incorporate stand-by pumps. Usually, the main and spare pumps are identical in design. This is particularly true if dynamic pumps are used, since their performance curves should be similar to ensure trouble free operation.

There are exceptions; some systems use a main, shaft-driven centrifugal pump and a positive displacement stand-by pump. In this case, the positive displacement pump must be controlled such that its maximum discharge pressure does not force the centrifugal pump to a low or no flow operating point, when both pumps are operating.

The stand-by pump is combined with a control system that will automatically start the pump on an abnormal system condition (low flow or low pressure). The stand-by pump unit (pump, coupling and driver) must be designed for fast, reliable start-up and operation on demand.

Best Practice 7.8

Install differential pressure transmitters to alarm on high differential, for control room monitoring, around pump suction strainers – especially screw and gear pumps.

Since only the oil film keeps gear and screw components from contacting each other, a plugged main pump suction strainer will rapidly increase pump clearance and cause the auxiliary pump to start.

The source of the main pump strainer blockage will eventually plug the auxiliary strainer and result in auxiliary pump damage.

Lessons Learned

Failure to detect rotary pump (screw or gear) strainer blockage has resulted in excessive pump wear that has caused critical machinery trips when main pump wear caused the auxiliary pump to start too late.

Benchmarks

This best practice has been used since 1990 to ensure that rotary pump wear can be prevented, which has resulted in optimum oil console reliability and critical machinery train reliability above 99.7%.

B.P. 7.8. Supporting Material

This best practice is rarely used by project teams, but can easily impact critical equipment reliability. The additional cost of adding suction strainer alarms is small compared to the potential reliability reduction and loss of revenue.

Best Practice 7.9

Consider motor drives or shaft-driven and not steam turbine main pump drivers if the electrical system is reliable.

Small steam turbine reliability is the lowest of all the lube oil pump driver alternatives.

If the electrical system is reliable, dual motor drives operating from different electrical systems should be a favored alternative or, if the electrical system is not totally reliable, a shaft-driven main pump is also a viable alternative.

Lessons Learned

Numerous case histories have shown that steam turbine, main pump drivers have the lowest reliability of any oil pump driver alternative, since they cause critical machinery unit trips.

Benchmarks

This best practice has been used for projects and field modifications since the mid-1990s to ensure the highest critical machinery reliability.

B.P. 7.9. Supporting Material

Types of lubrication system

The classical lube system arrangement consists of a steam turbine-driven main pump and motor drive auxiliary pump. In this section, we will devote our attention to other system variations.

Shaft-driven, positive displacement main pump; motor-driven auxiliary pump

Refer to Figure 7.9.1. In this system, a shaft-driven main pump is supplied. The function of having a shaft-driven pump is to continuously provide some amount of lubrication to the system bearings while the critical equipment is in operation. The idea

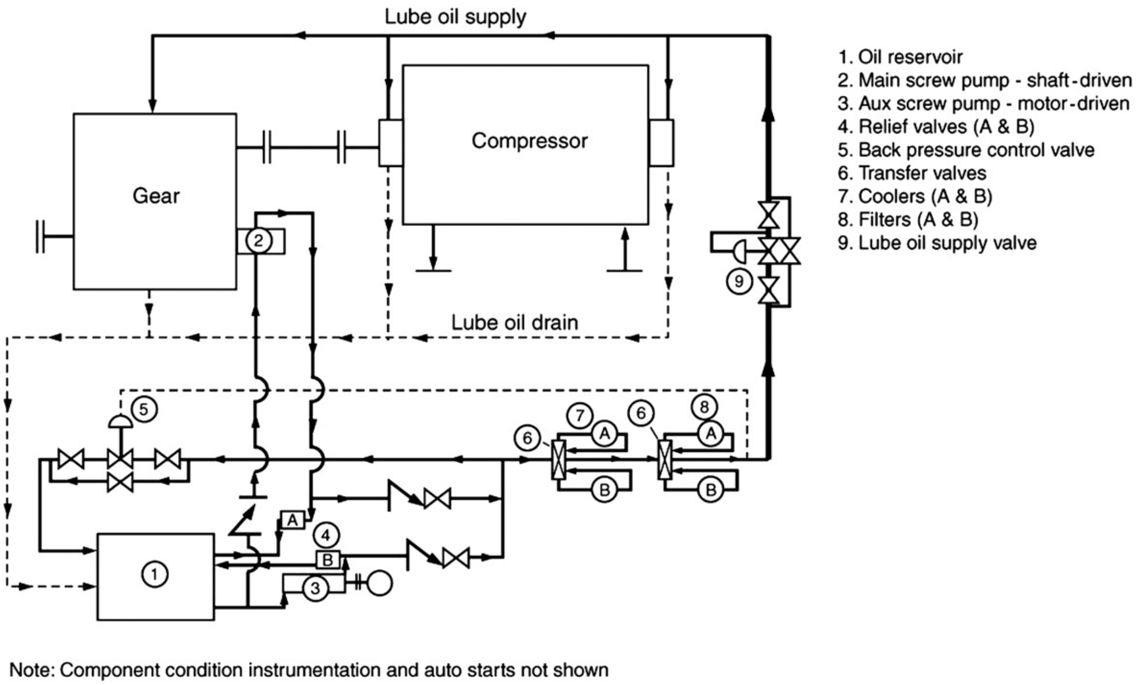


Fig 7.9.1 • Lube oil system main pump shaft-driven (Courtesy of Elliott Co.)

being that, on coastdown, or start-up, the system would never be without lubrication oil supply, even in the event of failure of the stand-by pump. Such a system is useful in areas where an alternative form of energy is not available for main and spare pumps, as on platforms or in chemical plants where a utility steam system is not installed.

The system must still be designed to effectively lubricate all bearings before the main pump attains full flow rate (speed). To facilitate this, the auxiliary pump is designed to start before the main critical equipment driver is started (permissive arrangement) and either automatically or manually shut off once the shaft-driven pump attains sufficient speed. On shutdown, the reverse occurs. The auxiliary pump will start as the main pump decelerates. The signal to start the auxiliary pump can be critical equipment speed or system pressure. If a system is not designed for automatic shutoff of the auxiliary pump, care must be taken to ensure that the system is designed for continuous two-pump operation. It is recommended that auxiliary pumps be shut off during critical equipment main pump operation.

A particular concern with this design is to ensure priming of the main pump. Frequently, this is at a significant height above the fluid reservoir. In this case, care must be taken to ensure proper priming of main pump prior to start-up. Many systems incorporate a priming line from the auxiliary pump to fill the main pump suction line. This requires a check valve (foot valve) in the main pump suction line and a self-venting device to ensure that air is not entrained in the suction line of the main pump. Frequently, systems of this type do not incorporate a vent line and are susceptible to main pump cavitation on start-up.

Referring to Figure 7.9.1, let's examine the response of the bypass valve in the event that the auxiliary pump does not shut off after the shaft-driven pump has attained full flow (speed). Upon initial auxiliary pump start-up, flow is supplied to the system, and the bypass valve opens to control system pressure to a preset value, thereby bypassing excessive auxiliary pump (stand-by pump) flow. Upon starting of the critical equipment unit, the bypass valve will react to increasing main pump flow and will gradually open to the point that it would be at maximum stroke if the stand-by pump remained in operation. When the stand-by pump is shut down, the response of the bypass valve must be equal to the decrease of flow from the auxiliary stand-by pump such that system pressure does not drop below trip setting.

Refer back to the concepts of an equivalent vessel and orifice discussed in previous sections. In this case, the supply to the equivalent vessel (from the auxiliary pump) instantaneously drops while the demand from the critical equipment is constant thus causing an instantaneous drop in equipment vessel pressure. Prior to shutdown of the auxiliary pump, excessive supply was re-circulated by the bypass valve. Upon rapid decrease of supply flow, the bypass valve must decrease demand at the same rate to ensure that pressure in the system (equivalent vessel) is maintained at a constant value. Failure to do this can result in critical equipment shutdown.

Main and auxiliary AC motor-driven pumps, DC motor-driven emergency pump

Refer to Figure 7.9.2. This application would be used in facilities where steam systems are not available, such as chemical plants, or in operations at remote locations, or on offshore platforms. In this case, both the main and stand-by pumps would shut down in the event of a power failure. If the critical equipment unit is also motor driven, it too will cease operation.

The design objective of this auxiliary system is to provide a sufficient flow, via an emergency pump driven by a DC (direct

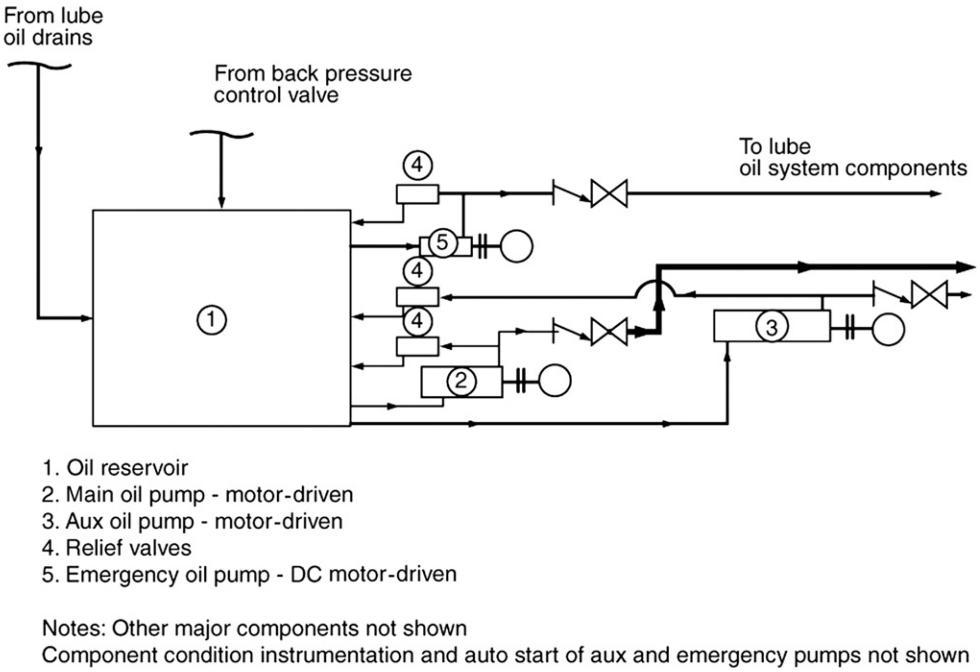


Fig 7.9.2 • Lube oil system main pump shaft-driven (Courtesy of Elliott Co.)

current) source, for enough time to promote the safe coastdown of the critical equipment unit. The DC power source can be obtained from a UPS (uninterrupted power supply) system or other source.

Coastdown times of the driven equipment vary between 30 seconds to in excess of 4 minutes. Sufficient emergency pump energy must be available. The emergency pump would be activated in this case on auxiliary system pressure drop. Note that the emergency pump is not a full capacity pump – it is sized only to provide sufficient flow during equipment coastdown to prevent auxiliary system component damage.

AC motor-driven main pump, steam turbine driven auxiliary pump

Refer to Figure 7.9.3 for a diagram of this system. A system of this type would be used in a plant where dual pump driver power is available, but steam is at a premium. Therefore, the turbine driver is not designed to be used for continuous operation. This system requires continuous attention in terms of steam turbine steam conditions. The turbine must be capable of accelerating rapidly since a system accumulator is not included. Therefore, steam conditions at the turbine flange must be maintained as specified. That is, steam must be dry. A steam trap is recommended to be installed to continuously drain condensate from the steam. To ensure rapid turbine acceleration, the turbine inlet valve must be a snap-open type (less than 1 second full open time) to minimize auxiliary pump start-up time. Experience has shown that this type of system is not reliable in terms of auxiliary pump starts. If used, it should employ a large accumulator to provide sufficient flow to the system for approximately ten seconds as a minimum since the start-up time of a steam turbine is much slower than that of a motor-driven pump.

Centrifugal pump system

Figure 7.9.4 depicts a dual centrifugal pump system. The main pump is steam turbine-driven and the stand-by pump is motor-driven. Such a system can be used in areas of the world where large ambient temperature fluctuations are not present, such as the Middle East. Since the pumps are the dynamic type, note that a bypass valve is not used in this system. This is because the flow rate of dynamic pumps is determined by system resistance, therefore, the control valve in this application senses pressure in the system and adjusts system resistance at the pump discharge to supply the required flow.

Referring back to the concept of an equivalent vessel, let us examine the case of increased system demand – as in the case of bearing wear. Bearing wear will gradually increase system demand and reduce system equivalent vessel pressure. Consider in this case the system to be the equivalent vessel. Since the control valve senses equivalent vessel pressure, a reduction in the equivalent vessel pressure will open the valve to provide greater pressure at the sense point. This action will in turn reduce pump discharge pressure. Referring to the characteristic curve of a centrifugal pump, reduced pump discharge pressure results in increased pump output flow, thereby compensating for the increased demand requirement.

In the event of a sudden system flow increase, a sudden change in the equivalent vessel pressure would be experienced. This would result in a sudden drop of equivalent vessel pressure, resulting in a sudden opening of the control valve. However, in this case, the sudden pressure drop would initiate starting of the stand-by pump. As soon as the stand-by pump was to start, supply flow to the equivalent vessel would quickly increase, causing a corresponding increase in pressure. However, the control valve sense point would note the pressure increase and rapidly shut the control valve, thus increasing the system resistance to both pumps, and forcing pumps to a lower flow rate.

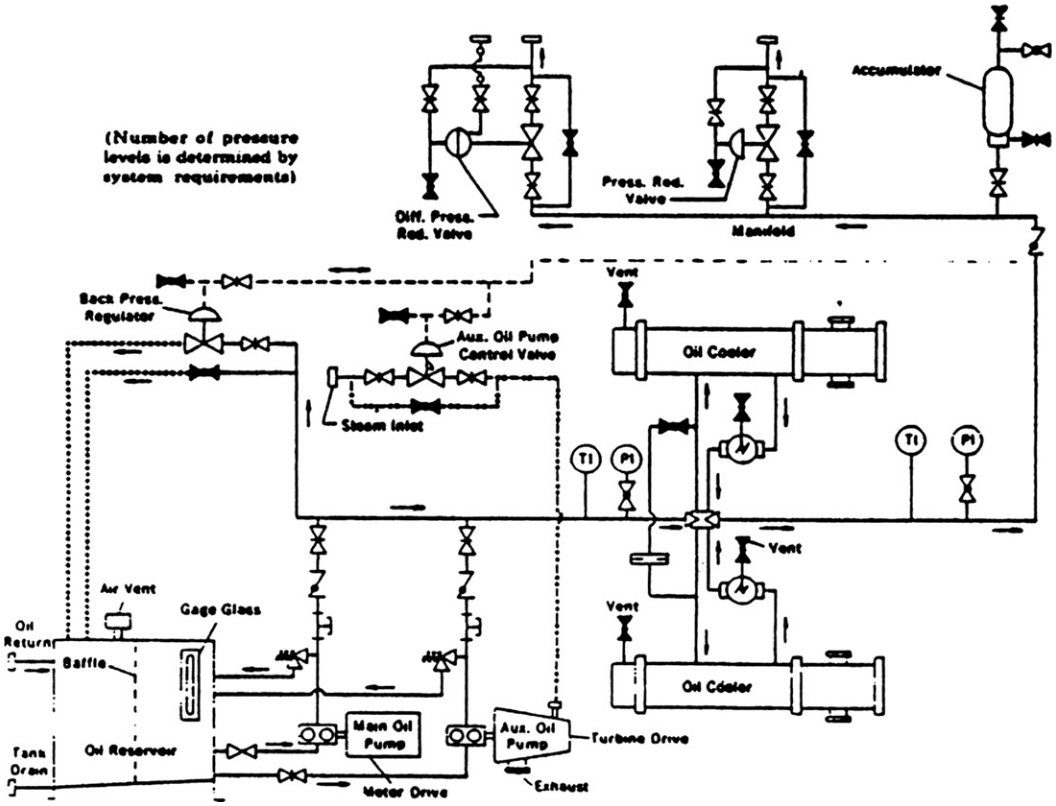


Fig 7.9.3 • A.C. motor-driven main pump, steam turbine auxiliary pump drive (Courtesy of Elliott Co.)

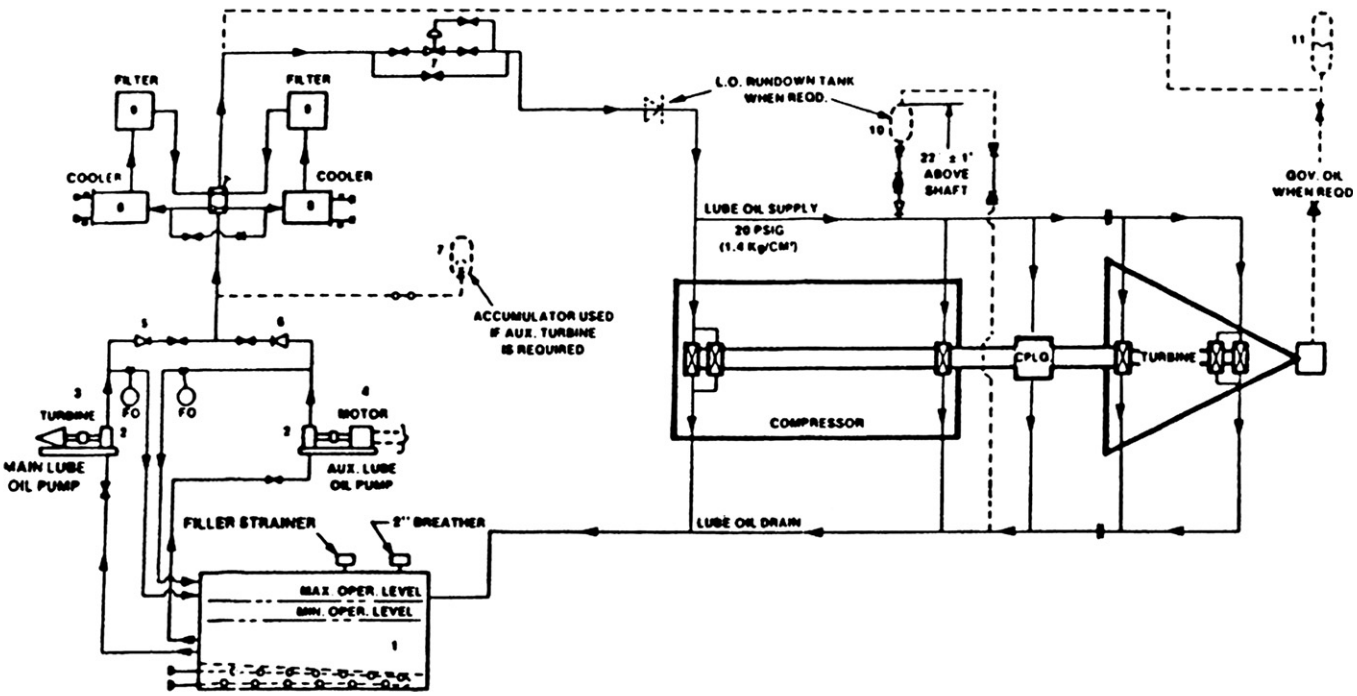


Fig 7.9.4 • Lubrication system – centrifugal pumps (Courtesy of Dresser-Rand)

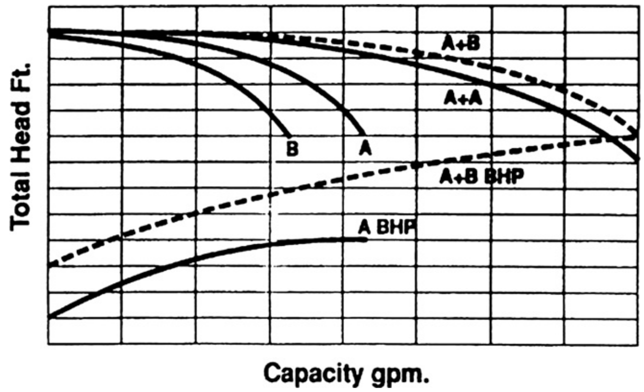


Fig 7.9.5 • Non-identical pumps with stable head curves

An important consideration in this system is that both main and auxiliary pumps operate at essentially the same speed and have essentially the same characteristic curves. (This would not be the case if one pump had excessively worn internals.) Therefore, the total output flow of the pumps would be a result of the combined curve of the pump output. Refer to Figure 7.9.5 for a view of the combined effect of both pumps operating in parallel. From this example, the importance of ensuring correct similar characteristics of both main and auxiliary pumps in systems incorporating dynamic pumps can be seen. If one pump is steam turbine-driven, it is important to periodically check speed, and correct the turbine governor setting if necessary. The motor and turbine speeds should be the same.

Dual motor-driven positive displacement pump system with a rundown tank

This system is similar to that shown in Figure 7.9.2 with the exception of the absence of an emergency pump. In this case, the user has elected to have a rundown tank in the system that will supply sufficient lube oil for rundown (refer to Figure 7.9.6).

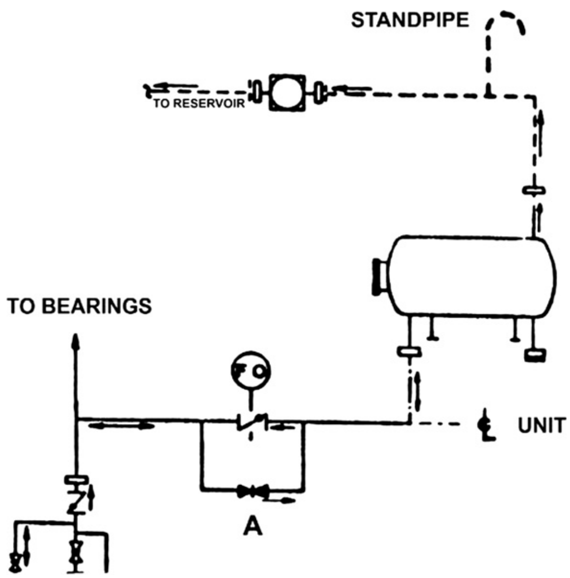


Fig 7.9.6 • Lube oil rundown tank system with continuous overflow
(Courtesy of Elliott Co.)

The function of the tank is to provide oil to all bearings at a lower pressure than normal, but still sufficient to preclude bearing damage during emergency shutdown. In this case, the tank will drain when both main and auxiliary pumps cease to function. An important consideration is the material of the rundown tank, since it represents a large vessel downstream of the filter in the system. Any rust scale or corrosion present in this tank will go directly into the bearings on shutdown and could cause a significant problem. Rundown tanks should also be sized to provide sufficient flow for the entire coastdown period. The calculation of the equipment coastdown time must include process system information. The lower the system pressure (resistance) during the coastdown period, the longer the equipment will continue to turn. The vendor and user must coordinate closely regarding the rundown tank sizing. It is recommended that a small amount of flow be continuously circulated through the rundown tank to prevent sediment accumulation and maintain operating viscosity (in cold regions). This is accomplished by installing a return line from the tank to the oil drain line that incorporates an orifice to regulate flow (approximately 8 LPM [2 GPM]).

It is strongly recommended for lube systems with rundown tanks or emergency pumps that in the case of equipment shutdown, even though equipment is furnished with rundown tanks and emergency pumps, bearings should be checked for wear upon coastdown. Failure to do so could result in catastrophic damage to the equipment if bearings were failed at shutdown. Additionally, in systems employing gears, gear box inspection covers should be removed and the gear mesh checked since spray elements, if clogged, may not distribute sufficient lubricating flow to the gear mesh in emergency conditions.

Centrifugal shaft-driven main pump, motor-driven positive displacement auxiliary system pump

Refer to Figure 7.9.7. This system combines two types of pumps; dynamic and positive displacement. An important consideration is that the maximum pressure of the auxiliary (positive displacement) pump be limited below the maximum pressure of the shaft-driven (dynamic) pump. If this is not done, the shaft-driven pump output pressure, when operating with the stand-by pump, will be less and its output flow will be reduced to zero. Continued operation in this mode will result in overheating of the shaft-driven pump and failure. The stand-by pump discharge pressure setting is regulated by a bypass valve or a relief valve in this case. The setting of this valve must be checked to be sure it is less than the dynamic pumps discharge pressure at the dynamic pump minimum flow point.

Arrangement options

In this section we will briefly discuss a few arrangement options available for lubrication systems. As mentioned, the arrangement of auxiliary systems directly determines system reliability since arrangement determines accessibility to component parts that must be serviced and calibrated while critical equipment is

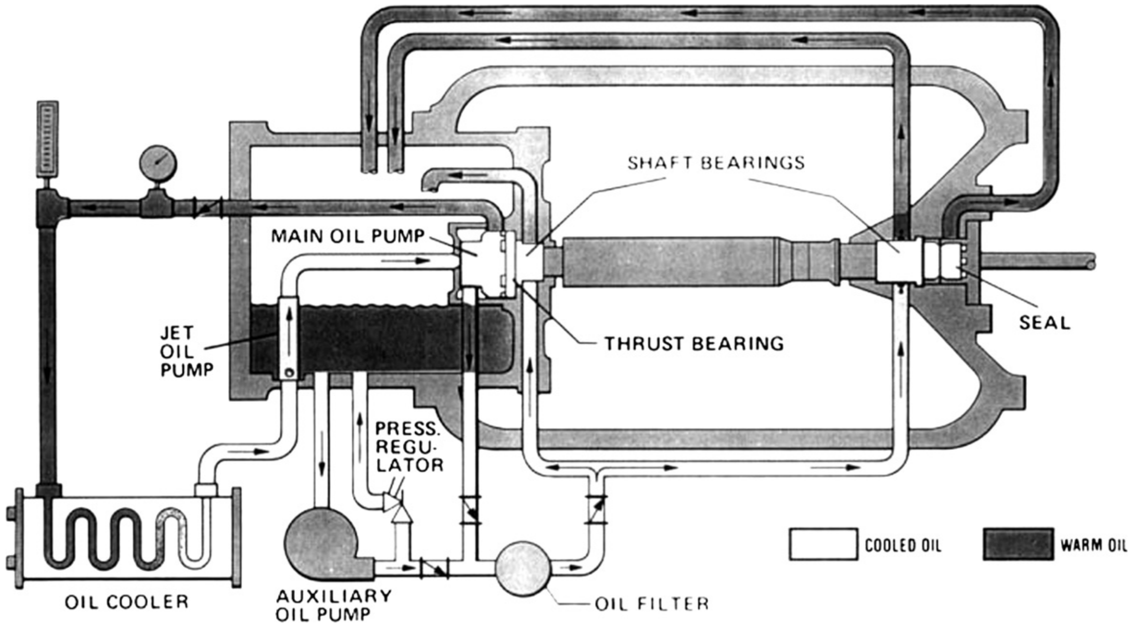


Fig 7.9.7 • Shaft-driven centrifugal pump motor-driven auxiliary pump (Courtesy of York International Corp.)

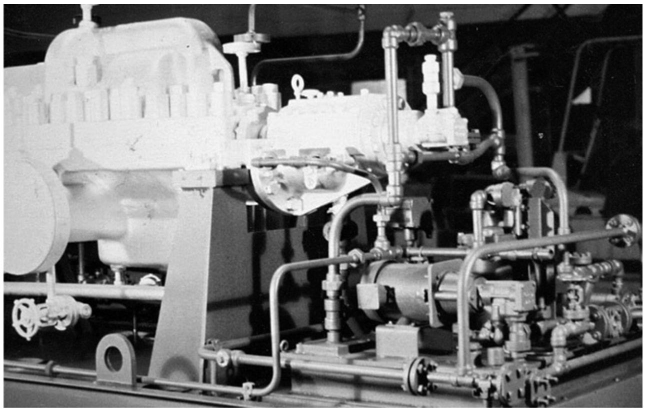


Fig 7.9.8 • Modularized oil console arrangement (Courtesy of Fluid Systems, Inc.)

operating. Attention must be drawn to particular applications and the need to maximize component accessibility. It is recognized that certain applications contain minimal space for auxiliary equipment and that equipment must be arranged for the available spaces.

Integral auxiliary systems

Refer to Figure 7.9.8. Such a system incorporating the lube oil system in the baseplate of the critical equipment is used in remote applications and frequently on platforms since space is at a premium. This system should be reviewed thoroughly in the design phase to optimize accessibility. Note that even though

Fig 7.9.9 • Horizontal oil console arrangement (Courtesy of G.J. Oliver, Inc.)

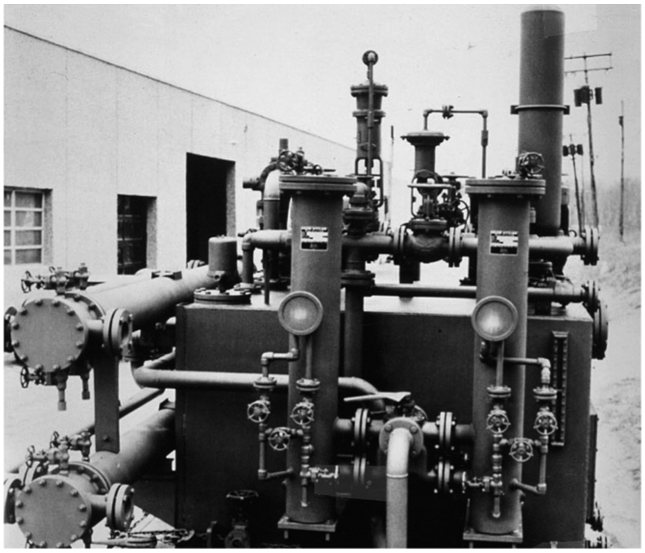


Fig 7.9.10 • Modularized oil console arrangement (Courtesy of Fluid Systems, Inc.)

space is minimal, major components can be arranged such that accessibility is possible.

Horizontal console (no components on the reservoir)

Please refer to Figure 7.9.9. This console arrangement is fairly typical of a critical equipment lubrication system. Note that the positions of components afford ample space for maintenance and equipment calibration. In addition, note the placement of interconnecting piping, thus allowing for maximum mobility on the console.

Reservoir integral with component

Refer to Figure 7.9.10. This arrangement is frequently used in restricted space locations, or could be the result of an attempt by the original equipment vendor to minimize cost. Careful scrutiny of arrangement design early during the project can avoid problems. Note from the figure that even though all components are mounted on the reservoir, accessibility is maximized to components. In this case, the reservoir was mounted approximately three feet below grade, thus allowing all components to be within easy maintenance reach.

Best Practice 7.10

If the main oil pump driver must be single stage steam turbine observe the following best practices for maximum steam turbine reliability:

- Use an eductor for the steam seal system
- Select a control valve actuator with 50% additional force than required
- Do not use hand valves
- Install oil condition monitoring bottles at the bottom of each bearing bracket
- Do not use a sentinel valve

The above best practices will ensure optimum main oil pump driver reliability and result in a serviced unit of 99.7% reliability or greater.

Lessons Learned

Steam turbines used for main oil pump drivers have the lowest reliability of oil system components and are responsible for most oil system trips.

Benchmarks

This best practice has been used since 1990 to produce oil systems of highest reliability, and has resulted in unit reliabilities above 99.7%.

B.P. 7.10. Supporting Material

See Chapter 5 of this book for Single Stage Steam Turbine Best Practices (B.P.s 5.14, 5.15, 5.16, and 5.17).

Best Practice 7.11

Always test oil system relief valves on the oil console and not on the PSV test rig to ensure that the settings are not lower than specified.

All oil system relief valves are the modulating type.

Modulating type relief valves start to open at the specified set point but require additional pressure (‘accumulation’) to open fully.

PSV test rigs are set up for ‘poppet’ type relief valves that open fully at their set point.

Frequently, oil system relief valve set points are set erroneously for their full open set point on the PSV test rig which results in their opening prematurely when re-installed on the oil console.

Using a calibrated pressure gauge, and testing the relief valves on the oil console saves time and ensures the proper setting.

Lessons Learned

Many unit trips have been traced back to improper setting of relief valves that caused them to open at lower than set pressures which required the auxiliary pump to start. Starting of the auxiliary pump was either too late or caused control valve instability resulting in a low oil pressure trip and a unit trip.

Benchmarks

This best practice has been used since the early 1980s when commissioning a large petrochemical plant. Since that time, implementing this best practice has resulted in plant oil system and unit reliabilities above 99.7%.

B.P. 7.11. Supporting Material

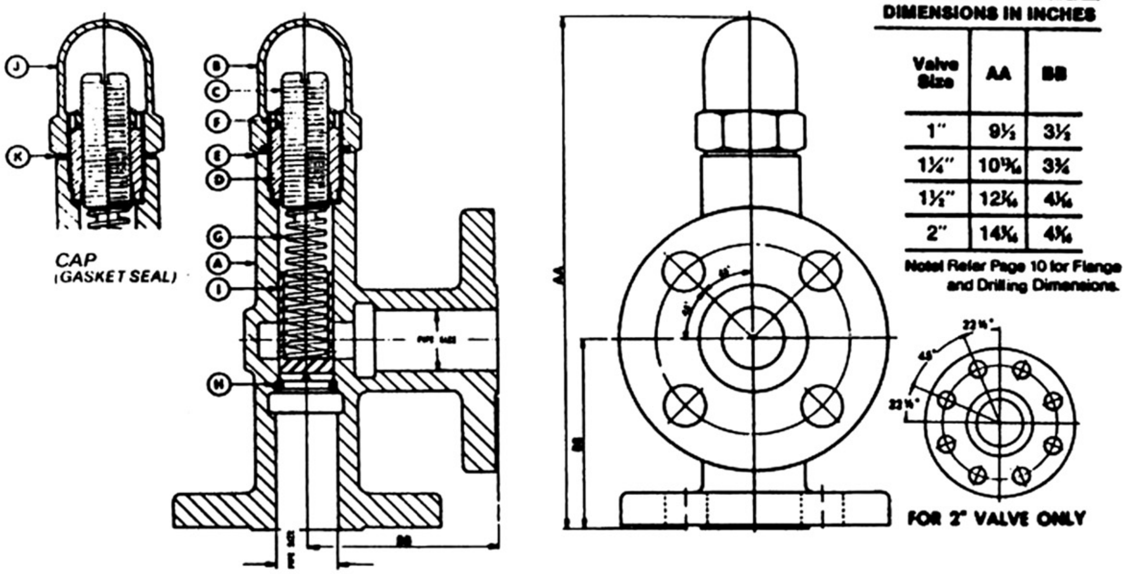
Relief valves for positive displacement pumps

Since positive displacement pumps are not self-limiting, i.e., they can produce increasing pressure if sufficient driver power is available, hence a device to limit pump pressure and horsepower is required.

The function of a relief valve as a protection device is to limit pump discharge pressure and horsepower to a specified value without generating any valve instabilities. While the function of a relief valve is simple enough, valve chatter (instability) and failure to positively reseat can cause the shutdown of the critical equipment. Relief valve chatter can cause high pressure pulses that will activate shutdown pressure switches and damage valve seats and plugs.

The inability to reseat properly will introduce an 'equivalent orifice' into the system that will reduce or totally eliminate the system flow to the critical system components.

DIMENSIONS



DIMENSIONS IN INCHES

Valve Size	AA	BB
1"	9 1/2	3 1/2
1 1/4"	10 13/16	3 3/4
1 1/2"	12 3/16	4 1/16
2"	14 3/16	4 5/16

Note! Refer Page 10 for Flange and Drilling Dimensions.

PARTS LIST

SYM.	NAME	MODEL	VALVE SIZE 1"	1 1/4"	1 1/2"	2"
A	BODY	VJF, VJF-SP VBF VSF VSSF	500-F 500-BF 500-SF 500-SSF	600-F 600-BF 600-SF 600-SSF	700-F 700-BF 700-SF 700-SSF	800-F 800-BF 800-SF 800-SSF
B	CAP (O-RING SEAL)	VJF, VJF-SP VBF VSF VSSF	501-R 501-BR 501-SR 501-SS	601-R 601-BR 601-SR 601-SS	701-R 701-BR 701-SR 701-SS	801-R 801-BR 801-SR 801-SS
C	ADJUSTING SCREW	VJF, VBF VSF, VJF-SP VSSF	502-B 502-S 502-SS	602-B 602-S 602-SS	702-B 702-S 702-SS	802-B 802-S 802-SS
D	RETAINER	VJF, VBF VSF, VJF-SP VSSF	503-B 503-S 503-SS	603-B 603-S 603-SS	703-B 703-S 703-SS	803-B 803-S 803-SS
E	O-RING †	VJF, VJF-SP, VBF, VSF VSSF	504-* 504-RT	604-* 604-RT	704-* 704-RT	804-* 804-RT
F	LOCKNUT	VJF, VJF-SP, VBF, VSF VSSF	505-S 505-SS	605-S 606-SS	705-S 705-SS	805-S 805-SS
G	SPRING †	ALL MODELS	507-**	607-**	707-**	807-**
H	STOP RING	VJF, VBF VJF-SP, VSF VSSF	508-B 508-S 508-SS	608-B 608-S 608-SS	708-B 708-S 708-SS	808-B 808-S 808-SS
I	PISTON †	416 STAINLESS STEEL HARDENED STEEL 316 STAINLESS STEEL	506-A 506-S 506-SS	608-A 608-S 606-SS	706-A 706-S 706-SS	808-A 808-S 808-SS
J	CAP (GASKET SEAL)	VJF, VJF-SP VBF VSF	501 501-B 501-S	601 601-B 601-S	701 701-B 701-S	801 801-B 801-S
K	GASKET †	VJF, VBF VJF-SP VSF	504 504-S	604 604-S	704 704-S	804 804-S

*See o-ring selection chart
**See spring pressure chart
† Recommended spare parts

Fig 7.11.1 • Modulating relief valve (Courtesy of Fulflow Specialties Co. Inc.)

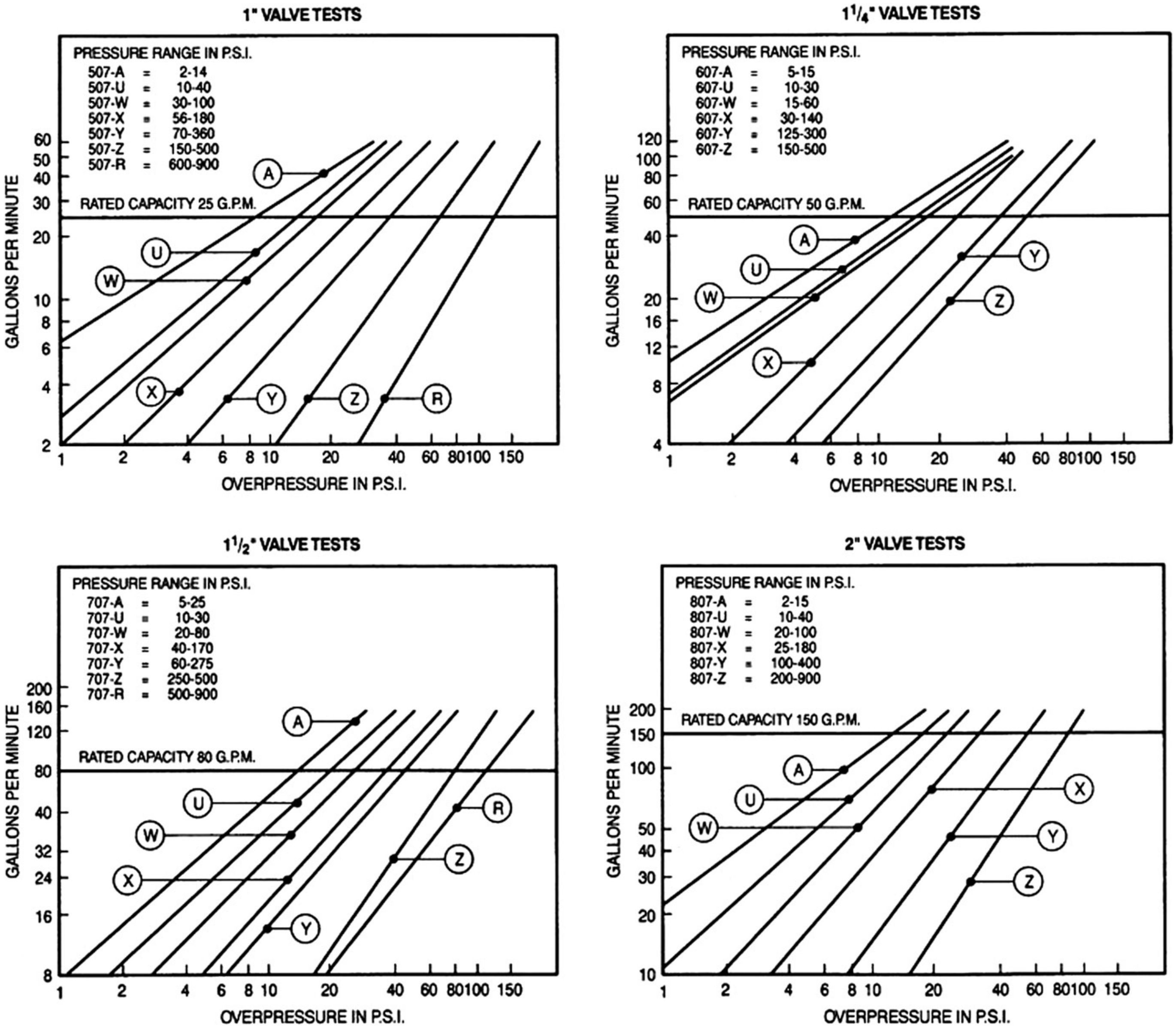


Fig 7.11.2 • Relief valve sizing chart (Courtesy of Fulflow Specialties Co. Inc.)

Experience has shown that a sliding, piston-type, relief valve, which is a modulating device, as opposed to a spring-loaded poppet valve, which is an on-off device, meets the requirements of stability and positive shutoff for liquid auxiliary system service. A typical relief valve is shown in Figure 7.11.1. A sizing chart for this type of relief valve is shown in Figure 7.11.2. Relief valve set pressure is usually set 10% above the pump maximum discharge pressure. However, the maximum pressure ratings of all system components must also be considered. Given the maximum pump flow and the relief valve set pressure, the maximum system pressure can be determined as follows:

$$\text{Maximum system pressure} = \text{relief valve set pressure} + \text{relief valve overpressure.}$$

Relief valve overpressure is the valve pressure drop necessary to pass full pump flow.

For the present example using a 2" valve:

1. Maximum pump discharge pressure = 200 PSIG
2. Relief valve set pressure (cracking pressure) = 1.1 × 200 = 220 PSIG
3. Maximum system overpressure = 220 PSIG + 25 PSIG = 245 PSIG (from Figure 7.11.2 for Y spring and 86 GPM flow)

Note that the overpressure values are sensitive to viscosity, and can only be used up to a viscosity of 500 SSU. Above this value, the overpressure can be estimated to vary by the relationship:

$$\begin{aligned} \text{overpressure}_{@\text{viscosity}} &= \text{overpressure}_{@500\ \text{SSU}} \times \sqrt[4]{\frac{\text{viscosity}}{500\ \text{SSU}}} \\ &= 35\ \text{psi}\sqrt[4]{\frac{1000}{500}} \\ &= 35\ \text{psi} \times 1.19 \\ &= 42\ \text{psi} \end{aligned}$$

Therefore, the maximum pressure at 1000 SSU will be 262 psi, or 19%.

Relief valve overpressure expressed as a percentage of relief-valve set point is defined as *accumulation*. Typical values of accumulation vary between 10 and 20%.

Best Practice 7.12

Install sight glasses in the drain lines of positive displacement pump relief valves to confirm that the relief valve is not passing.

It is very difficult to confirm that a positive displacement pump relief valve is not passing.

A friction-bound relief valve can cause an unexpected shutdown of an oil system by passing an amount of additional oil flow that can force the start-up of an auxiliary pump, thus exposing the unit to a shutdown if the auxiliary pump does not start in time.

Requiring all pump relief valve discharge lines to be supplied with a sight glass of the proper pressure rating will enable personnel to monitor the condition of the relief valve, and so be able to take appropriate action when necessary.

Lessons Learned

Critical (un-spared) unit trips have been caused by friction-bound relief valves being stuck open, resulting in a required start of the auxiliary pump which did not occur in time to prevent a low oil pressure trip.

Feeling the discharge line of the relief valve gives a false impression of valve condition, since most oil system relief valves are the modulating type with a small bypass hole to prevent sticking of the valve that allows a small continuous of flow to pass through the valve.

Therefore, the use of sight glasses in relief valve discharge lines of the proper pressure is the most effective way to confirm the proper operation of relief valves.

Benchmarks

This best practice has been used since 1990, when a project with a Europe-based compressor vendor used relief valve sight glasses as a standard scope of supply. This practice has saved many unnecessary unit shutdowns for critical equipment trains and has optimized train reliability (99.7% or greater).

B.P. 7.12. Supporting Material

Please refer to material in B.P: 7.11.

Best Practice 7.13

Use oil-actuated control valves with a sensing line anti-pulsation device whenever possible in oil systems, for rapid transient response and stability.

Oil systems can use oil or pneumatically actuated control valves.

It has been my experience that pneumatically actuated control valves require more corrective adjustment than oil-actuated valves, and they do not respond at the required speed during most oil system transient events (main oil pump trip, auxiliary pump start, etc.).

Oil-actuated valves can always be used for the backpressure (bypass) valve, lube oil supply valve, control oil supply valve and seal oil differential pressure control valve.

Oil-actuated valves should be supplied with an oil sensing line anti-pulsation device to ensure stability during transient events.

Lessons Learned

End users have found, after many oil system trip events, that the use of oil-actuated control valves to replace pneumatically actuated valves ensures optimum oil system and serviced unit reliability.

Benchmarks

This best practice has been used since the early 1970s, when I was an oil system designer for a compressor/steam turbine vendor. Many field installed systems with pneumatic actuated control valves have been replaced with oil actuated valves, resulting in trouble-free system operation and optimum unit reliabilities (99.7% or greater).

B.P. 7.13. Supporting Material

Referring to the general definition of an auxiliary system which is '*to continuously supply cool, clean fluid to each specified point at the required pressure, temperature and flow rate*', we can see that the controls and instruments play a major role in the reliability of auxiliary systems. The function of the controls and instrumentation is to continuously supply fluid to each specified point at the required pressure, temperature and flow rate. While it is true that pumps and coolers must be present, system controls modify the operational characteristics of these components to achieve the desired results. In addition, system instrumentation initiates transient system response, continuously monitors operation and shuts down critical equipment in the event of an auxiliary system malfunction. In this section, we will examine important concepts that are at the heart of auxiliary system reliability, define the function of major control and instrumentation components and discuss items that can significantly reduce auxiliary system reliability.

Types

Types of major auxiliary system controls and instrumentation are outlined in Figure 7.13.1. Note that types are defined by function. As an example, a positive displacement pump system flow control consists of a pressure control valve that bypasses excess flow from the pump back to the system reservoir to maintain a set system pressure. The function of this component, however, is to continuously supply the required flow of fluid to the system under varying system pressure drops and critical equipment component conditions (worn bearing, seal, etc.).

Controls	Instrumentation monitor and alarm
■ Positive displacement pump system flow control ■ Dynamic pump system flow control ■ Stand-by pump automatic start ■ Cooler temperature control ■ System differential supply pressure control (constant reference pressure) ■ System differential supply pressure control (variable reference pressure)	■ System reservoir level ■ Pump operation ■ System pressure ■ System temperature ■ Filter differential pressure ■ System differential pressure (variable reference pressure) ■ Variable speed pump driver speed indicator

Fig 7.13.1 • Major auxiliary system controls and instrumentation (by function)

All system controls and instrumentation must function perfectly under both steady state and transient conditions. Under normal operation, a steady state control mode is approached since flows, pressures and temperatures change very slowly if at all. While this mode of operation may appear to be ideal, it can be dangerous since control valves and instrumentation can bind up due to debris and lack of movement. In transient mode, components must have response times of the order of milliseconds. When one considers the function of an auxiliary system and the fact that the slowest of critical equipment units operate at approximately 60 revolutions per second (3,600 RPM), the necessity of rapid system response time can be appreciated. If the controls cannot respond to a transient response, the instrumentation and the critical equipment shutdown system (circuit breaker, steam turbine trip valve system, etc.) must operate on demand to stop equipment operation. If the system controls and instrumentation do not have sufficient response times, a system liquid supply source (accumulator) is required to provide flow during transient conditions. Using our system as an example, 60 GPM – or one gallon per second – are supplied to the unit. Suppose the main pump trips, and the normal flow to the equipment is not reached for three seconds (until the stand-by pump is at full speed and flow rate). An accumulator with a liquid capacity of three gallons would enable the system to function normally during the upset since it would supply the required flow of one gallon per second. Note an accumulator size greater than three gallons would be required. This will be covered separately.

Concepts

The use of concepts can be helpful in understanding the function of auxiliary system components and systems. In this section we will discuss:

- An equivalent orifice
- Sub-systems
- An equivalent vessel
- Control valve liquid coefficient – C_v
- A flow meter in every system

The concept of an equivalent orifice

Bearings seals, etc. can be reduced to the concept of an equivalent orifice (see Figure 7.13.2). The equation for orifice flow is:

$$Q = C \times C_f \times D^2 \sqrt{\frac{\Delta p}{S.G.}}$$

From the above equation it can be seen that flow to any component is the function of the dimension 'D^2' and ΔP across that component. System components essentially experience two types of flow changes; the gradual flow change due to component wear (i.e., D^2 change as in the case of bearing wear) or the sudden flow change due to a pressure change in the system. As can be seen from the above equation, a sudden change of pressure as in the case of a hunting control valve, or a sudden pressure spike due to component starting or stopping, will cause a corresponding sudden change in flow rate to the component. Considering the speeds involved in critical equipment, one can appreciate that a short term, transient flow change can lead to significant component damage of the critical equipment (bearing, seals, etc.). The concept of reducing each individual critical equipment component (bearing, seals, orifices, etc.) to an equivalent orifice helps enormously in conceptualizing transient system reactions.

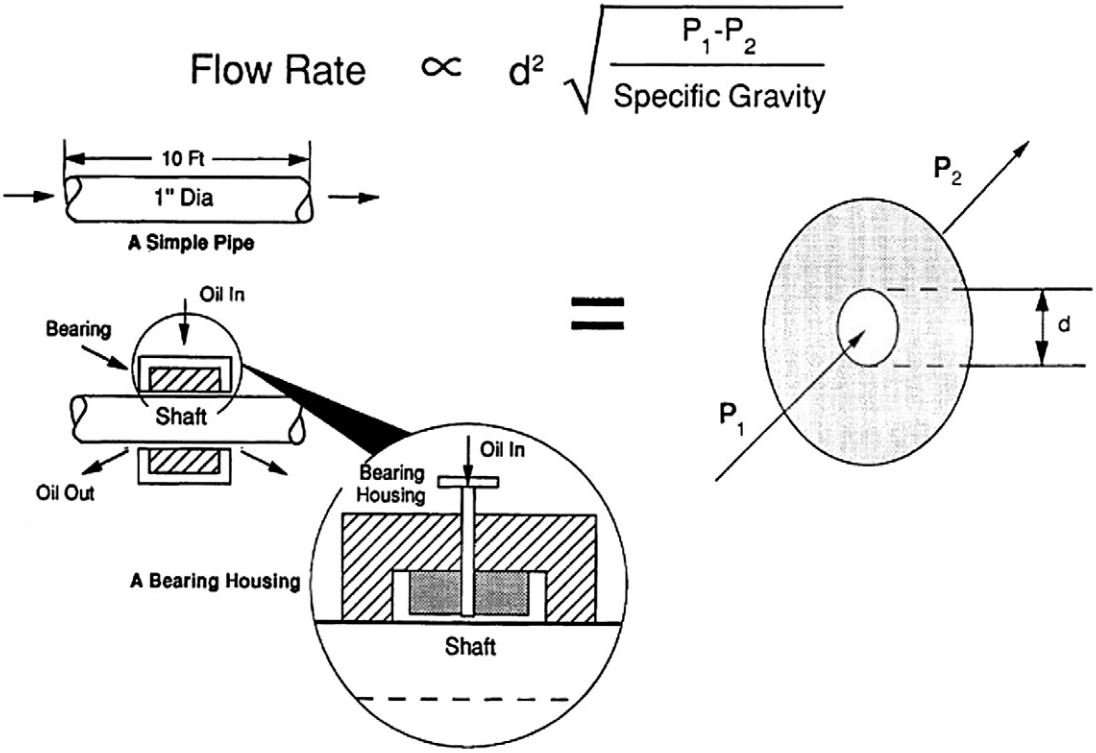


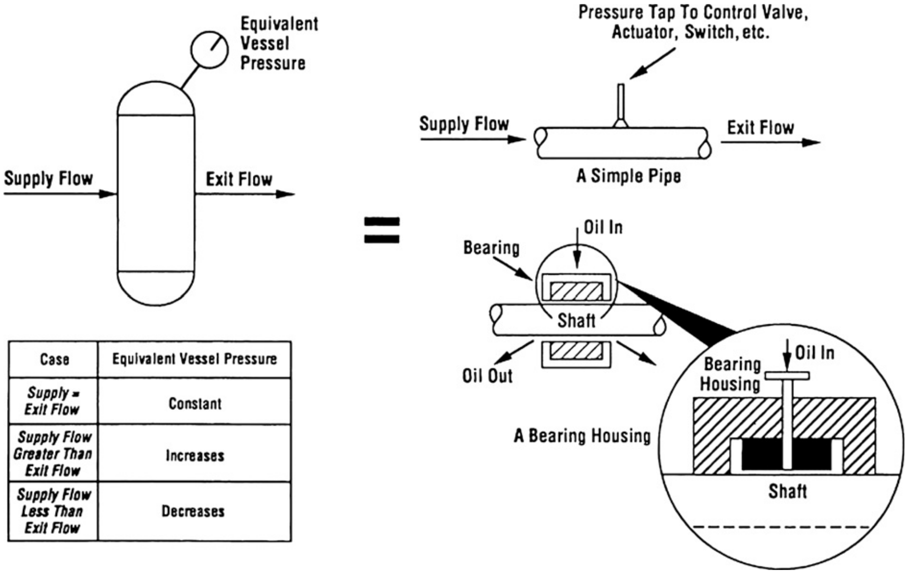
Fig 7.13.2 • Reduce it to an equivalent orifice

Sub-systems

Both positive displacement and dynamic pumps alone do not contain the desired characteristics for operation within an auxiliary system. These components must be combined in a controlled sub-system to achieve the desired results. The sub-system is the combination of the pump and its control valve, which together produce the required flow characteristic. Viewing components in control and instrumentation as being part of various sub-systems also helps in understanding the total function of auxiliary systems.

Equivalent vessel

Refer to Figure 7.13.3. Systems and sub-systems can be reduced to equivalent vessels. As an example, the supply pipe from a lube oil console can be reduced to an equivalent vessel as shown in Figure 7.13.3.



Case	Equivalent Vessel Pressure
Supply = Exit Flow	Constant
Supply Flow Greater Than Exit Flow	Increases
Supply Flow Less Than Exit Flow	Decreases

Fig 7.13.3 • Reduce it to an equivalent vessel

When supply flow equals exit flow, the pressure in any equivalent vessel remains constant. If supply flow is less than exit flow, the pressure reduces rapidly. The function of an accumulator can be easily understood by using this equivalent vessel concept. If a vessel is installed downstream of the equivalent vessel in Figure 7.13.3, during the period of reduced inlet flow the vessel would supply flow to the system. This is exactly the function of an accumulator.

Another example of using the equivalent vessel concept is as follows: Imagine again the equivalent vessel is a supply pipe from a lube oil console. Suppose the main pump trips on overload, and the auxiliary pump does not start immediately. Since the auxiliary pump did not immediately start, the supply flow to the equivalent vessel is less than the exit flow. As a result, the pressure in the equivalent vessel will drop. This is why pressure switches in auxiliary systems are used as alarm, auxiliary pump start or trip devices. Using our concept of an equivalent vessel, it can be seen that the pressure switch actually acts as a flow indicator and will activate on low flow even though it is measuring pressure.

Control valve liquid sizing coefficient – C_v

'C_v' is an important concept that must be understood when dealing with any type of control valve in liquid service. C_v, the 'valve sizing coefficient', is defined by the following equation:

$$C_v = Q\ (\text{GPM})\sqrt{\frac{\text{S.G.}}{\Delta P}}$$

Where: S.G. (specific gravity) = 0.85 (for oil)

ΔP = value pressure drop (psi)

Solving this equation for GPM we see that:

$$Q(\text{GPM}) = \frac{C_v}{\sqrt{\frac{\text{S.G.}}{\Delta P}}}$$

We can see by referring back to 'the concept of an equivalent orifice' that this equation is similar to that of an orifice. Naturally, the only difference is that a valve is a variable orifice. Valves are sized using this concept of C_v (valve coefficient). Each valve has a maximum C_v. Depending on the type of internal valve design, seats, plugs, and body, a valve will exhibit a certain characteristic. Refer to Figure 7.13.4 which is a graph of valve characteristics.

Plotted on the ordinate (y axis) is valve flow in percent of maximum flow and plotted on the abscissa (x axis) is travel of the valve plug in percent of rated travel. The characteristics of particular valves will be discussed when we cover specific valve application later in this section. Referring back to the relationship for a valve coefficient shows it to be dependent on flow rate, differential pressure across the valve, and fluid characteristics.

As an example, suppose that a valve is sized to pass 20 GPM under normal conditions of 150 psi pressure drop. The fluid in this case is light turbine oil at 150°F (60 SSU). Solving for the valve C_v with the above equation, we arrive at a figure of 1.51. If the valve pressure drop were to decrease to 100 lbs, and we still required 20 gallons per minute to pass, the valve coefficient would be 1.84. This change represents approximately a 22% change in the valve coefficient. Depending on the characteristic curve of the valve in question, it would represent a given amount of valve plug opening (increase of travel). In the same example, now let us assume that the flow changes to 40 GPM with 100 lbs pressure drop across the valve. The C_v now would be 3.69 or approximately 200% of the previous value. Depending on the valve size, this coefficient may or may not be obtainable. Refer to Figure 7.13.5 which is a typical valve coefficient table showing valve coefficients for % travel of a particular valve. When sizing all control valves, C_v maximum, C_v normal and C_v minimum must be calculated. A general rule is that all of the above values should fall between 10% and 90% of the maximum C_v for a particular valve selected.

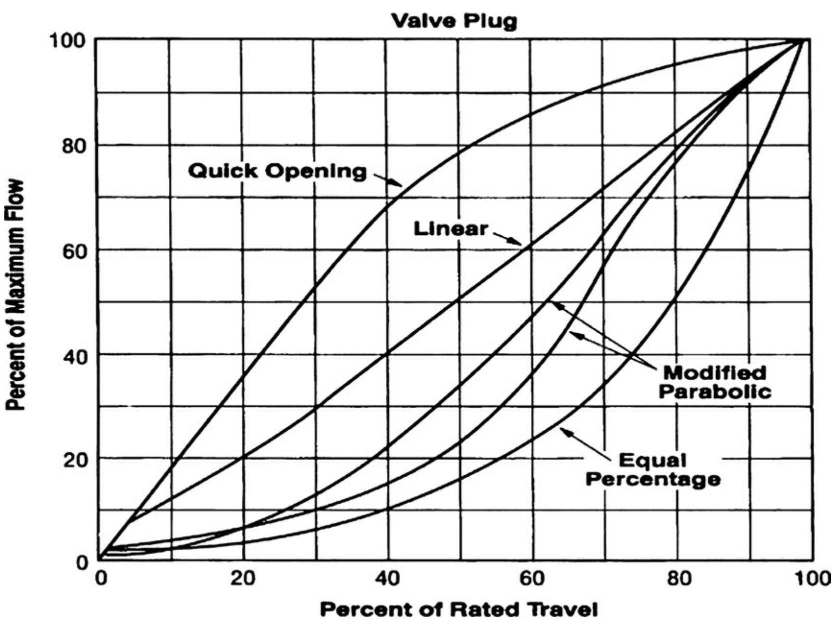


Fig 7.13.4 • Control valve flow characteristics (Courtesy of Fisher Controls Inc.)

Body size	Port size	% Travel (12.5%)	(25.0%)	Valve Travel (50%)	(75%)	(100%)
*	*	1/32"	1/16"	1/8"	3/16"	¼"
1"	3/4	1.4	3.1	4.2	5.3	7.0
1"	1"	2.4	4.2	7.0	10.0	12

Fig 7.13.5 • Typical liquid valve sizing coefficient table

When dealing with viscous liquids, as in the case of oil, valve coefficient viscosity corrections must be made. For the example case mentioned above, the correction factor for 220 centistokes (1,000 Sabolt Universal Seconds [SSU]) would be approximately 1.5 to 2. Therefore the valve coefficient required would be 1.5 to 2 times that required at normal viscosities (60 SSU for light turbine oil at normal operating temperatures). Viscosity correction nomographs are available from control valve manufacturers for determining valve sizes required under high viscosity conditions.

A flow meter in every system

Considering the relationship discussed above, it can be seen that every control valve can be considered as a flow meter if the fluid differential pressure across the valve, valve travel and a valve characteristic chart is known. While not a completely accurate flow measuring device, this concept can be extremely valuable when troubleshooting auxiliary systems. Obtaining the valve travel and using the valve coefficient chart, the C_v can be obtained. Calculating for GPM knowing the C_v, the pressure drop across the valve and the specific gravity of the liquid can then yield the flow rate. It is important to note that with small valve travels, of the order of 1/4 inch maximum, an accurate means of measuring valve travel must be obtained. It is my experience that travel indicators are not often supplied with the valve. It is strongly recommended that valve travel indicators be supplied or retrofitted in the field.

Bypass control

The first application to be discussed in this section will be that of a bypass control valve. A bypass control valve and actuator pictured in Figure 7.13.6 is used with a positive displacement pump to alter the pump's flow characteristic to that of variable flow.

Refer to the schematic of a lube oil system typical of the example in Figure 7.13.7. This system incorporates positive displacement pumps. The control valve's function is to control flow to the critical equipment continuously, such that the required flow is supplied under normal and under transient conditions. Since a positive displacement pump is essentially a constant flow device, in bypass mode the control valve must allow for excess pump flow to be recirculated back to the reservoir. Utilizing the concept of an equivalent orifice, as the bearings in the system wear, the orifice diameter becomes larger, therefore the flow required to the critical equipment will be greater. Since the downstream pressure across the bearings is atmospheric pressure, the upstream pressure will initially decrease when the bearing area becomes larger for the same flow. The bypass valve will sense the upstream pressure reducing, and will close to force the additional required flow to the critical equipment. Even though the bypass valve is a pressure device, it is acting as a flow control device to divert bypass flow to required system components. Therefore using the concept of a sub-system, the bypass valve and the positive displacement pump form a variable flow sub-system that will supply variable flow to the critical equipment on demand.

In addition to accounting for small changes in system flow requirements, the bypass control valve must also act under

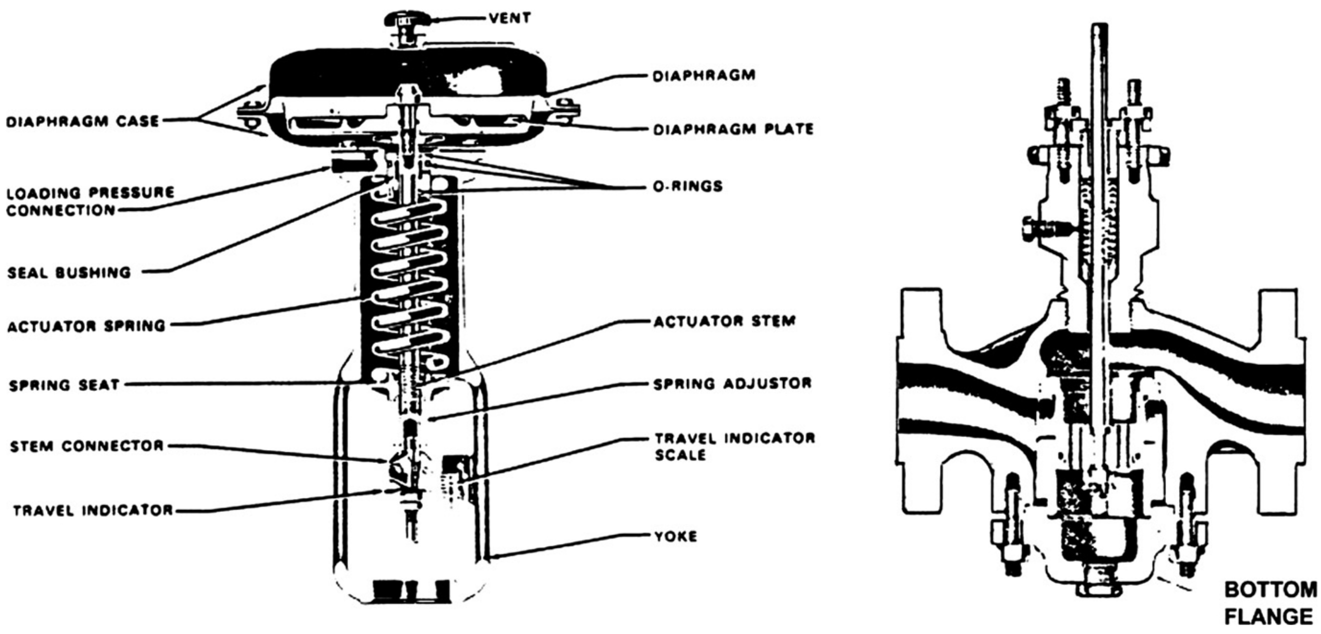
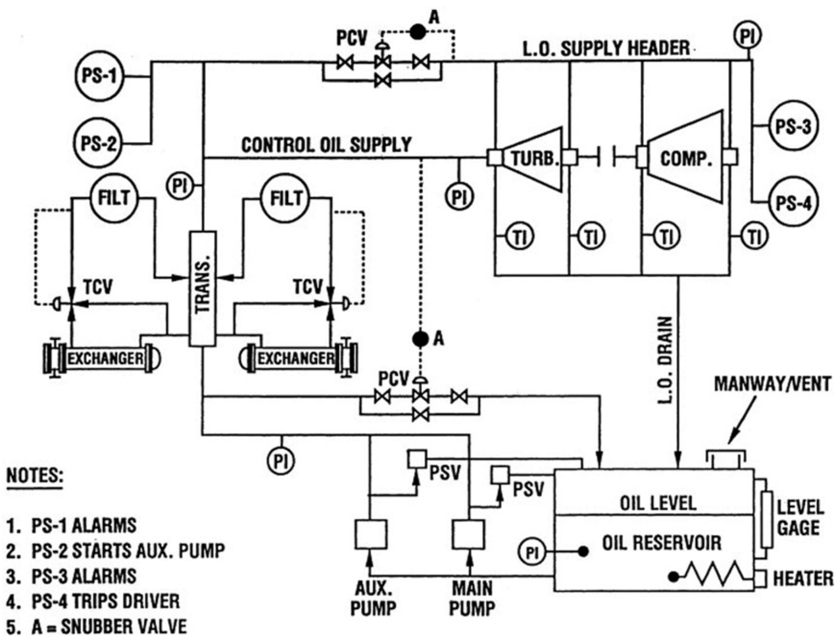


Fig 7.13.6 • Reverse acting actuator and valve body typically used as a back pressure regulator (bypass control) (Courtesy of Fisher Controls Inc.)

Fig 7.13.7 • Typical lube oil supply system



ITEM	HORSEPOWER	HEAT LOAD BTU/HR	GPM
1	30	76,350	15
2	10	25,450	5
3	10	25,450	5
4	10	25,450	5
5	10	25,450	5
6	30	76,350	15
7	8	20,500	10
TOTAL	108	275,000	60

transient conditions. If the main pump were to suddenly shut off, the system would immediately sense a pressure decrease. Referring to the equivalent orifice concept of bearings, the flow to these components would drop proportionately to the square root of the pressure drop across the component. At hundreds of revolutions per second, the bearing shaft interface would not last long in the absence of system flow. In this transient mode, the control valve must close quickly to divert all bypass flow to the system to account for the absence of flow from the pump. The control valve characteristic, its actuator and supply to its actuator whether direct (hydraulic) or indirect (pneumatic) must function instantaneously. If the valve system experiences instabilities or excessive friction, as in the case of valve stem binding, the system will experience an instantaneous loss of flow and will (hopefully) be shut down on this signal. Again referring to the concepts discussed above, the concept of an equivalent vessel is useful in ascertaining how pressure and flow are related and why pressure switches are used to determine loss of flow under transient conditions. This concept also shows why time delays in auxiliary systems are not desired to be used with any trip devices. It is true that a time delay would preclude a trip of the unit under transient conditions, but it could also cause severe and perhaps catastrophic damage to the critical equipment.

The bypass control valve also must exhibit rapid transient response in the open direction. In the case of dual pump simultaneous operation, the amount of flow to be recirculated to the reservoir will be equal to the normal bypass flow of one pump plus the full flow of the stand-by pump. If the bypass valve does not act as a variable orifice, and opens at a slower rate than the flow rate increase, referring to the orifice equation, the pressure drop across the valve will simultaneously rise. This increase may exceed the setting of the relief valve in the system. If this is the case, the system is exposed to the potential of the relief valve not re-seating. If this were to occur, a new 'orifice' would be introduced into the system and the flow to the critical equipment would be reduced to the point of requiring the stand-by pump to start and possibly causing critical equipment shutdown. In order to meet the above control and transient requirements, the bypass control valve must be sized properly. An example of valve sizing using the system shown in Figure 7.13.7 is shown in Figure 7.13.8. We wish to re-emphasize that once the valve is sized properly, the actuator and the sensing lines in the system that supply the force to operate the valve must be designed for rapid response. In many systems, sensing line snubbers are used to dampen impulse signals that can lead to valve instability. It must be noted that snubbers are designed to provide quick response in

Given:

1. Normal valve flow	=	pump flow – normal system requirement
	=	73 GPM – 60 GPM
	=	13 GPM
2. Maximum valve flow	=	main and auxiliary pump flow – minimum system requirement
	=	146 GPM – 60 GPM
	=	86 GPM
3. Maximum valve P	=	pump discharge pressure @ maximum supply flow and component P
	=	250 PSIG
4. Minimum valve P	=	pump discharge pressure @ minimum supply flow and component P (clean system)
	=	160 PSIG
5. Oil specific gravity	=	0.85
DETERMINE:		C_v Minimum
		C_v Maximum
1. C_v Min.	=	$\text{Q NORMAL}\sqrt{\dfrac{\text{S.G.}}{\Delta\text{P Max.}}}$
	=	13×0.0583
	=	.758
2. C_v Max.	=	$\text{Q MAX.}\sqrt{\dfrac{\text{S.G.}}{\Delta\text{P Min.}}}$
	=	86×0.0729
	=	6.268

Refer to Figure 6.5 for 1" valve with 3/4" port and obtain:

Valve maximum	C_v = 7.0
Valve operating maximum	C_v = 6.268
Valve operating minimum	C_v = 0.758
Valve maximum travel (opening)	= 90%
Valve minimum travel (opening)	= 9%

Note: Valve minimum and maximum openings are at the limit for satisfactory operation.

Fig 7.13.8 • Valve sizing example – back pressure (bypass) control

one direction and retarded or slower response in another. It is of extreme importance that these devices be installed properly. Understanding the function of the particular valve in question and examining the direction of the snubber device in a sensing line is essential to correct system operation. Many times these snubber devices are installed improperly in the wrong direction.

Pressure reducing control

Pressure reducing control has two primary applications in auxiliary systems:

- To control the flow from a dynamic pump.
- To reduce the pressure in the system.

A typical pressure reducing control valve and actuator are shown in Figure 7.13.9.

For the first case, the flow characteristic is variable. The flow is therefore determined by the pressure at the discharge flange of a dynamic pump. A pressure reducing valve set to sense the pressure downstream of the valve will automatically regulate the discharge, or the back pressure on the dynamic pump for the desired flow of the system. Referring back to the equivalent orifice concept, if a bearing were to wear, the equivalent diameter of the orifice would increase. Therefore, initially for the same flow rate, the pressure in the system would decrease since the flow is the same. If the bearing clearance increases (equivalent D), the ΔP must decrease. Therefore the pressure control valve sensing decreasing system pressure will open to increase the system pressure. This action will result in a decrease of resistance on the dynamic pump discharge flange and allow the centrifugal pump to operate at a greater capacity to provide the desired flow to the critical equipment. It can be seen that in this case, the pressure reducing valve and the dynamic pump combine to produce a sub-system that meets the objective of providing continuous flow to the critical equipment. The control valve essentially renders the variable flow, constant head device a variable head device by compensating for changes in system pressure. The above case represents the normal control case. Let's now examine the transient case.

If the main dynamic pump were suddenly to trip, the system pressure will suddenly fall since greater flow is exiting the system than is entering it ('equivalent vessel concept'). In this case, the pressure reducing valve sensing downstream pressure would instantaneously open allowing the dynamic pump to move to a higher flow point on its curve, while the auxiliary or stand-by pump starts. As soon as the auxiliary pump starts, the pressure reducing valve, sensing additional flow into a fixed system resistance, would then close, so meeting the flow requirements. Dynamic systems in general tend to be somewhat softer than positive displacement systems. That is, they are more tolerable to transient system changes.

In the case of the auxiliary stand-by pump and the main pump operating simultaneously, the pressure reducing valve would automatically compensate for the increased flow by reducing its travel or increasing the system resistance at the discharge flange of both pumps. That is, increasing the discharge pressure to the level where the combined flow of both pumps would exactly equal the critical equipment system required flow. Again referring to the concept of an equivalent orifice, if excessive flow were forced through the orifice (the bearings) the pressure drop would increase. The pressure reducing valve, sensing the increased system pressure drop, would tend to close to reduce the pressure at its sense point. In doing so it will increase the discharge pressure on both dynamic pumps, and since their characteristic is reduced flow on increased pressure, the desired flow will be obtained. Therefore it can be seen again that the dynamic pump pressure reducing valve sub-system has the function of flow control to the critical equipment even though it is sensing pressure.

The other primary application for pressure reducing valves in auxiliary systems is to reduce system pressure to other desired pressure levels. Refer to Figure 7.13.7, and observe the pressure reducing valve at the discharge of the lube oil system. Its function is to reduce pressure from control oil pressure to lube oil pressure. Control oil pressure is controlled by the equivalent orifice in the control system and the set point of the bypass control valve, as shown in Figure 7.13.7. The bypass control

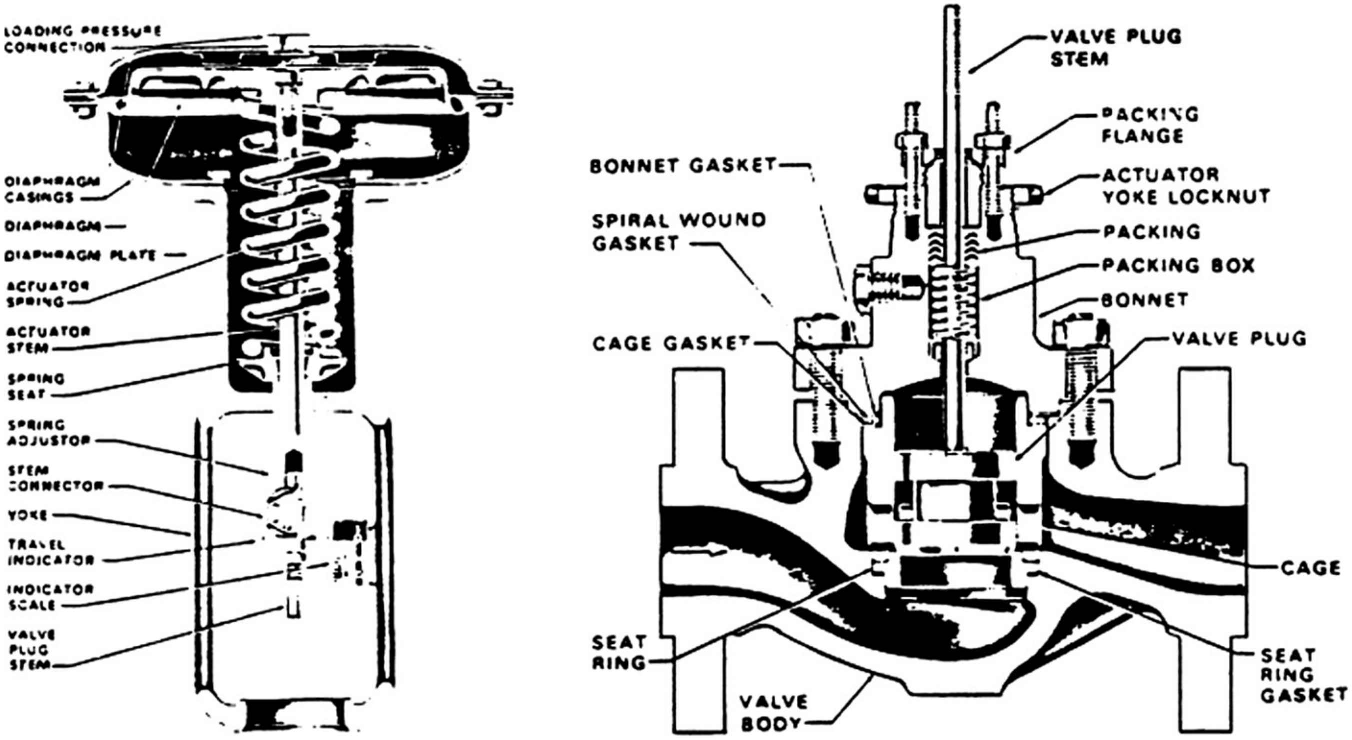


Fig 7.13.9 • Direct acting actuator and valve body used for PRV (pressure reducing) control (Courtesy of Fisher Controls Inc.)

valve senses pressure, and controls flow to satisfy the requirement of the equivalent orifice in the control system and the equivalent orifices in the lube system. The pressure reducing valve simply senses pressure downstream of the valve and controls it at the preset value. It should be noted that in most auxiliary systems, the console (reservoir, pumps, etc.) is usually below the level of the critical equipment; therefore the set point of any pressure reducing valves on the console should compensate for the height or head difference between the console and the critical equipment. Control valves used in pressure reducing service are not usually exposed to system transient changes as in the case of bypass valves. Therefore their sizing is relatively easy and their valve C_vs do not significantly change. A sizing example for a direct acting pressure reducing valve is shown in Figure 7.13.10.

Temperature control valves

Temperature control valves are usually required in auxiliary systems to regulate the supply temperature to the critical equipment components. Especially in systems where liquids have viscosity characteristics (oil systems), temperature control is important to ensure correct oil viscosity to components. Referring to concepts previously discussed in this section, the temperature control valve plus the system coolers make up a cooling sub-system whose function is to continuously supply the required fluid to critical equipment at a specified temperature. Two types of control valves are used; direct acting, three way valves and air operated, two way valves. Both valves sense the mixed temperature downstream of the cooler.

Given:

1. Minimum and normal lube oil flow to unit = 60 GPM
2. Maximum lube oil flow to unit (Bypass valve failed closed) = 73 GPM
3. Valve ΔP = 120 PSIG − 25 PSIG
   = 95 PSI (15 PSIG supply + 20 PSIG) pressure drop for elevation

Note: This example is for a PRV located on the lube oil console at grade.

4. Oil specific gravity = 0.85

Determine: C_v Normal, C_v Maximum

1. C_v Normal $= Q \times \sqrt{\frac{S.G.}{\Delta P}}$ Normal
   $= 60 \times 0.0946$
   $= 5.675$
2. C_v Maximum $= Q \times \sqrt{\frac{S.G.}{\Delta P}}$ Maximum
   $= 73 \times 0.0946$
   $= 6.906$

Refer to Figure 7.13.5 for a 1' valve with a 1' port and obtain valve maximum

C_v = 12.0

Valve operating normal C_v = 5.675
Valve operating maximum C_v = 6.906
Valve normal travel (opening) = 40%
Valve maximum travel (opening) = 50%

Fig 7.13.10 • Valve sizing example – pressure reducing control

A two way valve is a simple bypass around the cooler, while a three way valve is a true mixing valve. It should be noted that when sizing a two way valve, the pressure drop across the cooler must be known, to ensure that the valve coefficient is large enough to pass the required flow. Many systems using two way valves are insufficiently sized. This can result in cooler oil constantly being supplied to the system, since the pressure drop through the cooler is less than the minimum pressure drop through the control valve.

Instrumentation

The instrumentation in any auxiliary system is extremely important in ensuring quick system response, accurate monitoring of system condition and rapid system shutdown in the event of upsets. In this section we will examine stand-by pump start-up operation, critical equipment shutdown and monitoring functions of the instrumentation in the auxiliary system.

Stand-by pump automatic start

As previously mentioned, interrupting pumped flow to critical equipment results in a rapid deterioration in system flow and pressure. Referring again to the concept of an equivalent vessel, the absence of inlet flow to the system while exit flow is continuing will instantaneously produce a pressure drop. This concept is used in a pressure switch that signals the immediate start of the stand-by pump. Practice has shown that locating the pressure switch takeoff as close as possible to the pump discharge results in the quickest response time to start the stand-by pump. Some systems incorporate dual pressure switches, one close to the pump discharge and another up close to the critical equipment. Both switches start the stand-by pump on signal. The pressure setting of the switch is usually set just below the lowest discharge pressure that the pump will produce. In order to ensure the rapid start of the auxiliary pump when required, many systems incorporate an on-off-automatic switch on the auxiliary pump, or on the stand-by pump motor starter for testing the system. It is extremely important that the position of the switch always be in *the automatic mode* during critical equipment operation. It is recommended that an alarm be supplied and annunciated in the event that the auxiliary or stand-by pump is not in the auto position during critical equipment operation.

Critical equipment trip instrumentation

A general critical equipment design philosophy is to avoid trip circuits as much as possible. That is, to only install trip switches in those situations which are absolutely necessary. Typically, auxiliary systems incorporate only one trip function, such as the low lubricating oil trip in lube systems and the low seal oil differentials trip in seal systems. Sometimes a high temperature switch is also installed to trip, but this is not usually the case. The setting of the trip switch is very important. It must be selected such that the equipment will shut down when actuated in order to prevent any long term damage. It must also be selected to prevent spurious, unnecessary trips of the unit, since they are extremely costly in loss of revenue. In addition, the quality of any trip switch is extremely important since this relatively low cost device could cost millions of dollars of lost product revenue per day if it malfunctions. Attention is drawn to correct selection of switch component materials to prevent corrosion or any abnormality that would cause drifting of switch setting and unnecessary unit shutdowns. Again the concept of a system is extremely important to consider. It must be remembered that the trip switch and the shutdown system for the critical equipment must function accurately together in order to terminate equipment operation immediately upon signal from the initiating trip switch. ‘Best practice’ is to use smart (self-diagnostic), triple redundant (two out of three voting) transmitters for all pump start and trip services.

Auxiliary system monitoring

Refer to Figure 7.13.1 and observe the different monitoring and alarm functions normally used in an auxiliary system. In order to ensure reliable auxiliary system operation, the personnel must continuously observe and record any changes in instrument readings and promptly attend to alarms to ensure that the system continues to operate as required. Changes in any of the system instruments indicates a change in the operating condition of the system, which must be followed through to make sure that components are operating as required. As an example, slowly deteriorating lube oil supply pressure could indicate either a valve malfunction, reduction in speed of a main turbine pump driver, excessive pressure drop in the system oil filter or several other types of problem. It is extremely important to maintain a program of auxiliary system instrumentation calibration to ensure all instruments are reading properly. This will aid greatly in determining system malfunctions and assist in the site preventive maintenance program.

Best Practice. 7.14

Install dual SS accumulators in critical equipment lube oil systems to positively prevent unit low oil pressure trips during transient events.

Even a properly designed lube oil system will eventually experience trips during transient events due to the following facts:

- The bypass (backpressure) valve response will change (packing friction)
- The bypass (backpressure) valve sensing line pulsation valve can become clogged
- The auxiliary pump start time will increase (electrical system changes)

Installation of two (2) stainless steel accumulators, each sized for 4 seconds of oil supply, will prevent unit low pressure trips and allow plant personnel to check accumulator pre-charge and bladder condition periodically (every 3 months) without taking the accumulator out of service.

It is also recommended that an orifice bypass line with a globe value be installed around (in parallel to) the accumulator supply line for personnel use to ensure that the accumulator is put back into service slowly to prevent a decrease in oil pressure.

Oil systems can be easily modified for installation of an accumulator during a turnaround.

Lessons Learned

Lube oil systems installed without accumulators will eventually cause critical (un-spared) unit trips that will expose the user to significant revenue losses.

Clients with critical lube oil systems without accumulators will often install them eventually, after experiencing unit trips that can easily justify the modification costs.

Benchmarks

This best practice has been used since the 1990s when FAI performed numerous field audits for auxiliary systems. Installed accumulators immediately increased critical unit MTBFs and increased unit reliability significantly.

B.P. 7.14. Supporting Material

An accumulator is simply a vessel which compensates for rapid short term flow disturbances in the auxiliary system. Most accumulators contain bladders (see Figure 7.14.1). It is important to remember that transient disturbances are often of the order of micro seconds, and usually less than five seconds in duration.

A schematic for a pre-charged accumulator is shown in Figure 7.14.2.

The pre-charge pressure is set at the pressure that the volume of the accumulator flow is required in the system. (This value is usually around 60–70% of the normal header pressure in which the accumulator is installed.) The quantity of oil available from a pre-charged accumulator is extremely low.

As an example, consider a system with a flow capacity of 120 GPM, which has a motor driven auxiliary pump that requires three seconds to attain full speed when started by a pressure switch or transmitter at 140 PSIG. Normal header pressure equals 160 PSIG. Determine the amount of oil that is required to prevent the pump header pressure from falling below 100 PSIG, and the number of pre-charged 10 gallon accumulators required. (See Figure 7.14.3.)

Accumulators are often improperly sized because of the misconception that its stated size is in fact the capacity contained therein. The actual capacity in any accumulator is equal to the internal volume minus the gas volume over the liquid volume. Typically these values are 50% of the stated capacity or less.

System reliability considerations

A number of reliability considerations are worthy of mention concerning auxiliary system control and instrumentation.

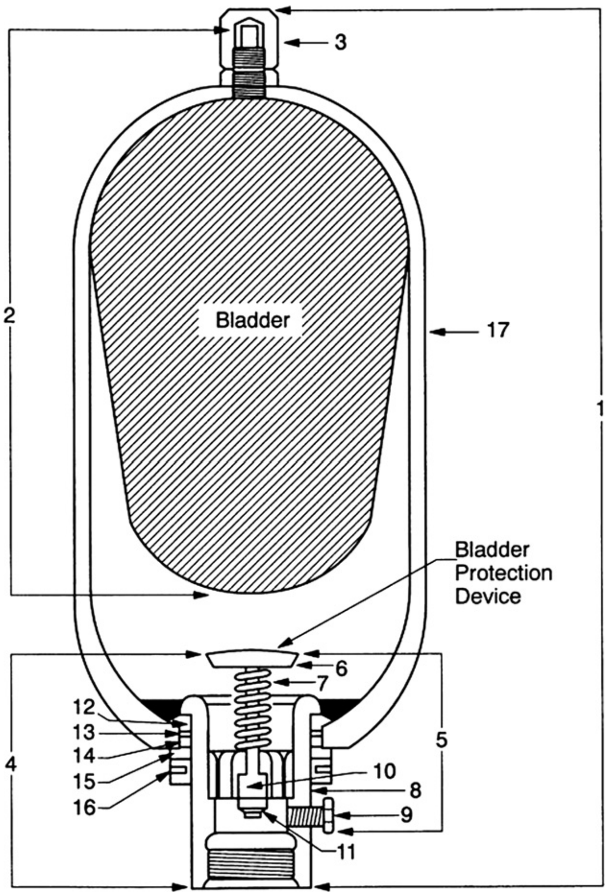


Fig 7.14.1 • Typical oil system accumulator (Courtesy of Greer)

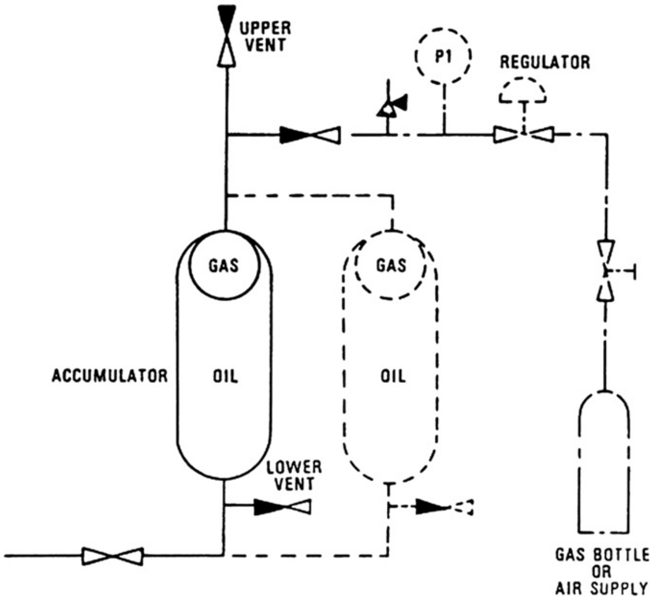


Fig 7.14.2 • Accumulator precharging arrangement (Courtesy of Elliott Co.)

Control valve instability

Control valve instability can be the result of many factors, such as improper valve sizing, improper valve actuators, air in hydraulic lines or water in pneumatic lines. Control valve sensing lines should always be supplied with bleeders to ensure that no liquid is present in pneumatic lines or air in hydraulic lines. The presence of these fluids will usually cause instability in the system. Control valve hunting is usually a result of improper controller setting on systems with pneumatic actuators. Please consult instruction books to ensure that proper settings are maintained. Direct-acting control valves frequently exhibit instabilities (hunting on transient system changes). If checks for air prove inconclusive, it is recommended that a snubber device (mentioned previously) be incorporated in the system to prevent instabilities. Some manufacturers install orifices which sufficiently dampen the system. If systems suddenly act up where problems previously did not exist, any snubber device or orifice installed in the sensor line should be checked immediately for plugging.

Excessive valve stem friction

Control valves should be stroked as frequently as possible, to ensure minimum valve stem friction. Excessive valve stem friction can cause control valve instabilities or unit trips.

Control valve excessive noise or unit trips

Squealing noises suddenly produced from control valves may indicate valve operation at low travel (C_v) conditions. Valves installed in bypass functions that exhibit this characteristic may be signaling excessive flow to the unit. Remember the concept of control valves being crude flow meters. Periodic observation of valve travel during operation of the unit will indicate any significant flow changes.

Given:

- System required flow = 120 GPM
- System pressure at accumulator (at which accumulator effect is desired) = 140 PSIG – 154.7 PSIA (P2)
- Gas precharge pressure (pressure at which accumulator oil flow ceases, assuming system pressure does not fall below this level) = 110 PSIG = 124.7 PSIA (P_1)
- Volume of accumulator = 9 gallons (V_a) (accounts for volume of internal parts)

Determine:

- Amount of oil required
- Number of 10 gallon accumulators required

Amount of oil required:

- System flow per second $= \dfrac{120 \text{ Gal/Min}}{60 \text{ Sec/Min}}$

  = 2 Gal/Sec
- Oil required = 3 Sec. × 2 Gal/Sec

  = 6 Gallons
- Volume of oil entering system for each 10 gallon accumulator.

$$V_{oil} = (V_a)\left[1 - \left(\frac{P_1}{P_2}\right)\right]$$

$$= (9 \text{ Gal})\left[1 - \left(\frac{124.7}{154.7}\right)\right]$$

= 1.75 Gal. per accumulator

- Number of 10 gallon accumulators required

$$\text{Number of 10 gallon accumulators} = \frac{\text{Oil quantity required}}{\text{Quantity available per accum:}} = \frac{6 \text{ Gal:}}{1{:}75 \text{ Gal:}}$$

= 3.42 accumulators required
= four 10 Gal. accumulators

This is a large number of accumulators and is caused by:
The conservative setting of P2 and the neglect of the effect of system control valves and partial auxiliary pump flow during pump acceleration. Let's set P2 just (1 PSIG) below the normal header setting and recalculate the number of accumulators required.

$$V_{oil} = (9 = (9 \text{ Gal})\left[1 - \left(\frac{124.7}{175.7}\right)\right]$$

= 2.6 Gal/accumulator
= 3 accumulators required

The above example demonstrates the importance of properly sizing an accumulator.

Fig 7.14.3 • Accumulator sizing

Control valve sensing lines

Frequently, plugged or closed control valve sensing lines can be a root cause of auxiliary system problems. If a sensing line that is dead-ended (see Figure 7.13.7) is plugged or closed at its source, a bypass valve will not respond to system flow changes and could cause a unit shutdown. Conversely, if a valve sensing line has a bleed orifice back to the reservoir (to ensure proper oil viscosity in low temperature regions), plugging or closing the supply line will cause a bypass valve to fully close, so rendering it inoperable and may force open the relief valve in a positive displacement pump system.

Valve actuator failure modes

Auxiliary system control valve failure modes should be designed to prevent critical equipment shutdown in case of actuator

failure. Operators should observe valve stem travel and pressure gauges to confirm valve actuator condition. In the event of actuator failure, the control valve should be designed for isolation and bypass while on-line.

This design will permit valve or actuator change out without shutting down the critical equipment. During control valve on-line maintenance, an operator should be constantly present to monitor and modulate the control valve manual bypass as required.

Accumulator considerations

Concerning accumulators, checks should be made when unit is shut down for accumulator bladder condition if supplied with bladders. One area which can cause significant problems in auxiliary systems is accumulators that are supplied with a continuous charge – that is, charge lines (nitrogen or air) which come directly from a plant utility system. Any rupture of a diaphragm will provide a means for entry of charge gas directly into the lube system. Most plant utility lines contain pipe scale that could easily plug systems and cause significant critical equipment damage.

In addition, the following reliability factors should be noted (refer to Figure 7.14.2):

- Be sure to install a check valve upstream of the accumulators to ensure all accumulator oil is delivered to the desired components.
- Accumulators should be checked periodically (monthly) for proper pre-charge and bladder condition by isolating and draining the accumulator. Note that the accumulator pre-charge pressure cannot be determined while on line.
- When refilling the accumulators, care must be taken not to suddenly open the supply valve. Best practice is to install an orificed bypass valve to be used for filling the accumulator.
- Best practice is also to install two (2) full size accumulators to ensure that one accumulator is always on line during monthly checks.

Best Practice 7.15

Use SS oil coolers and filters to allow maximum bearing, seal and turbine control component MTBFs (120 months+).

Shell and tube oil coolers typically have the water in the tubes and oil in the shell and are made of carbon steel for cost reasons.

Oil filters are therefore usually positioned downstream of the oil coolers to prevent carbon steel (iron sulfide) particles from entering the machinery components and causing pre-mature wear/failure.

However, it is rare that filters do not experience bypassing around the seals which will expose machine components to early wear and/or failure.

Considering the daily revenue for critical units today (2010), the use of stainless steel coolers and filters can easily be justified and will ensure maximum machine component life.

Lessons Learned

Filters, although positioned downstream of the coolers, do experience bypassing and expose critical machines installed in large plants to extended downtime for component replacement (3–5 days) which results in lost revenue in the millions of USD.

In addition, a complete stainless steel oil system will greatly reduce oil flush time and offer the opportunity for increased unit revenue.

Benchmarks

This best practice has been used since 2000 when daily revenues for medium sized process units exceeded $1,000,000 USD, making it viable to install dual stainless steel coolers and filters in oil consoles used for critical (un-spared) units.

B.P. 7.15. Supporting Material

Referring to the generic definition of an auxiliary oil system, the function of the coolers, filters and transfer valves is '*to continuously provide cool, clean fluid to the critical equipment components*'. Within this definition, '*continuously*' requires that components be dual main and spare filters and coolers. In addition, a means for on-line transfer of one bank of coolers and filters to the stand-by bank is required. This transfer must be accomplished without any significant pressure fluctuations, or flow disturbances in the auxiliary system. In order to ensure pulsation-free transfer, all components involved must be purged of air and must be full of auxiliary fluid. This requires that properly sized vent, bleed and fill lines be incorporated in the system. The function of providing cool oil is naturally that of the coolers. As will be discussed in this section, coolers can be supplied in many different types and configurations. The function of the filters is to supply clean oil. They must continuously ensure that all fluid supplied to the critical equipment is within the system specification of cleanliness.

The sub-systems

Coolers and filters actually are a part of the cooling and filtration subsystems in any auxiliary system. The cooling sub-system consists of the transfer valves, the temperature control valves,

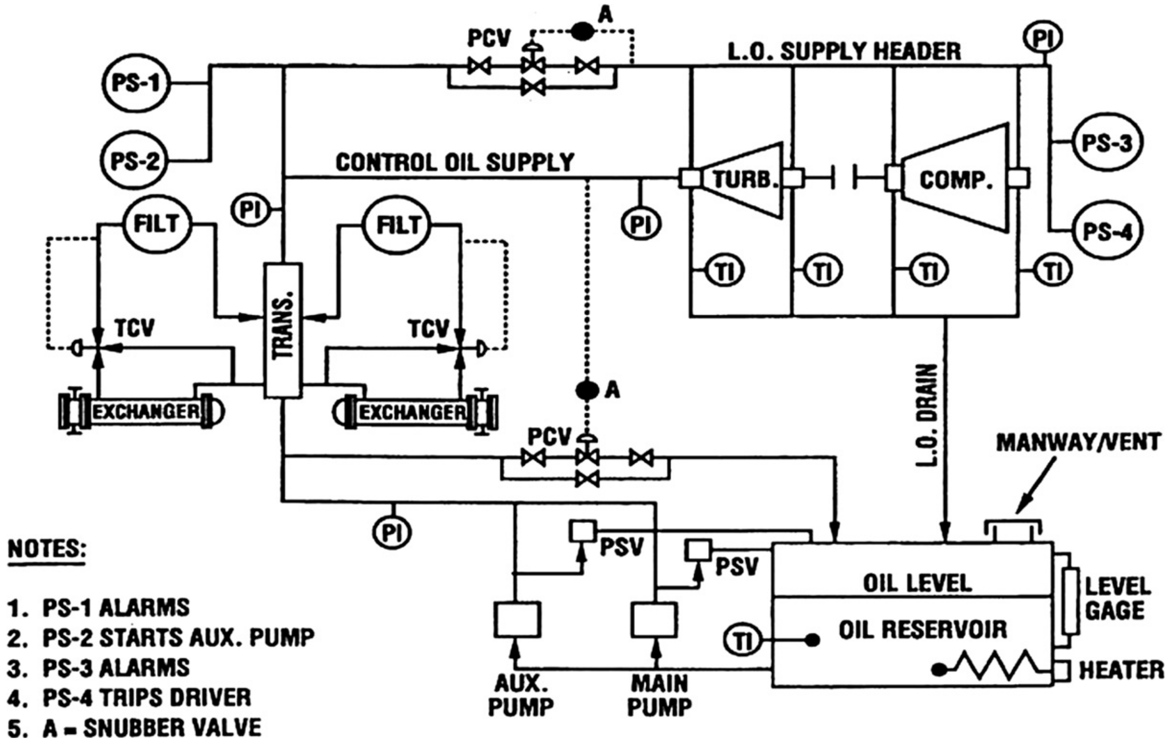


Fig 7.15.1 • Typical lube oil supply system (Courtesy of M.E. Crane, Consultant)

coolers and the cooling medium – whether this is a liquid (from the utility system) or air. If air coolers are required, the electrical utility system reliability will impact the reliability of the coolers and must be considered. The total function of this sub-system, then, is to continuously supply cool fluid at the required temperature to the critical equipment.

The filtration sub-system consists of the filters, the transfer valve and the differential pressure indicators that monitor filter conditions. The function of the filter sub-system is to continuously supply clean fluid at the required filtration level (usually ten (10) microns) to the critical equipment.

Arrangement of components

Various transfer valve and cooler-filter arrangements are utilized in auxiliary systems. A single transfer valve configuration which would incorporate a cooler-filter bank is often used (refer to Figure 7.15.1). If complete interchangability between coolers and filters are required, two transfer valves are utilized.

The location of coolers relative to filters varies with auxiliary system design. If viscous fluids are used, filter location upstream of the cooler is economical, since a lower pressure drop will be experienced and a smaller size filter can be used. This arrangement does not, however, provide absolute protection for the critical equipment. The location of the filter as the last vessel in an auxiliary system prior to supply to the unit is a good idea since it will ensure complete filtration. Coolers downstream of the filter provide a vessel which can contain debris that could break loose under system pulsations and enter critical equipment components, causing significant damage.

The recommendation is to seriously consider filters as being the last element in a system. One other consideration concerning arrangement is the location of the cooler relative to the total auxiliary system pump output. Many systems incorporate coolers for removal of heat of system flow for the critical equipment components only. That is, any bypass flow is not cooled. Consideration must be given to heat generated by pumps, and the amount of flow continuously bypassed. If this amount introduces excessive head load into the system, the cooler should be placed before the bypass valve, thus ensuring constant oil reservoir temperature. We will now direct our attention to the major components in this section.

Transfer valves

The function of the transfer valves in the auxiliary system is to allow transfer from one bank of components (coolers, filters, etc.) to the stand-by bank of components without significant pressure pulsations being introduced into the system. In addition, transfer valves must be designed to positively shut off the unused components to allow for maintenance while the system is still in operation.

Types of transfer valves vary widely. A common type is shown in Figure 7.15.2. The type of valve shown is a six port transfer valve allowing flow into the valve to be diverted either to the left bank or right bank of components as shown in the standard schematic. Other types of transfer valves utilized include a standard globe type valve shown in Figure 7.15.3.

Both types exhibit the characteristic of minimal pressure change when transferring from one bank to the other. The six

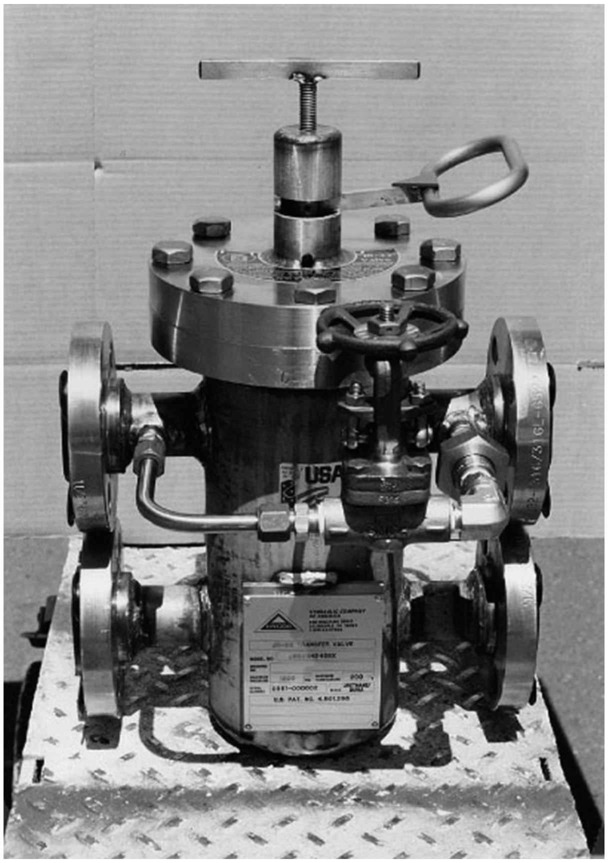


Fig 7.15.2 • Port transfer valve (Courtesy of Hycoa)

ported valve should be designed with a lifting jack when a taper plug is used. This feature allows transfer of valves from one bank to another easily and still ensures tight shutoff of the valve when in its proper position.

Selection of valves should consider material and sizing. Materials of construction should be carbon steel as a minimum, with stainless steel internals. Bronze components should not be considered in systems that can incorporate gas entrained in the system fluid; in such systems brass, copper or bronze is not permitted. Valves are sized to match piping sizes in the system which are normally designed for fluid velocities of four (4) to six (6) feet per second.

Reliability considerations

As previously mentioned, transfer valves that include lifting jacks are susceptible to valve plug damage. Frequently operators do not employ lifting jacks when transferring valves. This type of valve can become bound and even break the stem, thus necessitating critical equipment shutdown. Prior to transferring flow from one bank to another, the bank to be transferred to must be full of auxiliary system fluid and properly vented. If it is not, the capacity of coolers, filters and piping in the bank to be transferred to act as a vessel and consequently reduce flow and pressure to the critical equipment which will cause shutdown of the unit. Also, improperly vented components containing air or gases can cause control valve instabilities which can also lead to equipment shutdown.

Coolers

The coolers form a cooling sub-system whose function is to continuously provide fluid to the critical equipment at the correct temperature. Most types of coolers in use are of a shell and

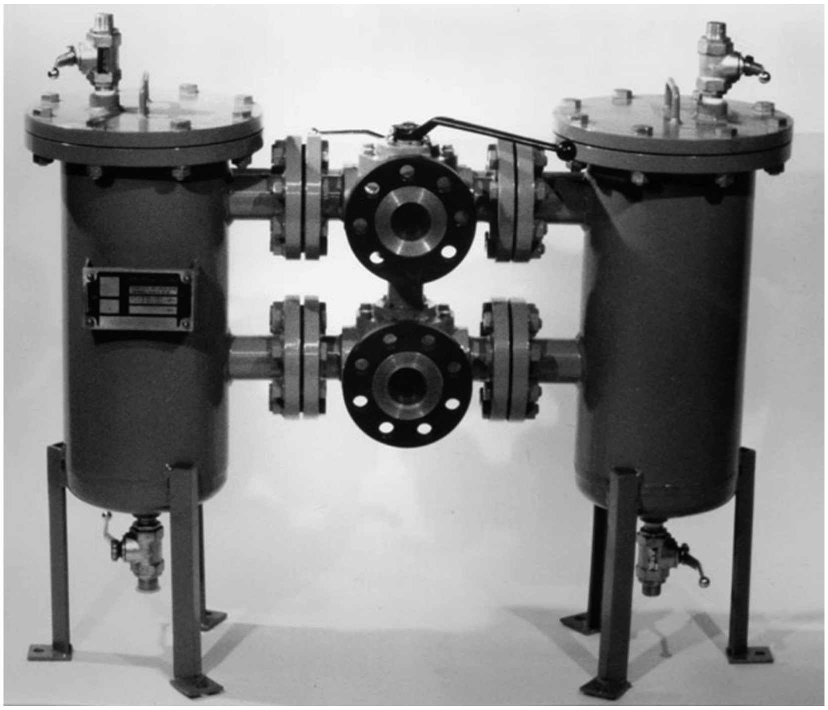


Fig 7.15.3 • 6 way (double, 3 way, flexibly coupled) (Courtesy of Hycoa)

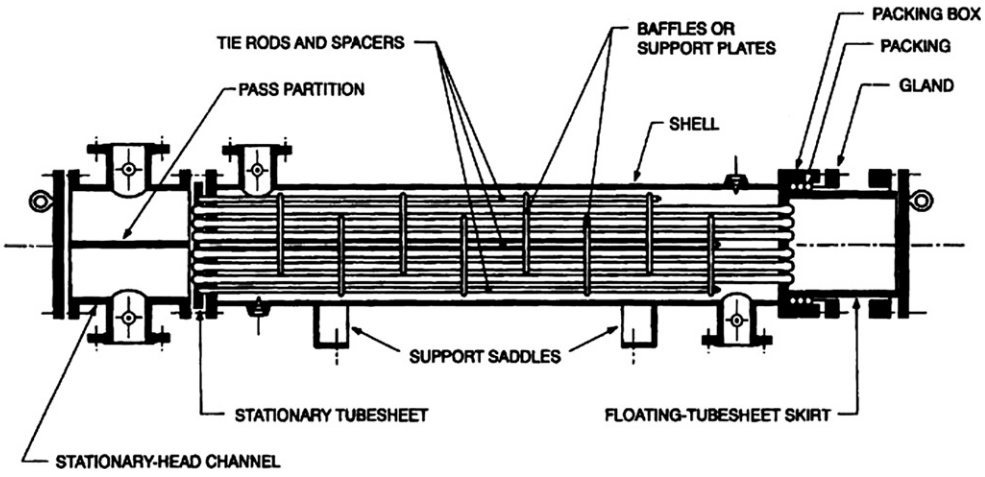


Fig 7.15.4 • Typical shell and tube cooler

tube design, if a suitable cooling liquid is available (refer to Figure 7.15.4).

If water is not available on site, an air cooler (fin fan) is used, and is usually mounted on the auxiliary equipment skid (see Figure 7.15.5).

Cooler selection is based on total auxiliary system heatload. This is obtained by adding individual component heatloads, including the pumps, and incorporating a service factor based on the duty. Typically, a service factor of 10 to 20% excess exchanger surface is used. As in any critical equipment auxiliary system, proper component specification is mandatory. A typical cooler data sheet is shown in Figure 7.15.6.

Attention is drawn to proper specification of the cooler fouling factor. Frequently, the condition of the utility cooling water system for shell and tube exchangers is not properly stated, and excess fouling leads to insufficient cooling capacity available in the system.

As mentioned for pump drivers that incorporate expansion turbines, the same principle applies in this case. That is, the utility cooling water system must be specified for conditions present at the cooler flanges and not at the unit boundary. It is recommended that cooling system calculations for grass root systems and pressure surveys for existing utility systems are performed. In specifying coolers for auxiliary systems, attention should be paid to minimizing gasket and 'O' ring interfaces that could cause leakage of cooling fluid into the auxiliary system. Good auxiliary system design dictates that the auxiliary system fluid always be at a greater pressure than the cooling fluid. This is to ensure that leakage, if it exists, will be from the auxiliary system fluid to the cooling fluid, thereby eliminating the possibility of introducing cooling fluid into the auxiliary system, and providing a means to determine such an abnormality (by observing auxiliary system fluid changes).

Reliability considerations

- Attention should be paid to water side oil cooler pressure drop and the valve position of temperature control of valves, if present, during normal operation to ensure knowledge of cooler performance condition.
- The sudden decrease of oil reservoir level in systems where cooler system pressure exceeds cooling fluid pressure indicates the possibility of internal cooler leakage. In

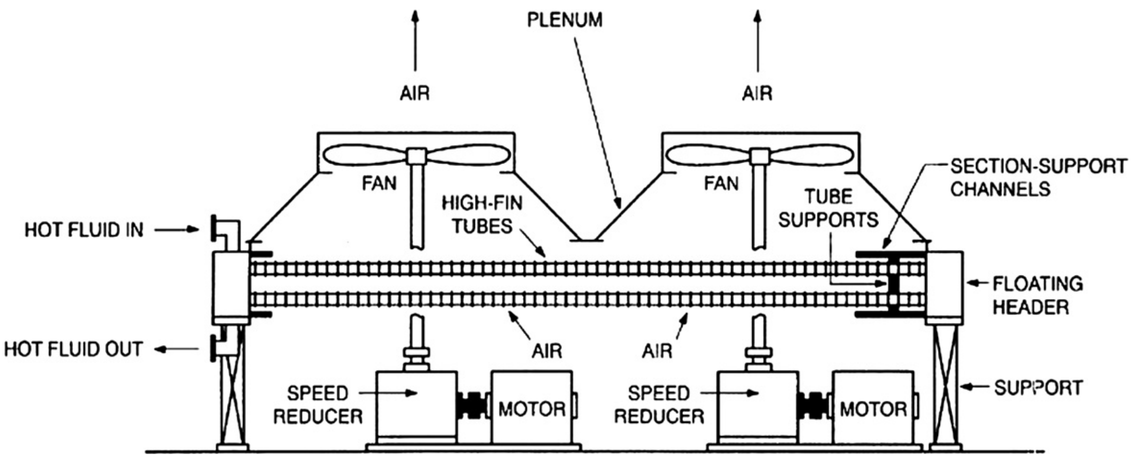


Fig 7.15.5 • Typical air cooled heat exchanger

1			Job No.	
2	Customer		Reference No.	
3	Address		Proposal No.	
4	Plant Location		Date	Rev.
5	Service of Unit		Item No.	
6	Size Type	(Horiz) (Vert)	Connected in	Parallel Series
7	Surf./Unit (Gross) (Eff.) ft²	Shells/Unit	Surf./Shell (Gross) (Eff.) ft²	
8	PERFORMANCE OF ONE UNIT			
9	Fluid Allocation	Shell Side	Tube Side	
10	Fluid Name			
11	Fluid Quantity, Total lb/h			
12	Vapor (In/Out)			
13	Liquid			
14	Steam			
15	Water			
16	Noncondensable			
17	Temperature (In/Out) °F			
18	Density lb/ft³			
19	Viscosity, Liquid cP			
20	Molecular Weight, Vapor			
21	Molecular Weight, Noncondensable			
22	Specific Heat Capacity Btu/lb°F			
23	Thermal Conductivity Btu/h·ft·°F			
24	Latent Heat Btu/lb @ °F			
25	Inlet Pressure	(psig) (psia)	(psig) (psia)	
26	Velocity ft/s			
27	Pressure Drop, Allow./Calc. psi	/	/	
28	Fouling Resistance (Min.) h·ft²·°F/Btu			
29	Heat Exchanged	Btu/h; MTD (Corrected) °F		
30	Transfer Rate, Btu/h·ft²·°F Service	Clean		
31	CONSTRUCTION OF ONE SHELL			Sketch (Bundle/Nozzle Orientation)
32		Shell Side	Tube Side	
33	Design/Test Pressure psig	/	/	
34	Design Temperature °F			
35	No. Passes per Shell			
36	Corrosion Allowance in.			
37	Connections Size and Rating — In			
38	Out			
39	Intermediate			
40	Tube No. OD in. Thk (Min)(Avg)	in.; Length	Pitch in.	
41	Tube Material (Wld)(Smls)			
42	Shell (ID) (OD) in.	Shell Cover	(Integ.) (Remov.)	
43	Channel or Bonnet	Channel Cover		
44	Tubesheet-Stationary	Tubesheet-Floating		
45	Floating Head Cover	Impingement Protection		
46	Baffles-Cross Type	% Cut (Diam) (Area)	Spacing: c/c	Inlet in.
47	Baffles-Long	Seal Type		
48	Supports-Tube	U-Bend	Type	
49	Bypass Seal: Peripheral (Yes) (No). Pass Lane (Yes) (No): Tube-Tubesheet Joint			
50	Expansion Joint	Type		
51	ρv^2-Inlet Nozzle	Bundle Entrance	Bundle Exit	
52	Gaskets-Shell Side	Tube Side	Floating Head	
53	Code: ASME		Stamp (Yes) (No): TEMA Class	
54	Weights/Shell	Filled with Water	Bundle	lb
55	Remarks			
56				
57				

Fig 7.15.6 • Shell and tube exchanger data sheet

this case, operation should be transferred to the stand-by cooler and the affected cooler checked for internal condition.

- Systems incorporating air (fin fan) coolers should be sized conservatively, such that full cooling capacity is available in high ambient temperature conditions utilizing a realistic fouling factor. Experience has shown that air coolers are often undersized. In addition, air coolers should be furnished with dual channels, such that two separate cooler banks, as in the case of shell and tube coolers, are present.

Filters

The function of filters is to continuously provide clean auxiliary fluid to the critical equipment. Let's investigate that definition in greater detail. A typical hydrodynamic oil film bearing will have a continuous oil film on the order of 0.001 inch (20 to 25 microns) between the shaft and bearing surface at the load point. In order to ensure continuous trouble free operation of the bearing, the filters in the system must continuously prevent the entrance of any particle equal to or greater than 20 to 25 microns clearance into the oil film between the shaft and the bearing. A typical filtration specification for auxiliary systems is ten microns absolute particle size – that is, the greatest size of any solid particle in the oil film should be ten microns. This is provided of course, that the filter design does not allow the bypass of any particles. In general, filtration specifications of 10 to 25 microns are typical of the maximum particle sizes permitted through the filters. It is important to remember that the filter sub-system consists of the transfer valves, filters, differential pressure indication and alarm across the filter element.

Fig 7.15.7 • Surface type filter (Courtesy of Hilco)

Fig 7.15.8 • Depth type filter (Courtesy of Filterrite)

Increasing the filter debris load will increase filter differential pressure, thereby signaling the need for transfer to the other filter bank to allow maintenance to change the dirty filter elements.

There are essentially two main types of filters elements employed in critical equipment auxiliary systems. The first type is shown in Figure 7.15.7, and is a surface filter, and the second type, in Figure 7.15.8, is the depth type filter. Different users and vendors have different preferences. Both filters are efficient and are employed in critical equipment systems.

The depth type filter usually results in a larger filter for the same flow capacity. An assembly of a depth type filter is shown in Figure 7.15.9. The most important consideration for any filter, regardless of its type, is to ensure that it is designed and manufactured to positively eliminate the possibility of auxiliary system fluid bypass. Any bypassing of oil to be filtered will result in critical equipment wear and damage. Even though a filter is designed to eliminate bypass, care must be exercised during maintenance to ensure that all filter cartridges and sealing components are properly installed. In addition, during maintenance, care must be taken to preclude the possibility of dirt and debris falling from used filter cartridges and entering the system. In this case, reintroduction of this filter debris into the system will cause debris to directly enter the equipment components. Recent developments in filter design include modifications to ensure complete sealing and also minimize the possibility of entrance of debris into the system during maintenance.

Also of importance is the selection of filter cartridge material and configuration. Attention must be paid to the auxiliary system fluid and the selection of filter cartridge material to ensure deterioration of cartridges does not take place during operation. Also of note is the recommendation to minimize cartridge interfaces as much as possible in a filter. Use of elements stacked one high instead of typical multi-stacked elements minimizes potential for internal filter bypass. Note that most current filter applications can be modified to eliminate multi-stacked elements. The specification of clean filter pressure drop is important to ensure proper filtration and minimum filter maintenance. It is suggested that clean filter pressure drop is limited to a maximum of five (5) psi. Attention must be paid to maximum filter cartridge allowable pressure drop and differential filter alarms set at a level to preclude the possibility of filter collapse.

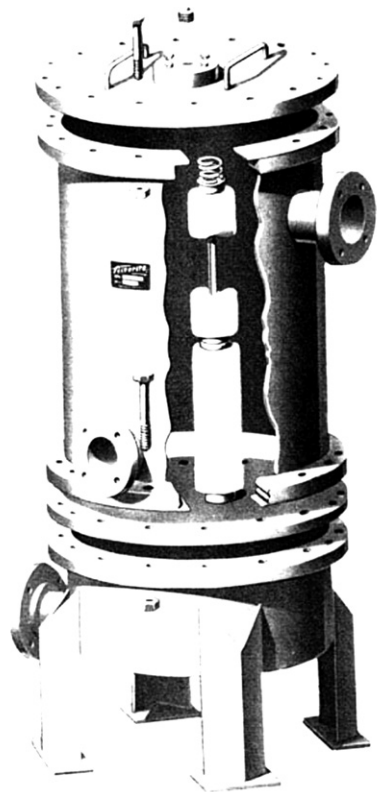


Fig 7.15.9 • Depth type filter assembly drawing (Courtesy of Filterrite)

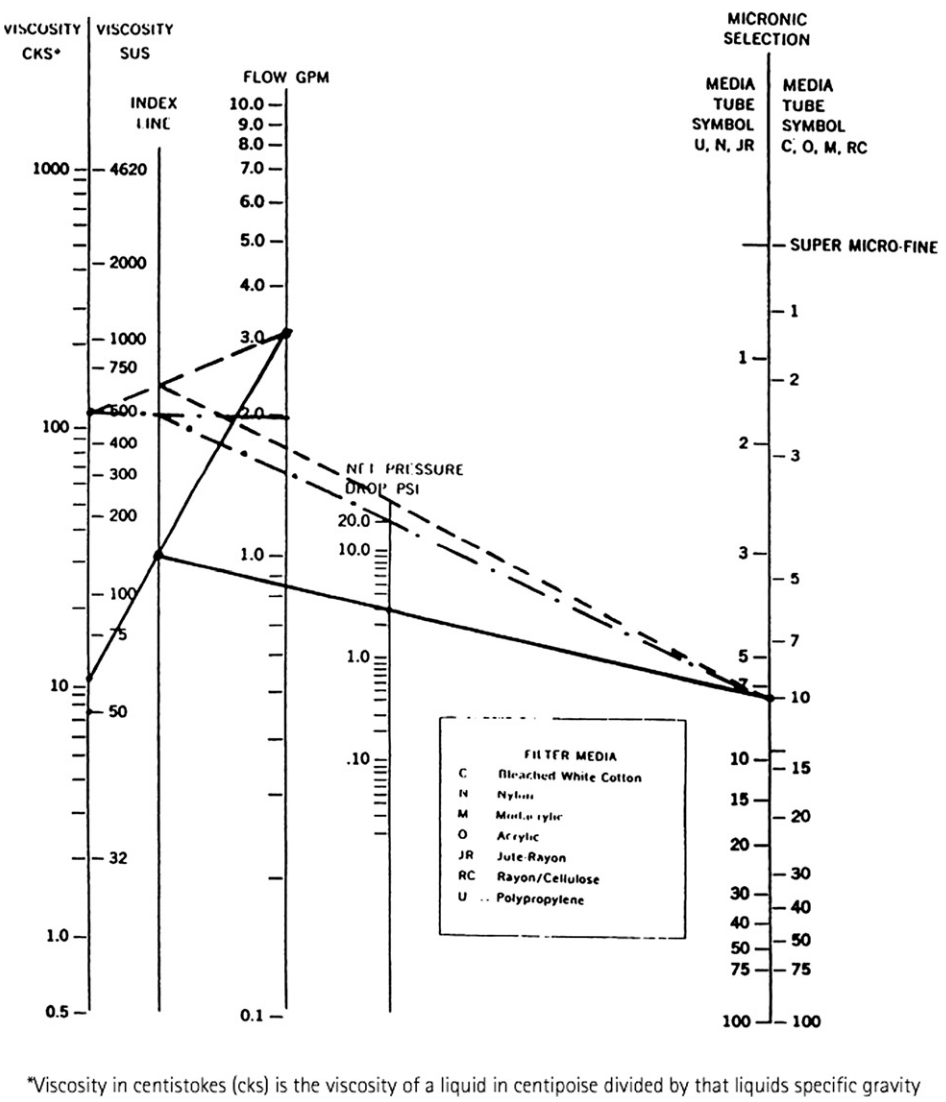


Fig 7.15.10 • Filter sizing nomograph
(Courtesy of Filterrite)

A sizing exercise for our example is performed below for a depth type filter. A sizing chart is shown in Figure 7.15.10. Note that the sizing must include considerations for the high viscosity (1,000 SSU).

Depth type filter sizing example

Given:

- Light turbine oil
- Normal viscosity = 60 SSU @ 150°F
- Max viscosity = 500 SSU @ 55°F
- Oil flow = 60 GPM
- Filtration required = 10 Microns
- Filter material = Orlon
- Clean pressure drop = 3 psi

Determine:

- Number of filter cartridges required
- Pressure drop @ 500 SSU operation

1. Determine number of cartridges

(Refer to Figure 7.15.10)

At 60 SSU, 3 psi Δp for 10 micron 'Orlon' cartridges, flow rate = 3.1 GPM/cartridge

$$\text{Number of cartridges required} = \frac{60\ \text{GPM}}{3.1\ \text{GPM/1 Cartridge}} = 19.5 = 20\ \text{Cartridges}$$

2. Pressure drop @ 500 SSU operation (refer to dotted lines in Figure 7.15.10) at 500 SSU, 3.1 GPM/Cartridge Δp exceeds 20 psi.

In the above example, to limit the filter Δp to 20 psi at 500 SSU, a 2 GPM cartridge would have to have been selected. In this case, the number of cartridges used would be:

$$\text{Number of cartridges} = \frac{60\ \text{GPM}}{2\ \text{GPM/Cartridge}} = 30$$

Reliability considerations

Several considerations concerning filters are noted below.

A. *Filter material* – As mentioned above, it is important to select proper filter material. Experience has shown that some filter materials will be shed during operation and thus present a risk of critical equipment component damage. Consideration should be given to elements that are impervious to the possibility of any foreign substance in the auxiliary fluid steam. Orlon has proven to be a good substance for filter element material. In addition, paper filters should not be used in system that can contain water (steam turbine units or in high humidity areas). Experience has shown that water will cause paper elements to swell, thus increasing filter pressure drop and giving the appearance of plugged filters.

B. *Filter high viscosity operation* – Attention is drawn again to operation of filters at high viscosities (usually during cold start-up). Excessive filter differential pressure can lead to its collapse. The system fluid should be preheated (additional reservoir heating or possible cooler heating) prior to system operation if the fluid is not at the minimum starting temperature recommended by the vendor. Many unit trips have been caused by transferring to a cold filter. Continuously open, orificed vent lines, leading back to the reservoir have become a 'best practice' in cold climates.

C. *Filter cartridge configuration* – The use of filter elements stacked one high has proven to be very successful in eliminating internal filter bypass. It is suggested that filter vendors be contacted for possible retrofits where filter elements are not stacked in this way. End seals on filter cartridges must be securely locked in place on every cartridge to prevent filtered fluid bypass. Prior to closing filters a thorough check of all cartridge end seals should be made.

D. *Filter cartridge changeout* – Care must be taken during dirty filter disassembly to preclude the possibility of entrance of debris into the system. If possible, the bottom head of the filter should be disconnected and thoroughly cleaned or back-flushed to ensure clean fluid only enters critical equipment. In designing new auxiliary systems the possibility of removable bottom filter heads should be investigated for easy cleaning.

E. *Filter housing filling and venting* – Following filter cartridge change out, the filter housing should be immediately filled using the system fill line (if furnished), or an external source of clean filtered fluid that meets the auxiliary system specification. Note that using the system fill line requires care to prevent a sudden opening of the fill valve that could cause a system pressure upset, which in turn could trip the critical equipment unit. All fill lines should be provided with properly sized orifices. However, they have been known to disappear from time to time, or have not been sized to prevent significant system pressure reductions when opened quickly.

Best Practice 7.16

Require the FAT (factory acceptance test) for the oil console to duplicate field conditions as closely as possible and record response times for transients (main pump trip and two pump operation) to ensure optimum oil system field reliability.

As a minimum, the following items should be included in the FAT:

- Auto start of the auxiliary pump
- Two pump operation
- Relief valve checks
- Bypass (backpressure) valve proper valve position and stability
- Transfer valve operation
- Cooler tube leak check
- Filter pressure drop and particle check for bypassing
- Accumulator pre-charge and bladder condition (if applicable)
- Supply valve(s) – proper valve position and stability
- Proper supply flow, pressure and temperature
- Chart recorder for all transient checks (pump trip and two pump operation) and transfer valve check, confirming that oil supply pressure during the transient event does not fall to the trip setting.

Lessons Learned

Failure to completely check all oil system component functions during the FAT will result in delayed start-up and possible lower than anticipated unit reliability for the life of the process unit.

Benchmarks

This best practice has been followed since the 1970s, and has resulted in oil systems and field unit operation of the highest possible reliability.

B.P. 7.16. Supporting Material

Factory testing and inspection

Having properly specified, designed and manufactured the unit, its proper arrangement and operation must be confirmed. This is accomplished during factory testing and inspection. The objectives of this phase are to confirm the proper arrangement details and functional operation of the equipment.

Test agenda

The equipment must be thoroughly tested prior to field installation. The test should confirm the functional operation of all

components as they will operate in the field. In order to ensure a valid factory test, a test agenda should be prepared approximately two months before test date and reviewed by the equipment purchaser. A typical test agenda outline is included at the end of this section. In addition, a typical auxiliary system test agenda is included courtesy of G.J. Oliver, Inc. The test agenda should endeavor to confirm the functional operation of every component in the system. Specific areas of concern are discussed below.

Flushing

Component system flushing is required as an inspection point and should be accepted prior to the initiation of the test. Additionally, all test agendas should be structured such that a limit for each item to be tested is specifically defined. The flushing acceptance criteria must be mutually agreed upon and be adhered to during the review of flushing the operation. Refer to a typical field flushing procedure in B.P. 7.18.

Confirm arrangement details

Prior to the commencement of the test, make sure that the arrangement of all components, controls and instruments agrees with the design requirements. Any discrepancies should be corrected prior to functional test.

Confirm proper test fluid and capacity are present to prior to initiation of the test.

Temporary test setup

If it is necessary to import utilities or switches for the test that are not normally present, as in the case of steam turbine steam generator and control switches, these items must be confirmed prior to test initiation. In addition, a means of confirming proper flow rates, temperature and pressure during the test must be provided. Supply lines to the unit must be provided with properly size orifices to duplicate unit flow requirements.

Functional testing

Having confirmed proper test setup, calibration of all instrumentation, proper fluid and test instrumentation, the functional test is now ready to be performed. As a minimum the following tests should be performed on any auxiliary system console:

- Relief valve test (if supplied)
- Transfer valve test
- Auto start test of auxiliary pump with the following conditions:
- Main pump tripped
- Two pump operation (main pump in operation, standby pump started)

During all functional testing, any system pulsations or pressure drops above specified values are reason for non-acceptance of test. All components not meeting requirements must be corrected and units must be completely retested.

The value of testing the auxiliary console with the unit should be seriously considered. Since the console and the unit piping from the specific auxiliary system, there is a significant benefit to testing both together. Additional costs of such a test should be evaluated against the potential reduction of reliability and loss of operation time in the field if any malfunctions exist that were not determined by test of the console alone.

Functional lube/seal system test procedure outline

Objective:	To confirm proper functional operation of the entire system prior to equipment start-up
Procedure format:	Detail each test requirement. Specifically note required functions/set points of each component. Record actual functions/set points and *all* modifications made.
Note:	All testing to be performed *without* the unit in operation.

I. Preparation

A. Confirm proper oil type and reservoir level.
B. Confirm system flush is approved and *all* flushing screens are removed.
C. Confirm all system utilities are operational (air, water, steam, electrical).
D. Any required temporary nitrogen supplies should be connected.
E. All instrumentation must be calibrated and control valves properly set.
F. Entire system must be properly vented.

II. Test procedure

A. Oil reservoir
  1. Confirm proper heater operation.
  2. Check reservoir level switch and any other components (TIs, vent, blowers, etc.).
B. Main pump unit
  1. Acceptable pump and driver vibration
  2. Absence of cavitation
  3. Pump and driver acceptable bearing temperature
  4. Driver governor and safety checks (uncoupled) if driver is a steam turbine
C. Auxiliary lube pump unit: same procedure as item B above.
D. Relief valve set point and non-chatter check.
E. Operate main pump unit and confirm all pressures, differential pressures, temperatures and flows are as specified on the system schematic and/or bill of material.
F. Confirm proper accumulator pre-charge (if applicable).
G. Confirm proper set point annunciation and/or action of *all* pressure, differential pressure and temperature switches.
H. Switch transfer valves from bank ‘A’ to bank ‘B’ and confirm pressure fluctuation does not actuate any switches.
I. Trip main pump and confirm auxiliary pump starts without actuation of any trip valves or valve instability.

Note: Pressure spike should be a minimum of 30% above any trip settings

J. Repeat Step I above but slowly reduce main pump speed (if steam turbine) and confirm proper operation.

K. Simulate maximum control oil transient flow requirement (if applicable) and confirm auxiliary pump does not start.
L. Start auxiliary pump, with main pump operating, and confirm control valve and/or relief valve stability.

Note: Some systems are designed to *not* lift relief valves during two pump operation.

III. Corrective action

A. Failure to meet any requirements in Section II requires corrective action and re-test.
B. Specifically note corrective action.
C. Sign off procedure as acceptable to operate.

Typical oil console test agenda (Courtesy of G.J. Oliver, Inc.)

I.

A. Flushing oil
1. Viscosity – 150–185 SSU @ 100°F
2. Flash point – 385°F
3. Oil soluble – yes
4. Rust inhibiting – yes
5. Cleanliness – 10 microns maximum

Reuse of oil should only be done if purified and tested after each use to determine if the properties are still satisfactory.

B. Flushing oil temperature
1. The flushing oil in the system must be brought up to temperature (120–130°F) as soon as possible. The oil temperature must be maintained at this range after the initial flush. Be careful not to exceed safe limits for system components, paying particular attention to pumps.

C. Flushing flow
1. Initial flushing flow in lines should be equal to or more than the contract design oil flow for the lines. This can be accomplished if two pump operation is used. In no case should the initial flushing flow in a line be less than the line's design flow.
2. Subsequent flushing flow in screened lines shall be at maximum design flow only since greater flows could damage screen.

D. Flow measurement
1. Oil flow at oil supply connections shall be measured with flow meters.

E. Filtration
1. Flushing oil shall be filtered to 10 micron nominal maximum before entering system or system reservoir.
2. G.J.O. standard filters shall be used while flushing. After flushing, contract elements must be installed.
3. Flushing screens shall be flat or cone SS 100 mesh screens (plain weave, .004" wire dia. with 0.0059" square openings) at each oil supply connection.

F. System filter delta p delta p
1. The filter ΔP must always be observed whenever starting a pump or switching a transfer valve. An excessive ΔP spike would require checking for filter element damage and modifying pump start-up procedure to avoid excessive spikes.

G. Temporary piping (jumpers)
1. All temporary piping used must have all particles such as weld slag or splatter, sand, grit, etc., removed and piping must be cleaned and pickled. The use of clean flexible hose is permissible.
2. Temporary connections into the system reservoir must be sealed against ingress of dirt.
3. Temporary return lines should be elevated such that the filters will remain full when pumps are turned off rather than allowing them to empty by gravity. This will greatly reduce possibility of filter damage when starting pumps and may eliminate need to continually vent filters on start-up during flushing operations.

H. Flushing procedure
1. GJO QC inspector to verify carbon steel pipe has been cleaned and pickled prior to flushing system.
2. Remove all shop test orifices from oil supply lines if present and remove transfer valve plug(s), clean valve if necessary.
3. Install GJO filter elements if necessary and install screen @ reservoir return flange only.
4. Inspect reservoir for cleanliness and clean if necessary.
5. Fill reservoir with specified flushing oil through a 10 nominal micron filter, seal all reservoir openings.
6. Close valves in valved orificed bleed lines that return to reservoir from instruments, etc.
7. Specified flow conditions, oil temperature should be increasing toward design during initial flushing.
   a. Flush all supply lines; be sure to include each side of a cooler or filter set and control valve hand bypass piping. Flush each line for one hour. The total flushing time will vary depending on number of sections to be isolated to obtain maximum flow.
   b. At one-half hour intervals the pumps are to be turned off and left idle for at least five (5) minutes after coast down. Flowing pipes must be vigorously hammered while pumps are running. Check flushing screen at reservoir return, clean if necessary.
8. Install flushing screens at all oil supply points.
9. Flush systems (screens installed) at specified flow and temperature conditions for one hour. After one-half hour switch all transfer valves to catch both sides of cooler and filter sets (if applicable). The piping must be hammered continuously.
10. Shut down pumps, remove and inspect screens; contamination levels on screens may indicate areas where extra flushing efforts might be directed or if pipe recleaning might be necessary before continuing.
11. Flush system with no screens in oil supply lines at specified flow and temperature conditions for 8 to 24 hours as required. Include the following:
    a. Hammer all pipes at one hour intervals while pumps are operating.
    b. Turn off pumps at two hour intervals and let set idle for five (5) minutes after down before restarting.
    c. Switch cooler and filter transfer valves every two hours (if applicable).

12. Install screens in oil supply lines and flush at specified flow and temperature conditions for one (1) hour. Hammer pipe frequently.
13. Shut down pumps, remove and inspect screens. System and screens shall meet cleanliness requirements of API 614 2nd Edition, sections 4.3.3.7.1 and 4.3.3.7.2.
14. If screens fail to pass the cleanliness test in step 13 then steps 11, 12 and 13 must be repeated until the screens indicate acceptable dirt levels. Good judgment should be used in length of time used in step #11.
15. Remove jumpers and inspect any components jumpered out during flush. Components must be cleaned and preserved before putting them back in system.
16. Drain and clean reservoir, seal all openings and prepare for shipment.
17. Inspect filter interior, clean out any sludge or dirt in filter casing and install contract elements.
18. Inspect transfer valve intervals for dirt (plug must be pulled). If dirty, clean and preserve prior to re-assembly.

II.

A. Testing

1. Cleanliness, hydrostatic and noise testing is not covered under this specification.
2. The system is to be checked by QC for proper piping arrangements, component installation and identification to ensure compliance with the systems diagram and bill of material.
3. Component instruction manuals must be checked by test personnel to ensure proper adjustments, lubrication, etc.
4. All oil leaks shall be corrected before operational tests are started. This does not include intended oil leakages as from packed seals or minor seepage from seals on oil pumps.
5. All tests are to be conducted with pressures and flows set within +/− 5% tolerance of the requirements.
6. Oil having a viscosity of approximately 150 SSU and 100°F. shall be used unless otherwise specified.
7. Oil temperature must be maintained at 110 to 120°F. During operational tests, if system is tested.
8. Contract switches which are mounted on consoles or equipment (press/temp. and level) shall be demonstrated to show set points and to be recorded.
9. It will be the responsibility of G.J. Oliver at which the operational test is conducted to provide all non-contract test instrumentation, utilities, power sources, etc. necessary to comply with the contract test agenda. These items are to be installed and ready for test before personnel are called for test witness.
10. The sequence of operational testing and specific comments relating to the individual tests shall be as follows:
    - **a.** Relief valve test — all relief valves on system shall be tested to demonstrate capability of relief at full flow required by system design; set press/accumulation press record shall be documented.
    - **b.** Transfer valve test — all transfer valves on system shall be tested to demonstrate maintenance of flow through transfer function and for minimal/no leakage to shut-off side of valve. Test shall be done at normal supply conditions.
    - **c.** Running test — a running test shall be conducted for a period of four (4) hours unless otherwise specified. During the course of the test, periodic checks shall be made to ensure stabilized conditions have been maintained within tolerances specified in para. 5.
11. The maximum flow conditions test and transfer valve test must all be performed successfully with equipment adjustments in the same position. If minor adjustments are made during any one of these tests, then any of the four tests made prior to the adjustment must be repeated using the new adjustment. If however, the adjustments are known not to affect prior tests, then prior test need not be repeated if approved by customer.
12. The transfer valve test and running test shall be conducted with equipment and lines operating at normal oil flow conditions.
13. All G.J.O. instrumentation that is used for testing console shall be calibrated with date of calibration.
14. Customers are to be notified at least 5–10 days prior to test schedule.
15. Data recorded for formal presentation shall be per the 'systems test data sheets for operational tests', plus any additional data as required by the test agenda.

Best Practice 7.17

Require triple modular redundant (TMR) transmitters for all shutdown functions in new project work or field modifications, to maximize serviced unit reliability.

TMR shutdown functions for two-out-of-three voting will positively eliminate shutdown instrumentation related failures and prevent spurious shutdowns.

It is recommended that existing systems be modified for TMR if any trip system un-scheduled shutdowns have occurred.

Lessons Learned

Oil systems not provided with TMR shutdown logic experience lower reliability than TMR systems and corresponding lower serviced unit reliability.

FAI performs many auxiliary system field audits. In the majority of these, listed reliability issues include unscheduled shutdowns caused by spurious low oil pressure shutdowns.

Benchmarks

This best practice has been used since the mid-1990s, resulting in oil systems and serviced units of the highest reliability (99.7% or greater).

B.P. 7.17. Supporting Material

Prior to the use of smart transmitters, single switches were used for shutdown logic on critical equipment trains. PM schedules usually were at three (3) month intervals to check switch settings on-line, but were usually ignored because they had caused unit trips in the past.

The use of TMR logic for all oil system shutdown logic ensures accurate response for shutdowns and prevents spurious unit trips due to transmitter malfunction.

Best Practice 7.18

Utilize a 'best practice' oil flush program to drastically reduce flush time.

Oil system flushes during construction and turnarounds can be very long (weeks or months) and subject the plant to lost revenue by postponing plant start-ups.

The 'best practice' oil flush procedure contained in the supporting information, developed by Merle Crane – Consultant, incorporates seldom-used procedures (nitrogen bubbling, oil cooler isolation and drain line monitoring) that reduces flushing times to less than a week if followed completely.

In addition, unlike most flushing procedures, it will produce an oil system that will not require frequent on line filter changeovers or expose the machinery components to lower MTBFs.

Lessons Learned

Poorly designed and/or implemented oil system field flushes result in significantly extended start-up times and usually are not followed completely after the original flush schedule time has been exceeded.

I have continually experienced poor oil flush procedures that have extended planned start-ups and/or have caused bearing wear and sometimes failure.

A recent experience (2010) resulted in bearing and rotor replacement, causing a three (3) month delay in start-up after over a month of oil flushing. The location of flush screens was determined by looking at the P&ID, and not the piping diagram, or going to the field to see the actual piping arrangement. The screens for the compressor were installed in a vertical flange. When the oil system was shut down for screen inspection, most of the debris fell by gravity off the screen making the acceptance of the flush premature and leaving significant debris in the supply pipe. This entered and failed both journal bearings and damaged the shaft. Note that a spare rotor was not supplied since there were four similar units installed.

Benchmarks

I have used this 'best practice' oil flush procedure since the mid-1980s when it was developed by Merle Crane – Consultant.

B.P. 7.18. Supporting Material

Auxiliary system flushing

One of the most important pre-start-up operations for an auxiliary system is flushing. The object of flushing is to render the entire auxiliary system free of excessive particles that could enter into critical equipment components, thus causing significant equipment damage. To ensure efficient field flushing preparation, understanding of the entire system and monitoring of flushing operations is required. An example of a field-proven flushing procedure is noted at the end of this section.

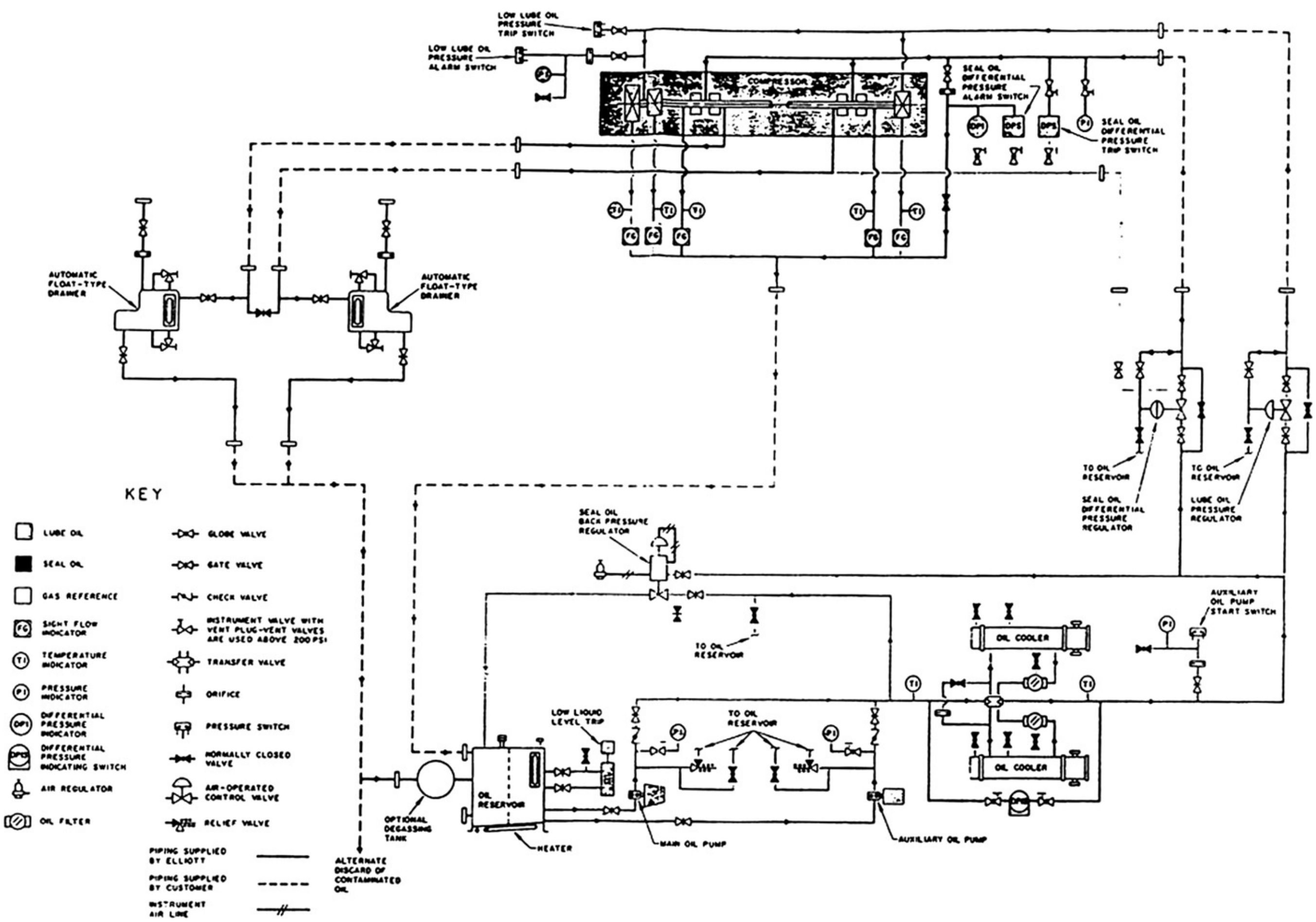


Fig 7.18.1 • API 614 lube-seal oil system for ISO-carbon seals (Courtesy of Elliott Co.)

Specific system design

A schematic showing the entire auxiliary system must be reviewed prior to establishing the flushing procedure (refer to Figure 7.18.1).

Each system will include an auxiliary system console, interconnecting piping, unit piping and drain piping. In the case of liquid seal systems, drain side components will also be included (automatic seal oil drainers degassing tanks, etc.). Review of the specific system is essential to determine the location of flushing screens and sequence of the flushing operation.

Flushing

The basic philosophy of any flushing system should be to avoid duplication in flushing that merely prolongs the action and minimizes the effect. Therefore, any possible pre-cleaning prior to installation is advisable. As an example, all interconnecting piping should undergo thorough cleaning when received in the field. Once clean, piping should be treated in 'white room conditions' prior to installation. Any special care taken at this time will significantly reduce flushing time. Considering the volume of interconnecting pipe, its cleanliness is a significant contributor to minimizing flushing times. Since the majority of interconnecting pipe volume exists in atmospheric drain pipe, any dirt in the unwetted (top) portion of the pipe will cause significant problems during the flushing operation, since these pipes are not pressurized.

Many flushing operations have experienced setbacks as the result of a high liquid level in the drain lines, which causes a significant amount of debris to appear in the flushing screens when the system appeared to be ready for acceptance.

Flushing loops

Once the entire system is installed, flushing should proceed in sequence using flushing loops. Begin at the pumps with the pump loop (pumps through relief valves), and proceed to the bypass pump loop, the cooler loop etc., until the console is relatively clean. This exercise will ensure the removal of large amounts of debris and minimize the possibility of flushing screen breakage. It must be noted that any special care taken from the point of manufacture to start-up in terms of preservation will be appreciated at this point.

Helpful hints

If possible, a pressure tap should be present upstream of flushing screens to determine changes in pressure, thereby signaling the

need to change screens. The specific flushing procedure will detail screen location, size and changing times.

The monitoring of the operation and record keeping is of utmost importance to the effectiveness of any flushing operation. A dedicated log book for each flushed console should be maintained, and all entries should detail the condition of flushing screens and mode of operation. This book can be extremely useful in determining the extent of the flushing operation and predicting the remaining flushing time. Samples of screens should be kept for review as the flushing procedure progress.

All unit piping should be flushed at high fluid velocities. This can be accomplished by operating both supply pumps, using a temporary high flow pump and/or installing temporary valves between each supply point and the drain header. All critical equipment components (bearing, seals, etc.) should be initially bypassed unit supply piping is deemed to be clean.

Special tools and materials will be required and must be made available. Clean oil should be available for the final fill.

- Dust and lint-free rags should only be used in cleaning major components (reservoirs, etc.).
- Hammers and vibrators should be available for mechanical vibration of piping to break loose debris.
- Bottled nitrogen or a clean air supply should be available for bubbling to minimize flushing time.
- An ample supply of stainless steel screen at required mesh sizes should be available.
- Specified system gaskets should be kept on site.

One note concerning gaskets; those which are frequently used in flushing, particularly in remote operations, are not as specified and can deteriorate, thus leading to additional debris loading.

- Spare filter cartridges must be available for insertion when the system is cleaned.
- Flexible, clean hoses that are oil resistant, of the proper pressure rating and have smooth interiors should be available for use as jumpers during initial flushing phases.
- Clean globe valves should be present to control velocities through each system supply line.

During the flushing operation, the use of mechanical vibration on piping to break debris loose at flange connections and/or any means of introducing turbulence into the pipe will significantly reduce flushing time. One specific means that has been found to significantly reduce flushing time has been the introduction of clean air or nitrogen into the system during flushing. This makes the system extremely turbulent and gives a scouring action that rapidly cleans all pipes. Care should be taken to only bubble downstream of the filters and only when the bearings are bypassed.

Filter cartridge assembly

Extreme care must be exercised when assembling filters to preclude the possibility of bypassing unfiltered oil. This action can significantly extend flushing time and result in the acceptance of a system that exposes the critical equipment to significant damage. After every filter cartridge change, the following checks should be made:

- Inspect multiple stacked cartridges to ensure inter-cartridge sealing.
- Confirm each bottom end seal is tight.
- Confirm top seals are tight and in place.
- Obtain an oil sample immediately downstream of the filter and ensure particle size is not excessive.

Determination of acceptability

The final check of flushing acceptability is performed by using screens, usually 100-mesh or finer depending on the system. It is important to be sure that screens are not incorporated into the system until the system is deemed to be reasonably clean. Initially 60-mesh or greater should be used. If this practice is not followed, screens can break, thereby causing long term problems with the system. Anytime that 100-mesh screens are used they should be backed by 60-mesh screens as a minimum to minimize the possibility of breakage.

The acceptance criteria should be based on a practical approach. Final acceptance should take into consideration that the drain side of most auxiliary systems is unpressurized and is vented to atmosphere. As a result, a small amount of debris can and will enter the system and will always be present. Once a consistent low debris load void of any hard, metallic particles is achieved, the drain side should be considered to be clean.

The supply side must be consistently free of all hard particles and debris. 100-mesh screens should be installed at *each* system final supply point to the equipment (note that all jumpers are removed at this point). A typical limit for soft (non-metallic) particles is 20 for a 2" schedule 40 pipe.

The acceptance criteria should be mutually agreed by the vendor and the user in advance of issuing the flushing procedure. Note that metallic particles are not acceptable in any location during final flushing acceptance. Since many systems today utilize stainless steel piping and components, the use of a magnet does not conclusively prove the absence of metallic particles since stainless steel is non-magnetic.

Once the system is accepted, *all* screens should be moved. It is recommended that the system be kept running until critical equipment startup. Shutting down the system at this point will only expose it to the possibility of additional debris loads from the environment.

Effective flushing – final comments

When one considers that this operation is the last significant pre-commissioning operation prior to the commissioning of new equipment, the tendency to rush and perform less than adequate flushing operations is surprising. It is strongly recommended that anticipated flushing times for each system are discussed frankly with management as early as possible in the project to ensure a clean auxiliary system. The importance of effective flushing is essential to the final reliability of critical equipment. Therefore, the flushing procedure and acceptance thereof should be number one agenda item in the field pre-commissioning operation. User-contract or concurrence of this procedure should be obtained as soon as possible when commencing filed construction. Once the procedure is accepted,

supervisory means should be established to accurately monitor and implement this procedure. A well-thought-out flushing procedure, supplemented by equipment specifics, and performed by a *dedicated crew* of experienced individuals, will result in accurate, effective flushing operations that are accomplished in approximately two to three weeks. Any misconception that a flushing operation can be accomplished in a shorter time is exactly that – a misconception. The effective flushing of equipment will go a long way towards long term, reliable operation of critical equipment.

Auxiliary system flushing procedure

Courtesy of M.E. Crane, Consultant

The following procedure is presented as a guide for field flushing of lube and seal systems. In order to be fully productive, it is recommended that all requirements noted herein be strictly followed.

1. General

1.1 Flushing operation will be carried out by the designated party (contractor or owner).

1.2 Cleanliness of oil console, equipment skid, overhead seal oil tanks, piping systems and screens shall be determined by mutual agreement between equipment vendor, contractor and owner.

1.3 Owner and vendor shall keep a log for general review of flushing progress. Master flow sheets shall be kept by owner and updated to progress. An entry shall be made during each shift.

1.4 In general the oil flush shall be performed using selected permanent auxiliary equipment which is part of the vendor supply package. This will include the following:

- Auxiliary oil pump (electrical) and main oil pump if possible.
- Main oil filter to be in position for all flushing.
- Main oil reservoir, degassing tank, overhead seal oil tanks and seal oil traps.
- Skid piping.
- Selected instruments and controls.

2. Preparation

2.1 Any residual oil from factory testing must be removed from the reservoir and filters. Relief valves must have been checked prior to flushing.

2.2 The reservoirs, degassing tanks and filter casing must be wiped clean, inspected and approved by the owner. All cleaning must be carried out using a lint free cloth. When filters are open for cleaning, special care should be taken to avoid contaminates falling into 'clean' side of filter housing.

2.3 The filters must be verified to be in place and satisfactory for flushing. Examination of filters will include checks for bypassing, inside or outside of filter housing.

2.4 All lube and seal oil interconnecting piping will be installed consistent with normal operating conditions in accordance with bypass piping arrangements as mutually agreed upon between vendor and owner.

- All equipment supply and drain piping is required to be flushed during entire flushing operation. Location of all valves, bypasses and screens shall be in accordance with marked-up P&IDs of lube oil, seal oil and control oil system.
- Add hand valves suitable to meet the operating requirements during flushing to all piping supply points.
- All lines to the steam turbine throttle valve, servo motors and dump devices will be flushed in accordance with a manufacturer's requirements.
- Overhead seal oil tanks will be flushed by jumpering to compressor reference gas lines between compressor, overhead tanks and drainers are required to be flushed.

2.5 Stainless steel 100-mesh screens with backup 60-mesh screens with a number of spares must be fabricated with retaining gaskets and installed at selected lube oil piping flanges. This fabrication involves cutting and fitting the screen to the gaskets. Screens must be clearly and permanently tagged for ease of identification. Location for screens must be agreed with by vendor and owner. Basically, they should be positioned at all inlets to the machine. Locations immediately after risers must be avoided. Additionally, 100-mesh screens, with 60-mesh backup screens, will be installed at the main oil return, degassing tank inlet and return and reservoir oil fill connection. These screens will be in place during all flushing operations. A drain should be fitted at lower point of return lines ahead of screen in order to deal with a blocked screen. Also, a pressure device (manometer – i.e. a simple length of plastic tubing) is to be installed at this point to monitor any pressure build-up due to blockage.

2.6 The reservoir shall be filled with the lube oil specified for permanent plant operation unless directed otherwise by the owner.

2.7 Lube oil flush should not occur until the compressor skid has been fully grouted.

2.8 The compressor rear bearing port cover and the load coupling guard must be installed in order to minimize any oil spill during flushing.

2.9 The vent piping must be checked out for mechanical completeness.

2.10 Check for correct operation of the auxiliary lube oil pump on/off/auto switch and the oil high temperature alarm before commencing flushing.

2.11 The instruments required for the flushing operation shall be identified on a P&ID mark-up for flushing and will be calibrated for normal operation prior to starting the oil system.

2.12 Add nitrogen bottles or instrument air connections at suitable tapping points downstream of lube and seal oil filters.

2.13 Water to oil coolers shall be provided.

2.14 An auxiliary boiler will be provided to heat the oil to approximately 180°F.

3. Flushing procedure

3.1 The range of temperatures for the hot oil circulation flush shall be 120°F to 180°F. Before initial circulation the oil should be heated to approximately 120°F.

3.2 The following parameters shall be documented in the log on an hourly basis:
- Pump discharge pressure
- Bearing header pressure
- Oil reservoir temperature
- Bearing header temperature
- Oil filter differential pressure
- Filter in use
- Sections of piping being flushed
- Start time

3.3 The drain oil sight flow gauges shall be continually monitored for flow at all places. The complete filling of the sight glass indicates a flow blockage. Immediate action shall be taken to stop circulation and clean filter screens. The debris obtained on the screen shall be collected into plastic bags, identified by screen location, machine number and time.

3.4 The following schedule shall be used for flushing:
- Add 100-mesh screen with 60-mesh backup screens at oil reservoir and degassing tank if applicable as stated in paragraph 2.5 and flush through total system at 15 minute intervals until screens are reasonably clean. Monitor closely the return lines on a continuous basis to ensure system is not backing up with oil.
- Flush through total system at intervals of one hour until screens are reasonably clean.
- Flush total system, alternating through systems section until screens are clean. Supply lines shall be alternated to ensure a greater than 150 percent oil flow is maintained at all times.
- Add 100-mesh screens with 60-mesh backup screens to lube and seal oil inlet lines to all equipment. Also add 100/60 mesh screens to overhead and seal oil tanks, reference lines and coupling guard feed lines (if furnished).
- Repeat 'flush total system' above.
- Alternate flushing through each filter/cooler section, control valves and their bypasses, overhead tanks, seal oil traps and reference gas lines. *Record.* Note: A differential pressure of 15 PSIG across either filter indicates the need for filter change.
- Bubble nitrogen through system at regular intervals. *Record.*
- Flush through all instrument connections. *Record.*
- Flush through all pressure control valve impulse lines. *Record.*
- Thermoshock the system by use of the lube oil coolers at regular intervals (varying oil temperature between 120°F – 180°F). *Record.*
- Rap exposed piping with a fibre hammer at one hour intervals. *Record.*

3.5 When the 100-mesh screens meet criteria in paragraph 3.9/3.10 reinstate all instrumentation, orifices, pipe spools, etc. Arrange entire system in normal, complete configuration with all controls, alarms, etc., in operation.

3.6 Add 100-mesh white cloth (backed with 60-mesh screen) to all lube and seal oil supply points on equipment bearings, seals and control oil inlets. Flush until the criteria as stated in paragraph 3.9/3.10 is achieved, but for a minimum period of 24 hours.

Note: During final flush, alternate flushing through each filter/cooler section, control valves and their bypasses.

3.7 Whenever practical, circulation shall be continued from the time of startup until completion on a 24 hour per day basis. Any irregularities shall be immediately reported to the vendor representatives and the applicable owner representative as designated, which will be posted on the accessory skid control panel.

3.8 Owner's quality control representative will monitor all operations for compliance and verify all records, test parameters and acceptance criteria on a surveillance basis. Final screen particle count will be verified and recorded by quality control.

3.9 The oil system acceptance criteria which shall be the basis for witness approval parameters for contractor and/or owner shall be as follows:

Screen contamination shall be within the particle count limits according to the size of pipe. It will be determined as follows:

20 non-metallic particles on pipe 1" to 2"
50 non-metallic particles on pipe 2" to 4".

3.10 Particles shall not be metallic and shall display random distribution on the screen.

3.11 The acceptance of the system shall be after installation/inspection 100-mesh cloth covered screens, then circulating the lube oil an additional four hours and re-inspecting the screens. The final four hours' flush should be with valves open allowing full flow through the system at operating temperature. Final acceptance requires an oil analysis to determine metal content, viscosity and water content.

3.12 Following acceptance the system shall be restored by the owner for normal operation including the following actions:
- Remove all screens.
- Visually inspect overhead seal oil tanks, degassing tanks, drainer modules, if required, wipe clean with lint-free cloth.
- Replace all components. Clean filter cartridges (less than 5 PSI pressure drop) can remain subject to owner approval. If new cartridges are fitted, the cleanliness of the system must be rechecked.
- Restore reservoir oil to normal operating level.
- Obtain oil analysis.

System identification____

Oil type_____________ Req'd circulation temp________________

Minimum duration req'd_____________________________________

Pre-flush checklist

	Owner	Vendor
1. System configuration per reference drawings.	___________	___________
2. Old oil has been removed and reservoir/filter housing wiped clean.	___________	___________
3. Filters and screens installed for flushing.	___________	___________
4. All required instrumentation calibrated.	___________	___________
5. Lube/seal oil filled to capacity.	___________	___________
6. Bearing housings ready to accept flow.	___________	___________
7. Initial oil temperature reached.	___________	___________
8. Approval to initiate oil flush.	___________	___________

Oil flush acceptance

The oil flush of the subject system has been conducted in accordance with site procedures and accepted.
Date of flush _______________Time started________________
Time completed_______________
Inspected and accepted by _____________ Date___________
Owner ________________
Vendor representative ____________

Best Practice 7.19

Monitor oil filter change time (when the filter differential pressure alarm is activated) to clean oil tank and rundown tanks.

All gravity drain oil systems accumulate debris (oil sludge, etc.) in the oil reservoir after a certain time.

Increasing frequency of oil filter change (significant increase – say from once per year to once every six months) indicates a need to clean the oil reservoir and rundown tanks at the next turnaround.

A preventive maintenance (time based) approach to cleaning oil reservoirs exposes the plant to reliability issues if not done properly.

This best practice is a predictive maintenance approach that will minimize oil reservoir and overhead tank cleaning cycles whilst still ensuring machinery reliability.

Lessons Learned

Failure to monitor and clean oil reservoirs and rundown tanks has caused many machinery component failures.

Many machinery component failures have had a root cause in failure to clean oil reservoirs and rundown tanks.

Particular attention is drawn to overhead seal oil tanks that have never been cleaned, but are exposed to the process gas and associated process debris (seal gas reference line located on top of the seal oil tank without the use of a separation bladder). Every time the unit is shut down, any oil contained in the tank and all associated debris enters the seal without the benefit of filtration.

Benchmarks

This best practice has been used since the early 1990s, when bushing seal failures were experienced because the overhead seal oil tanks had never been cleaned. The implementation of this best practice has optimized critical unit reliability and minimized tank cleaning previously performed on a preventive maintenance (time based) basis.

B.P. 7.19. Supporting Material

Please refer to material in B.P: 7.18.

Best Practice 7.20

Continuously vent the non-operating cooler and filter by installing orificed return lines, and keep the transfer valve bypass fill line opened to ensure warm-up of the cooler and filter prior to transfer, in cold climates, to prevent a unit shutdown on low oil pressure.

In cold climates (ambient temperatures below 15°C at any time of the year), cool, static oil in the non-operating cooler and filter will cause a transient pressure drop when it is changed on-line. This action can cause critical equipment unit trips on low oil pressure when the auxiliary pump did not start, or did not start in time.

Requiring orificed vents on all coolers and filters, and keeping the non-operating vent lines open as well as the cooler and filter fill lines, will ensure that non-operating coolers and filters are always maintained at the same temperature as the operating vessels, and are vented and ready for a successful transfer.

Where a trip has been caused by the issue noted above, operating procedures should be revised and orificed vents installed if required.

Lessons Learned

Unit trips experienced during cooler and/or filter transfers are usually blamed on a 'dead spot' in the transfer valve when in fact, cool static oil of high viscosity in the non-operating cooler and/or filter caused the unit to trip on low oil pressure.

It should be noted that all types of transfer valves (six-way, ball valve design, etc.) do not have a 'dead spot' that can cause an oil system pressure drop when transferred.

Benchmarks

This best practice has been used since the mid-1990s, when unit trips were experienced that were caused by a cooler and filter transfer to cold, static oil. Since that time, all projects, with ambient temperatures below 15°C at any time of the year, have been required to have cooler and filter vents, and P&IDs indicating that valves were in open condition. All field auxiliary system audits also recommended this approach when ambient temperatures could fall below 15°C.

B.P. 7.20. Supporting Material

Transfer valves

The function of the transfer valves in the auxiliary system are to allow transfer from one bank of components (coolers, filters, etc.) to the stand-by bank of components without significant pressure pulsations being introduced into the system. In addition, transfer valves must be designed to positively shut off the unused components to allow for maintenance while the system is in operation.

Types of transfer valves vary widely. Figure 7.20.1 shows a six-port transfer valve that allows flow into the valve to be diverted either to the left bank or right bank of components as shown in the standard schematic.

Other types of transfer valves include the standard, globe-type valve shown in Figure 7.20.2.

Both types exhibit the characteristic of minimal pressure change when transferring from one bank to the other. The six-ported valve should be designed with a lifting jack when a taper plug is used. This feature allows transfer of valves from one bank to another easily and still ensures tight shutoff of the valve when in its proper position.

Selection of the valves should include material and sizing considerations. Materials of construction should be carbon steel as a minimum with stainless steel internals. The use of bronze components should not be considered in systems that can incorporate gas entrained in the system fluid. In such systems brass, copper or bronze is not permitted. Valves are sized to match piping sizes in the system, which is normally designed for fluid velocities of four to six feet per second.

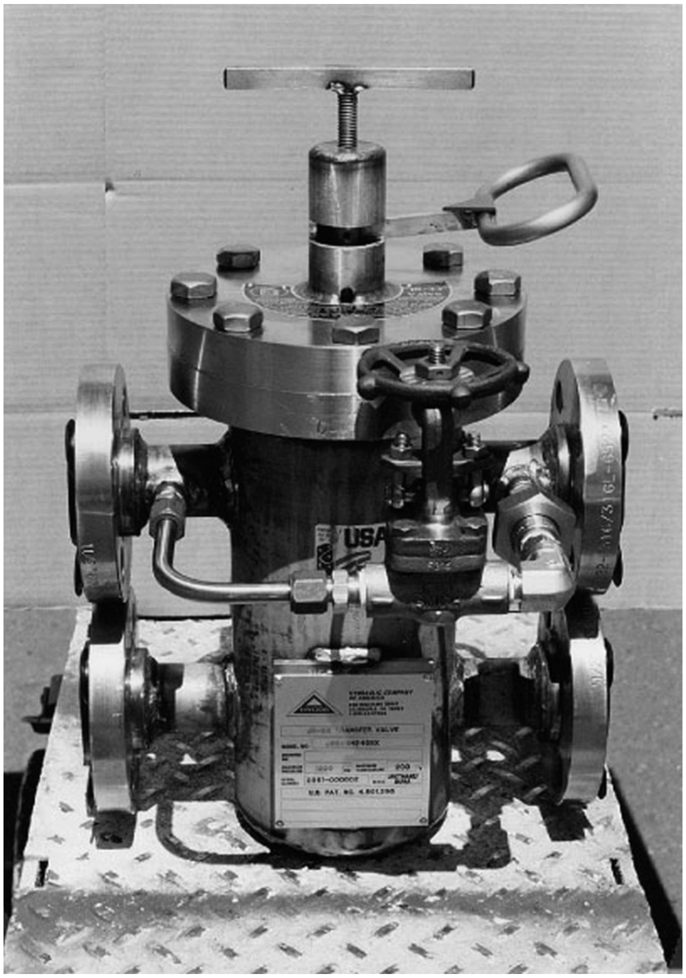


Fig 7.20.1 • 6 port transfer valve (Courtesy of Hycoa)

Reliability considerations

As previously mentioned, transfer valves including lifting jacks are susceptible to valve plug damage. Frequently, operators do

not employ lifting jacks when transferring valves. This type of valve can become bound and even break at the stem, necessitating critical equipment shutdown. Prior to transferring flow from one bank to another, the bank to be transferred to must be full of auxiliary system fluid and properly vented. If it is not, the capacity of coolers, filters and piping in the bank to be transferred to act as a vessel, and consequently reduce flow and pressure to the critical equipment, which will shut down the unit. Also, improperly vented components containing air or gases can cause control valve instabilities which can also lead to equipment shutdown.

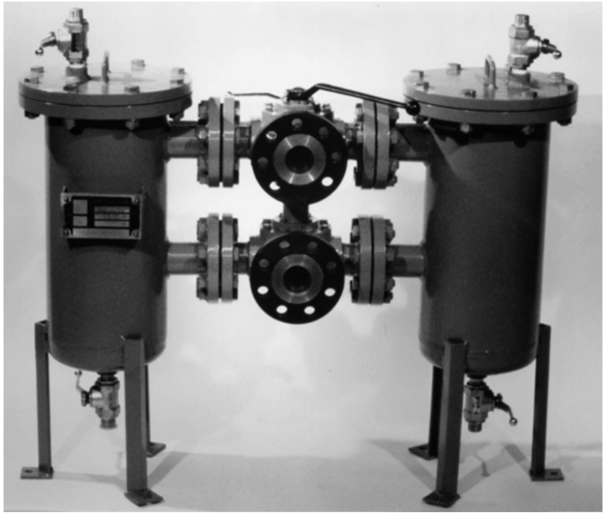


Fig 7.20.2 • 6 way (double, 3 way, flexibly coupled) (Courtesy of Hycoa)

Best Practice 7.21

Arrange critical machinery protection systems to have only one trip signal for each component to prevent spurious trips.

Most new PLC/DCS based protection systems installed today (2010) incorporate alarm and trip signals for each instrument.

Machine reliability is a function of simplicity.

Over instrumented trip circuits expose a critical machine unit to spurious trips.

FAI has followed the best practice of minimizing trips while totally protecting the machinery unit and all associated support systems by using only one trip signal per component or system.

A typical protection scheme for a steam turbine driven compressor train with dry gas seals and oil lubrication is as follows:

- Radial vibration trip – each radial bearing
- Axial displacement trip – each thrust bearing
- Dry gas seal primary vent trip – each seal
- Low lube oil pressure trip – train trip
- Low control oil pressure trip (integral in trip valve)
- High compressor discharge temperature trip (if applicable)
- Overspeed trip
- High suction vessel level trip

Lessons Learned

Excessive unit trips lower the reliability of critical machines and expose the plant to prolonged downtime and corresponding loss of revenue.

Benchmarks

This best practice has been used since the late 1990s, when two clients were starting a new plant that was essentially a duplicate of their past plants. Neither client could get their respective units commissioned and started on schedule and requested our help. In both cases, the root cause of the problems was ‘over-instrumentation of trip circuits’. Existing (“old”) successful plant trip logic was copied, excessive trips deleted and both plants achieved smooth trouble free start-up and optimum machine reliabilities.

Best Practice 7.22

PM control valves and pulsation suppression devices at every turnaround by changing diaphragms, checking valve stem freedom of movement and cleaning pulsation suppression devices to ensure optimum control valve reliability.

A friction bound control valve (tight and/or oil sludge in packing) can cause a unit trip during a main oil pump trip if sufficient system oil (because the valve does not immediately close) is not available during the transient event to prevent a low oil pressure unit trip.

A failed control valve diaphragm, which can be replaced on-line, exposes the plant to shutdown, since improper adjustment of the manual bypass valve around the control valve and/or taking out or putting the affected control valve back into service can cause a low oil pressure trip.

A plugged control valve sensing line pulsation suppression device can cause an oil actuated control valve to go unstable (hunting), which can also cause a unit low pressure shutdown.

Preventive maintenance (PM) of the above items during a turnaround is good insurance for a trouble-free run of the highest reliability. Note that most plants today (2010) are targeting six year uninterrupted runs of critical equipment.

Lessons Learned

Plants frequently experience unscheduled unit trips caused by control valve issues. It is good practice to PM the control valve systems during a turnaround.

Benchmarks

This best practice has been used since the mid-1980s, when I served as an interim reliability manager at a large petrochemical plant. It has been recommended to all our clients since that time, and has resulted in trouble free control valve operation during steady state and transient conditions.

B.P. 7.22. Supporting Material

System reliability considerations

A number of reliability considerations are worthy of mention concerning auxiliary system control and instrumentation.

Control valve instability

Control valve instability can be the result of many factors, such as improper valve sizing, improper valve actuators, air in hydraulic lines or water in pneumatic lines. Control valve sensing lines should always be supplied with bleeders to ensure that no liquid is present in pneumatic lines or air in hydraulic lines. The presence of these fluids will usually cause instability in the system. Control valve hunting is usually a result of improper controller setting on systems with pneumatic actuators. Please consult instruction books to ensure that proper settings are maintained. Direct-acting control valves frequently exhibit instabilities (hunting on transient system changes). If checks for air prove inconclusive, it is recommended that a snubber device (mentioned previously) be incorporated in the system to prevent instabilities. Some manufacturers install orifices which sufficiently dampen the system. If systems suddenly act up where problems previously did not exist, any snubber device or orifice installed in the sensor line should be checked immediately for plugging.

Excessive valve stem friction

Control valves should be stroked as frequently as possible, to ensure minimum valve stem friction. Excessive valve stem friction can cause control valve instabilities or unit trips.

Control valve excessive noise or unit trips

Squealing noises suddenly produced from control valves may indicate valve operation at low travel (C_v) conditions. Valves installed in bypass functions that exhibit this characteristic may be signaling excessive flow to the unit. Remember the concept of control valves being crude flow meters. Periodic observation of valve travel during operation of the unit will indicate any significant flow changes.

Control valve sensing lines

Frequently, plugged or closed control valve sensing lines can be a root cause of auxiliary system problems. If a sensing line that is dead ended is plugged or closed at its source, a bypass valve will not respond to system flow changes and could cause a unit shutdown. Conversely, if a valve sensing line has a bleed orifice back to the reservoir (to ensure proper oil viscosity in low temperature regions), plugging or closing the supply line will cause a bypass valve to fully close rendering it inoperable and may force open the relief valve in a positive displacement pump system.

Valve actuator failure modes

Auxiliary system control valve failure modes should be designed to prevent critical equipment shutdown in case of actuator failure. Operators should observe valve stem travel and pressure gauges to confirm valve actuator condition. In the event of actuator failure, the control valve should be designed for isolation and bypass while on line.

This design will permit valve or actuator change-out without shutting down the critical equipment. During control valve on-line maintenance, an operator should be constantly present to monitor and modulate the control valve manual bypass as required.

Best Practice 7.23

Monitor oil flash point in combined lube/seal systems to determine if the seal oil drainers and/or the atmospheric seal bushing is passing gas which will expose the plant to a safety hazard.

Oil flash points below approximately 150°C (325°F) in the oil reservoir indicate gas entrainment in the oil.

Combined lube/seal oil systems have the following possible paths of gas entry into the oil reservoir:

- Failure of the oil drainers to completely close, which allows oil and gas to enter the reservoir.
- Blocked degassing tank vents prevents the venting of the seal oil entrained gas mixture.
- Passing of gas through the atmospheric bushing due to seal oil internal compressor port blockage resulting in zero seal oil to gas differential at the atmospheric bushing seal face.

Monitoring the flash point along with all other parameters during periodic oil analysis will alert personnel to possible seal system component malfunctions before a safety hazard occurs.

Lessons Learned

There have been plant explosions and fires that have resulted from excessive gas in an oil reservoir that was not properly monitored.

Recent experience (2008) found that all four (4) of the bushing seals in a two-body compressor train were passing gas to the oil reservoir through the bearing oil return header and to the compressor deck. Indications of low viscosity in the oil sample led to taking flash point sampling and gas sniffing. The root cause was failure to ever clean the seal oil overhead tanks. Debris from the reference gas lines (over 20+ years of operation) had entered all compressor seal oil internal supply ports, which caused them to plug and render approximately zero oil to gas differential at the atmospheric bushing seal face, even though the external differential pressure gauges registered acceptable differential pressures.

Benchmarks

This best practice has been used since the late 1980s, when a flash point sample taken after a glass oil sample bottle broke indicated that the flash point was 35°C (95°F) in a combined lube/seal oil console reservoir. Since that time, it has been our best practice to always recommend that flash points be taken in all oil samples in combined lube oil/seal oil consoles.

B.P. 7.23. Supporting Material

The return side of a seal oil system consists of three distinct return lines:

- The atmospheric drain back to the seal oil reservoir.
- The pressurized drain back to the seal oil control valves (if installed).
- The contaminated seal oil drain.

All liquid seals are designed such that a small amount of oil leakage (38–76 liters, 10–20 gallons per day per seal) enters the drain between the inboard gas side seal and the compressor internals. This drain is commonly known as the contaminated oil drain. To ensure that gas does not leak into the seal, a small amount of liquid is designed to continuously flow through the seal and into the drainer. Depending on the nature of the process gas, the oil may or may not be recirculated to the degassing tank and the seal oil reservoir. The function of the contaminated oil drain system is therefore to collect all contaminated (gas entrained) oil and direct this oil to the desired location. This may be directly to the seal oil reservoir, to a degassing tank or other oil reclamation device, or to a contaminated oil drain. We will now examine basic contaminated oil drain system configurations and discuss the function of each component.

Basic system configuration

Refer to Figure 7.23.1 which shows a basic contaminated oil drain system. The system consists of:

- The contaminated oil drain line to a seal oil drainer for each seal in the unit.
- The drainer.
- The vent connection or reference connection from the top of the drainer.
- The drainer level control device (internal or external) which automatically drains upon reaching a preset level.
- The degassing tank (optional).
- The oil reclamation device (optional).

The following system design alternatives are available depending upon the operating conditions of the compressor and the process gas.

Sweet hydrocarbon or inert gas service

For sweet or inert gas service, the seal oil drain can be returned directly to the reservoir, provided the drainers are sized for adequate residence time and the seal oil leakage is reasonable (less than one gallon per hour per seal). A sweet hydrocarbon gas is defined as a gas that does not contain hydrogen sulfide (H_2S). The vent line on top of the drainer can be routed to a lower pressure source, atmosphere or back to the compressor suction. If routed back to the compressor suction, a demister should be installed to prevent oil from entering the compressor case. The sizing of the orifice in the vent line of each drainer is critical in that it ensures that all contaminated oil flow will enter the drainer. Too low a velocity will allow contaminated oil to enter the compressor. Too high a velocity could cause oil to enter the compressor via the vent or reference line. Typical velocities in this line should be 4.5–6 m/sec (15–20 ft/sec).

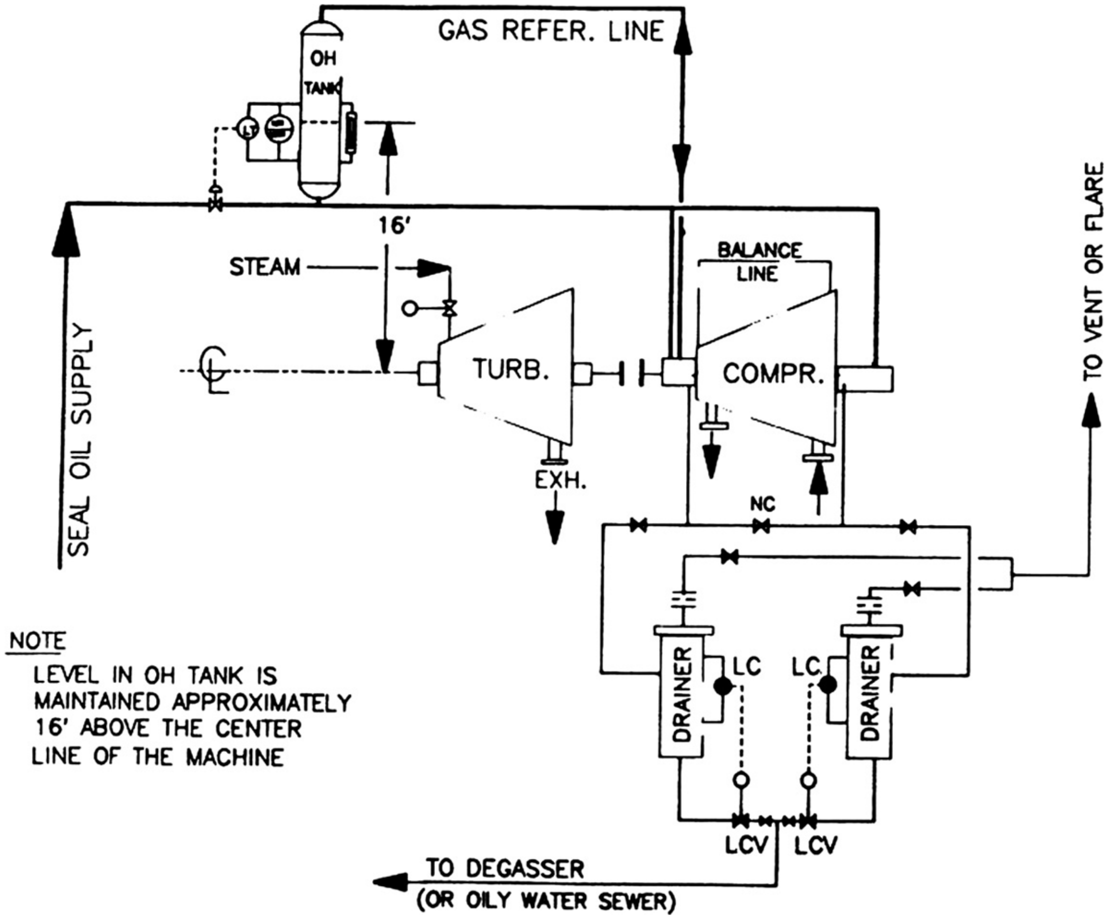


Fig 7.23.1 • Schematic – seal oil arrangement (Courtesy of M.E. Crane, Consultant)

High suction pressure

In high suction pressure applications, the system configuration changes somewhat as shown in Figure 7.23.2.

In this application, an external level control valve is recommended on each contaminated seal oil drainer. In addition, venting drainers to atmosphere or to flare will result in exceptionally high pressure drop across the orifice. In this application, the vent line is usually routed back to the compressor suction. Again, an adequately sized demister must be installed to prevent oil ingestion in the compressor.

Vacuum service

Compressors that act with suction pressure below atmospheric pressure, or use a suction throttle valve, can have a reference pressure less than atmospheric. In this case, the contaminated seal oil drainer vent line must be referenced back to the compressor suction, in order to ensure proper operation and not allow air to enter the demister through the vent line in the reverse direction. For this application, a buffer gas system must be installed that will be designed to maintain the drainer pressure above atmospheric pressure at all times in order to allow the drainer to be drained.

Sour process gas

Where the process gas can contaminate the seal oil leakage (as in the case of H_2S gas, etc.), the contaminated seal oil drain line is usually routed to the plant contaminated oil system and not back to a degassing tank or the reservoir.

Oil drain system component design

The function, sizing criteria and operating specifics of each major component of the contaminated oil drain system will now be discussed.

The contaminated seal oil drainer

The function of the contaminated seal oil drainer, as shown in Figure 7.23.3, is to contain all of the oil leakage from a specific seal.

Normally, drainers are automatic; that is, they drain oil when a specific preset level in the drainer is reached. The typical sour oil leakage per day varies from less than five gallons to an excess of 20 gallons (75 liters) per day. Normal vessel capacity of liquid is approximately 1–2 liters ($^1/_4$–$^1/_2$ gallon). Consequently, drainers must either be manually drained or automatically drained between 40–80 times per day. The inlet connection to the contaminated seal oil drainer is from the gas side seal oil drain and contains an oil-gas mixture. The gas is either the process gas or a clean buffer gas as will be discussed in following chapters.

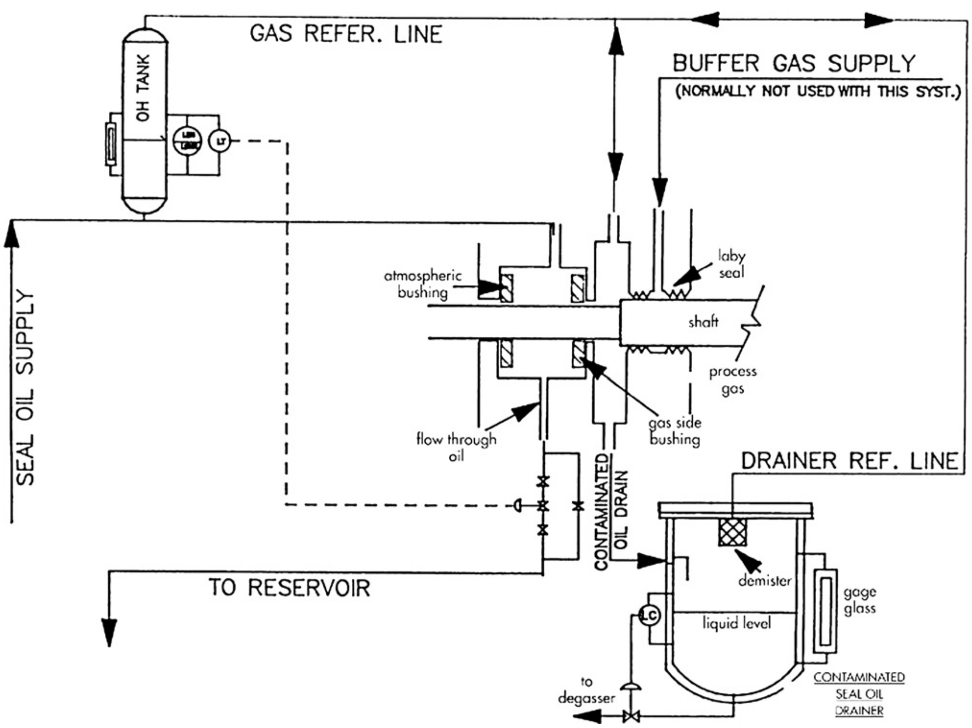


Fig 7.23.2 • Contaminated oil drain system (with referenced drainer) (Courtesy of M.E. Crane, Consultant)

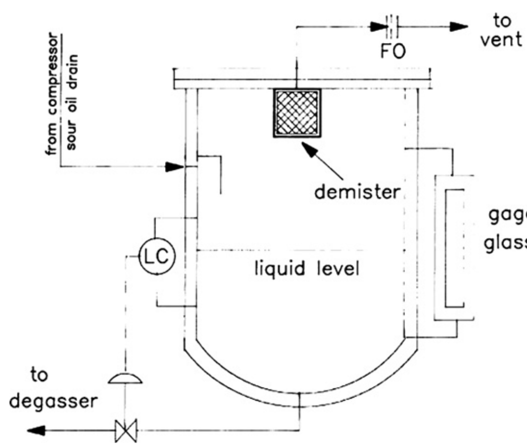


Fig 7.23.3 • Contaminated seal oil drainer (Courtesy of M.E. Crane, Consultant)

The top of the drainer contains a vent connection which may be routed to a reference connection in the compressor case or directly to vent. Regardless of the configurations, the drainer contains a mixture of free gas above the liquid and entrained gas in the liquid. Drainers are essentially of two designs.

Internal valve

This design incorporates a ball float valve shown in Figure 7.23.4 which is internal to the drainer and opens at a prescribed level.

In order to control the rate of drainage from the drain pot with the valve open, an orifice is installed. Note that this orifice is sized to allow controlled drainage time under normal operating conditions. Conceptually, this orifice represents another atmospheric bushing in the seal oil system. It is similar to the atmospheric seal, in that the differential across the orifice will vary with reference pressure in the compressor. Often, care is not given to adequately sizing this orifice. As a result, during start-up conditions with low compressor case suction pressure, there is insufficient pressure drop across this orifice to adequately drain the vessel. The vessel will not drain until there is sufficient pressure drop. That is, there must be a column of liquid high enough (head) to drain through this orifice. Considering the installation location of seal oil drainers, often the height available from the center line of the compressor down to the drainer is insufficient, at low or zero pressure conditions, to force a drain. Therefore, under these conditions, all contaminated seal oil will drain into the compressor case. As can be seen from Figure 7.23.5, drainer installations should be provided with bypass valves around the drainer. It may be necessary to open bypass valves during low pressure operation to allow drainage back to the reservoir. *Caution must be exercised during this operation if compressor gas is toxic or flammable since a full stream of gas will be exiting continuously back to the appropriate vessel.*

Caution: Prior to opening any system connections or sampling fluids, obtain a site work permit to ensure the area and conditions are safe for work or entry.

A properly sized orifice for low pressure conditions must be installed in this line to minimize loss of gas.

Contaminated seal oil drainer with external valves

This application is usually used in higher pressure situations, such as reinjection compressors where suction pressure and seal oil drainer pressure run between 7,000 and 10,000 kPa (1,000 and 1,500 lbs psi). However, in dirty gas applications, where clean buffer gas is not available, it is recommended that this type

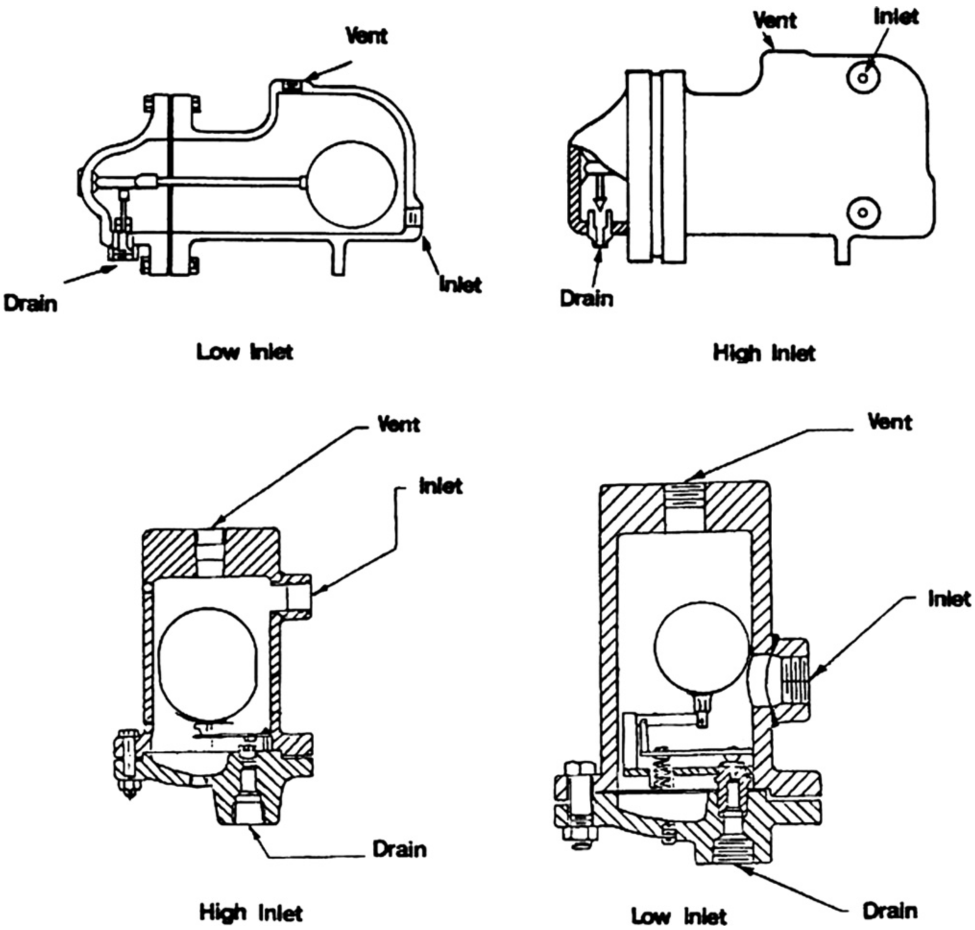


Fig 7.23.4 • Ball float valve. Above: WKM drainers; Below: Armstrong drainers (Courtesy of Elliott Co.)

of drainer be utilized to prevent malfunction of the drainer internals as shown in Figure 7.23.4.

The external valve is controlled by a level control transmitter which sends a signal to open the valve when the level reaches a specified amount, and quickly closes the valve when the level falls below a specified set point. Again, care must be taken in sizing the valve, to ensure that sufficient valve area is available under low suction pressure conditions to allow drainage back to the appropriate vessel. In addition, downstream piping must be adequately sized and designed to allow contaminated seal oil to exit the drainer.

Drainer reliability considerations – As can be seen, the reliable operation of the drainer valve, internal or external, is essential to safety and reliability. The valve must open and close tightly upon signal to ensure a minimum amount of processed gas exits the drainer. Many applications process a gas that tends to be sticky or has a high amount of carbon that will cause internal valves to bind, thus keeping the valve open at all times.

Attention must be paid to the gauge glass on the drainer. Gauge glasses must be kept clean so that levels can be observed. If level is not present, the drainer exit should be checked to ensure that gas is not exiting the drainer. If this is the case, the drainer should be isolated and inspected.

Caution: any action involving opening valves and drainers requires an area safety permit.

Systems should be designed such that drainers can be isolated during operation and one drainer can temporarily service two seals so that maintenance can be performed on a drainer while the unit is in operation. If a continued problem is experienced with clogging of drainer orifices or hanging up of internal valves, consideration should be given to injecting a clean buffer gas which will ensure satisfactory operation of the drainer. Note that external valves are also subject to this malfunction in that debris can enter the valve causing it to remain open.

Another reliability consideration is the drainer bypass line. Often this line is inadvertently left open, either after start-up or opened by operators during operation. This valve should be closed during operation, otherwise a continuous stream of gas exits and proceeds downstream.

If the drainer gauge glass is continually full, this could indicate that the drain valve is not properly opening, and contaminated oil could be forced into the compressor under this condition. Many processes prohibit the entrance of oil into the compressor and this action could cause shutdown of the unit. Again, the drainer should be isolated and inspected. Failure to properly monitor and maintain drainers leads to many unscheduled shutdowns of critical equipment.

One instance of false level gauge indication is shown in Figure 7.23.6. In this case, the top reference line of the level gauge is referenced to the drainer vent line. In this application, the vent line was referenced to the suction vessel and the

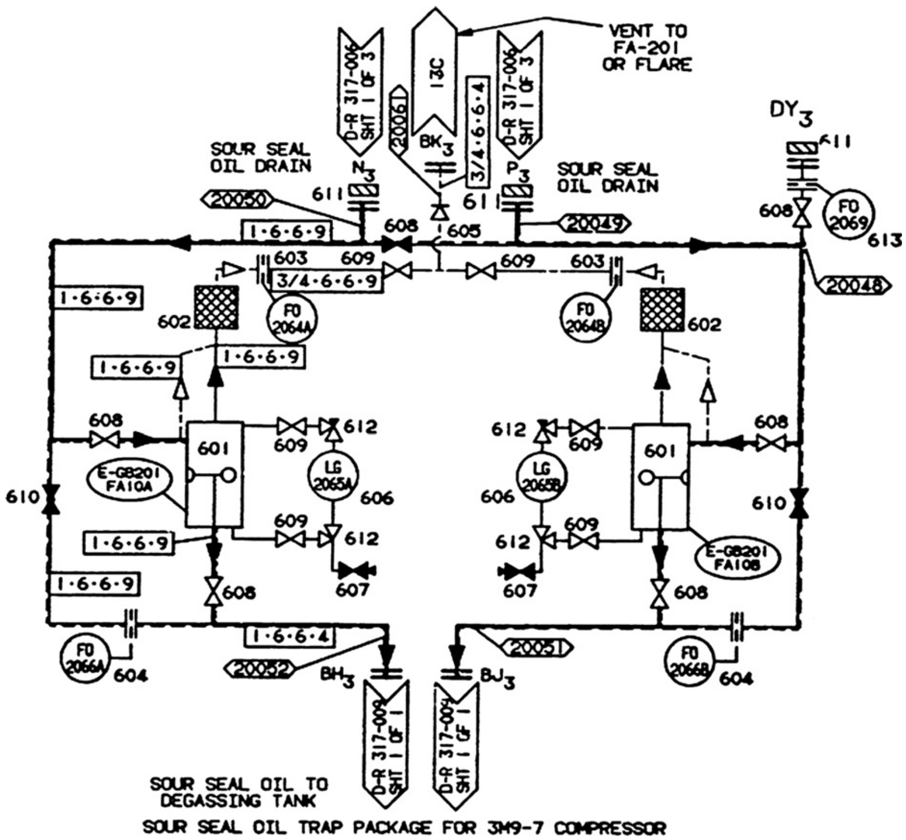


Fig 7.23.5 • Sour seal oil trap arrangement (Courtesy of Dresser-Rand)

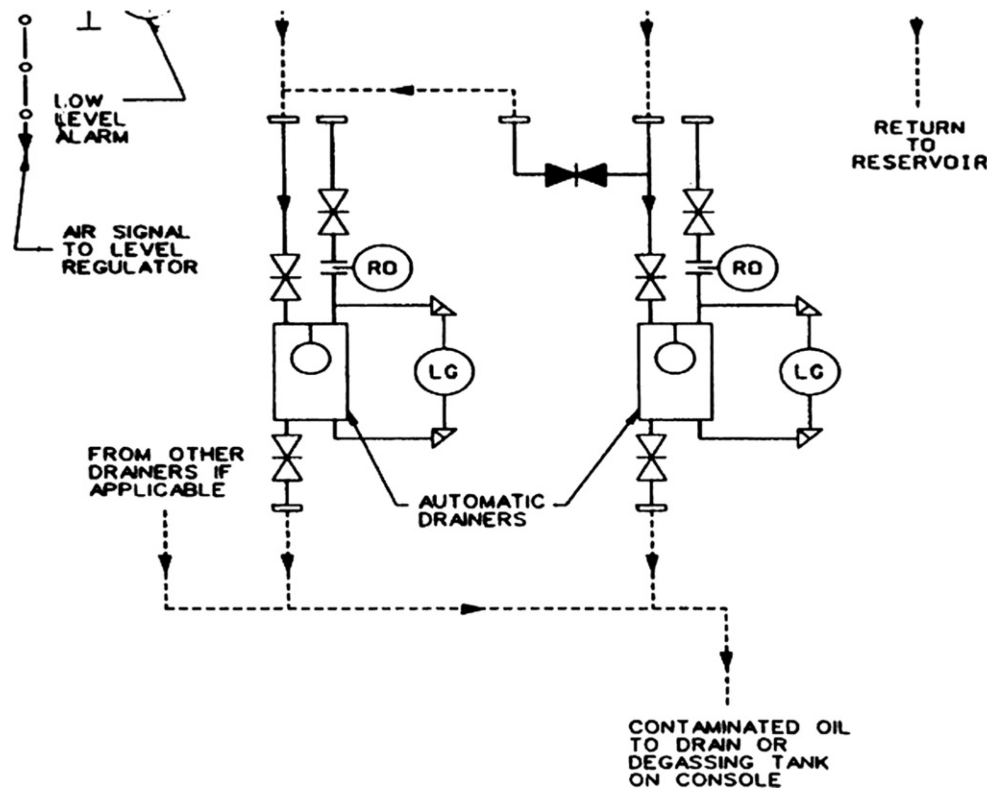


Fig 7.23.6 • False level gauge indication (Courtesy of Elliott Co.)

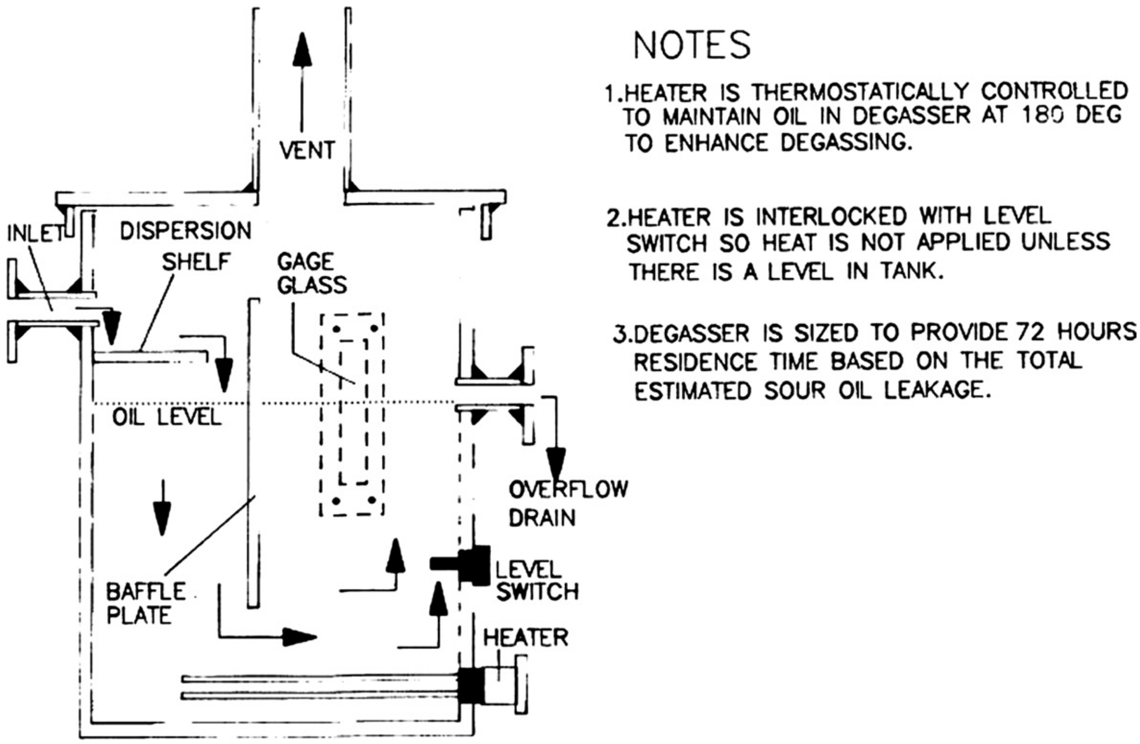


Fig 7.23.7 • Typical degasser arrangement (Courtesy of M.E. Crane, Consultant)

velocity through the orifice was big enough to create a vacuum on the gas in between the top reference line take off and the fluid in the level gauge. The result was that the liquid level was actually forced to the top of the glass (by greater pressure in the drainer), creating a false indication of a full oil drainer. The solution was to install a larger (four inch diameter) section at the drainer vent connection to minimize the velocity in this area. The high velocity steam acted as an eductor creating a lower pressure on the top of the liquid in the level gauge.

Characteristic		ISO-VG32 Light	ISO VG-46 Medium
Gravity, api		31.7	30.6
Pour, °F (°C)		20 (–7)	20 (–7)
Flash, °F (°C)		395 (201)	400 (204)
Viscosity	Sus at 100°F	150/165	215/240
	Sus at 210°F	44	48.8
	Csi at 40°C	28.8/32.0	41.4/46.0
	Csi at 100°C	5.2	6.5
VI, min		95	95
Iso viscosity grade		32	46
Color, astm. max.		1.5	2.0
Neutralization number, max.		0.20	0.25
Rust test (A&B) (astm D665–IP135)		Pass	Pass
Demulsibility (astm 1401) 3 ml max. at 130°F (54°C) ½ hr. at 180°F (82°C) 1 hr		Pass	Pass

Fig 7.23.8 • Typical oil flash points for seal oils

As an exercise, let's calculate the reduction in gas pressure on top of the oil in this application that would allow the oil level to rise six inches (the height of the level glass). Solving for pressure:

$$P = \frac{HD \times S.G.}{.102}\left(\frac{HD \times S.G.}{2.311}\right)$$

$$= \frac{.152(m) \times .85}{.102}\left(\frac{.5(ft.) \times .85}{2.311}\right)$$

$$= 1.27\ \text{kPa}\ (0.184\ \text{psi})$$

Therefore, if the high velocity gas stream can reduce the pressure in the trapped volume between the gas stream and the top of the liquid level by 0.184 psi, the level will rise to the top of the gauge glass and give the illusion of a full drainer. To see if this problem exists, on-line, briefly close the inlet valve to the drainer. If the level suddenly decreases, this would indicate this type of problem.

Caution: immediately open drainer inlet line to avoid seal oil entering the compressor.

Demisters

When the vent line from the top of the drainers is referenced back to any section of a compressor, a demister must be installed to minimize the amount of oil entering the compressor. The functions of demisters, therefore, are to eliminate oil migration into the compressor via the vent line. The demisters must be properly sized to ensure their efficiency. Care must be given to calculating the velocity through the demister which is dependent on the vent orifice. Maximum differential pressure conditions across the orifice must be considered. Frequently, mesh demisters are designed to be integral with the contaminated seal oil drainer as shown in Figure 7.23.3. In this case, any

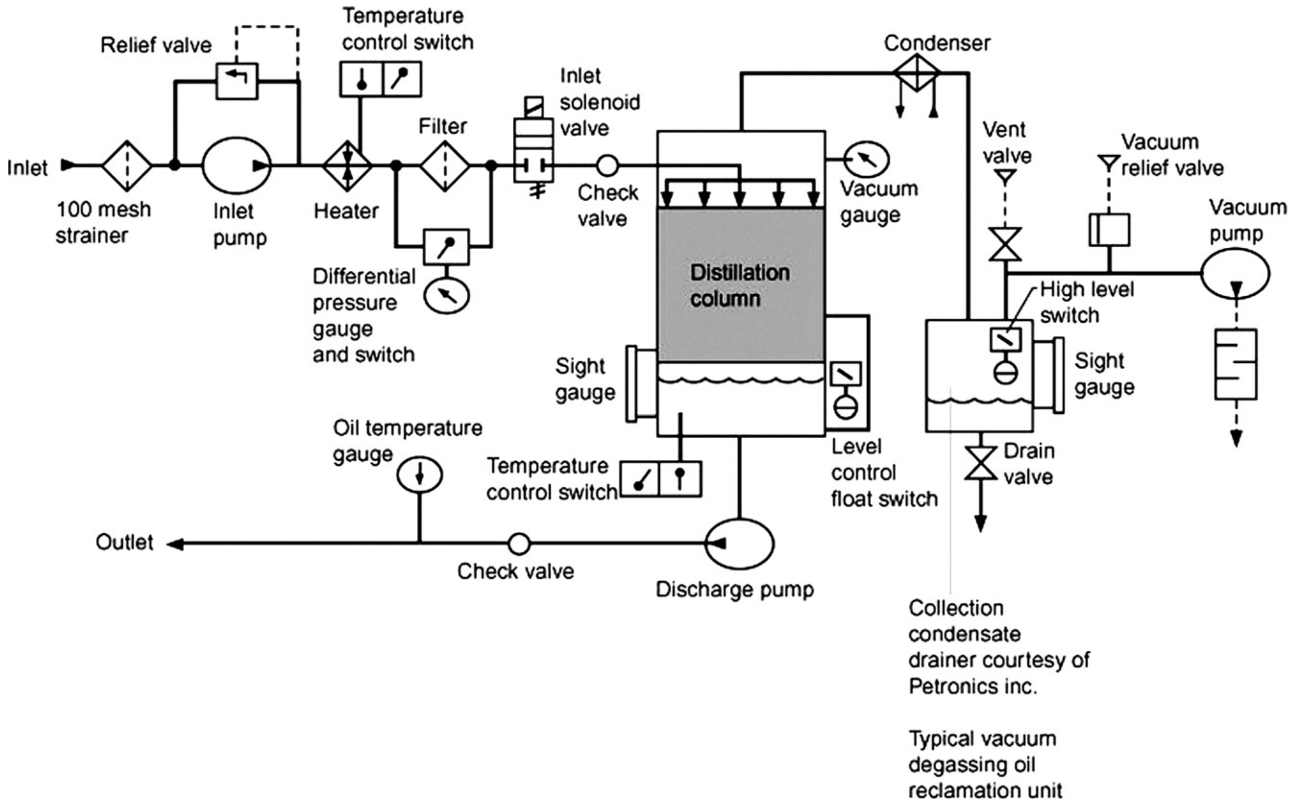


Fig 7.23.9 • Oil reclamation unit schematic (Courtesy of Petronics, Inc.)

oil mist exiting the top of the drainer will condense, and will fall back into the drainer. It must be understood that demisters are not 100% efficient. If the process cannot tolerate any seal oil, the vent line should be routed directly to flare or to atmosphere, depending upon the gas composition.

Caution: any toxic or flammable gas must be routed to a safe location.

Degassing tanks

As previously mentioned, the oil that enters the contaminated seal oil drainer is accompanied by free gas which exits the drainer through the vent, and entrained gas in the oil. The oil must be properly degassed prior to entrance back into the seal oil reservoir. The function of a degassing tank, therefore, is to degas the contaminated seal oil so that all oil exiting the degasser is within the original oil specification. A typical degassing tank is shown in Figure 7.23.7. The tank contains baffles, a heating device, and an overflow drain with a properly sized vent in order to degas all seal oil entering this vessel. Experience has shown that a degasser sized for 72 hours' residence time, based on the total estimated sour oil leakage, is usually sufficient to achieve the design objective.

As an example, if a compressor containing two seals each has a maximum specified leakage of 77 liters (20 gallons) per seal per day, the degassing tank should be sized as follows:

$$\begin{aligned}\text{Capacity} &= \text{total leakage per day} \times 3 \text{ days}\\ &= 463 \text{ liters (120 gallons)}\end{aligned}$$

A cursory inspection of any refinery or chemical plant will show that most degassing tanks are undersized. As a result, seal oil sampling usually shows a deterioration of oil viscosity and flash point. As mentioned in the previous chapter, flash point is the temperature at which the oil will sustain combustion. Light gasses (hydrogen, and hydrogen mixtures), significantly reduce oil flash points. Experience has shown that this value can approach the operating temperature of the system! It is strongly recommended that the following action should be taken when seal oil reservoir samples indicate a low flash point.

- Temporarily isolate seal oil drainer return, collect all seal oil and vacuum degas the seal oil. Provide make up fresh seal oil as required.
- Adequately size a degassing tank and install at earliest opportunity.
- Consider the installation of an oil reclamation device.

Figure 7.23.8 is a table of mineral oils used for seal oil service.

Please note the values of the oil flash points, and remember that many operating seal oil systems contain oil flash points that are of the order of 49°C (120°F). This is particularly dangerous in the case of combined lube and seal oil systems where the oil in the reservoir will actually enter the bearing system.

Oil reclamation units

In cases where the degassing tanks have proven not to be effective, or are inadequately sized, the use of an oil reclamation unit should be considered. All oil from drainers should be

collected or directly piped to an oil reclamation unit. Considering the cost of typical mineral oil (approximately $25 per gallon), a standard compressor with two seals can use $1,000 of oil per day if oil cannot be returned to the reservoir.

Figure 7.23.9 shows a typical oil reclamation unit which has capability to degas all oil entering the unit. In large installations, this unit may be justified for direct installation downstream of the drainers taking suction from the degassing tank and returning oil to the seal oil reservoir. For smaller systems, the purchase of one unit should be considered for the site. Gas entrained liquids can then be collected and transported to the unit for reclamation at specified intervals or the unit can be temporarily installed between the degassing tank and the seal oil reservoir.

System reliability considerations

The contaminated oil drain system is one of the most neglected subsystems in a compressor seal oil system. It is out of the way, its function is not fully understood, and it is not usually monitored with accuracy. Following are a few suggestions concerning drainer system reliability:

1. Care should always be given to the inspection of site glasses.
2. Oil samples to determine adequate degassing (flashpoint determination) should be taken periodically.
3. Leakage rates from the drainer should be regularly measured to determine the condition of the seals.
4. The atmospheric side seal drain should incorporate site glasses as a means of monitoring flow visually to determine if flow quantities have significantly changed.
5. It must be remembered that control valves can be used as rough flow meters when pressure across the valve and valve travel are known. A control valve can give an indication of change of flow rates in the system and will indicate any change in seal oil flows. It must be remembered that the atmospheric side seal oil flows will vary with changing with reference pressure. However, in most installations once units are on-line, reference pressure is relatively constant. Therefore, significant changes in seal oil control valve position will indicate deterioration of seals.

This system will be discussed further when the subject of buffer gas systems is studied in subsequent chapters.

Best Practice 7.24

Replace mature plant switches with TMR transmitters in all trip circuits for optimum oil console and serviced unit reliability.

Oil systems installed as late as the 1980s use single switches for pressure and temperature protection of machine components.

These old devices expose the plant to unscheduled shutdowns.

Triple modular redundant (TMR) smart transmitters (two-out-of-three voting for a trip) ensure accurate and reliable operation, and prevent spurious trips – which expose the plant to millions of USD in revenue losses.

Considering the current (2010) high daily revenue values, it is easy to justify the installation of TMR smart transmitters for all trip circuits in critical equipment installations.

Lessons Learned

Many plants have registered low machine reliability and corresponding revenue losses because of the malfunctioning of old instrumentation.

Benchmarks

This best practice has been in use since the late 1990s, and clients that we have audited are recommended to replace all trip switches in auxiliary systems with TMR two-out-of-three voting logic. Where our recommendations were followed, optimum machine reliabilities were achieved (99.7 % +).

B.P. 7.24. Supporting Material

Prior to the use of smart transmitters, single switches were used for shutdown logic on critical equipment trains. PM schedules were usually at 3 month intervals to check switch settings on-line but were usually ignored because they had caused unit trips in the past.

The use of TMR logic for all oil system shutdown logic ensures accurate response for shutdowns and prevents spurious unit trips due to transmitter malfunction.

Best Practice 7.25

Monitor control valve position in all oil systems to determine component wear (rotary pumps, bearings and seals) to ensure corrective action is taken during a turnaround.

Marking the position of control valves (marking the stem and valve yoke with a straight edge) at the beginning of a run will give an instant indication of component wear for the following items:

- Rotary pumps (screw or gear) – if the bypass valve is closing over time.
- Bearing wear – if the lube oil supply valve is opening over time.
- Control component wear – if the control oil supply valve is opening over time.
- Seal wear – if the seal oil supply valve is opening over time.

Check the position of all marked control valves prior to the turnaround meeting to determine if the affected components need replacement during the turnaround.

Lessons Learned

Failure to mark and monitor control valve stem position in oil systems has led to many surprises and replacements soon after a turnaround. Monitoring of valve stem position would have identified worn components and allowed replacement during a turnaround.

Remember that turnaround action does not affect product revenue, unplanned action does!

Replacement of an oil pump can take two days considering alignment.

Replacement of a bearing or seal can take three to five days!

Benchmarks

This best practice has been used since the late 1980s during all field visits. This practice has saved many millions of USD by moving component replacement to the 'turnaround revenue loss fee zone'!

B.P. 7.25. Supporting Material

Please refer to material in B.P: 7.13.

Best Practice 7.26

Check system transient functions (pump transfer) immediately before turnaround.

We recommend that the following procedure be followed immediately before each scheduled shutdown of critical (un-spared) machinery units:

1. Confirm that the auxiliary oil pump start switch actuates at the proper setting and starts the auxiliary oil pump.
2. Put the auxiliary oil pump into auto start mode, and station an operator at auxiliary oil pump with instructions to immediately manually start auxiliary oil pump if compressor unit trips.
3. Trip the main oil pump and observe the following (with strip charts, if possible):
   A. Auxiliary oil pump starts without unit trip
   B. Lowest tube oil pressure during transient
   C. Lowest control oil pressure during transient
   D. That all control valves remain stable

The above procedure is the only way to ensure that the transient response of the oil system will not cause a trip during operation.

The amount of oil taken by the machinery components during operation is more than during the stationary unit case (when the unit is not operating).

Most functional oil system checks are performed at turnarounds with the unit off-line, only to find during operation that the system cannot respond to a transient event without a unit trip.

Performing the transient check just prior to the turnaround will allow ample time for any corrective action.

Lessons Learned

In many cases, oil system functional checks during a turnaround confirm that the transient function of the system is acceptable, only to find after the turnaround that a unit trip has occurred during a transient event.

As mentioned above, the amount of oil taken by the machinery components during operation is more than during the stationary unit case (when the unit is not operating).

The only way to accurately confirm oil console transient response is while the unit is in operation.

The only way to avoid exposure to a unit trip during plant operation is to perform the transient test immediately prior to a planned shutdown.

Benchmarks

This best practice has been employed since 1990 to accurately check the transient response of critical machinery oil systems prior to a planned shutdown and to define an action plan for corrective action if required.

B.P. 7.26. Supporting Material

Functional testing

Having satisfactorily installed and flushed the auxiliary system, all auxiliary equipment should be functionally tested, and all instruments and controls checked for proper setting prior to operation of equipment. A functional test outline and procedure is included at the end of this section. We will highlight the major considerations of the procedure at this time.

It is recommended that the console vendor and/or the equipment purchaser prepare a detailed field functional test procedure and calibration check form. The format of this procedure can follow the factory test procedure if this is acceptable. As a minimum, the auxiliary system, bill of material, and schematic should be thoroughly checked, in order to include the calibration and functional test of each major component in the auxiliary system. That is, components on consoles, and up at unit interfaces. A detailed record should be kept of this functional test procedure – this will help significantly during the operation of the unit. The functional test procedure should be carried out initially without the critical equipment running, and then with the auxiliary system running as closely as possible to design operation conditions. The functional test procedure should first require that all instrumentation be properly calibrated before proceeding. Each specific functional test requirement should then be performed and the results noted. If they do not meet specified limits, testing should stop, and components should be corrected at this point. Each step should be followed thoroughly to ensure each component meets all requirements. It is recommended that the operators assigned to this particular unit should assist in functional testing to familiarize themselves with the operation of the system. In addition, site training courses should be conducted prior to the functional test to familiarize operators with the system's basic functions. This training, again, significantly increases understanding of the equipment and ensures unit reliability.

Satisfactory acceptance of a functional test then ensures that the unit has been designed, manufactured, and installed correctly such that all system design objectives have been obtained and that equipment reliability is optimized.

One remaining factor to be proven is the successful operation of the system with the critical equipment unit in operation. During initial start-up, it is recommended that the functional test be re-performed with the unit operating. While this advice may seem dangerous, unless the unit operators are assured that the subject system has the ability to totally protect critical equipment while operating, auxiliary equipment will never be tested while the unit is in operation.

Remember, critical equipment is designed for 30 years' life, or greater. The components that comprise the auxiliary system are many, and have characteristics that will change with time. Therefore, their reliability can only be maintained if the systems are totally capable of on-line calibration and functional checks. The functional pre-commissioning procedure should be modified to include an on-line periodic functional checking procedure.

At this point, we can clearly see that the major determination of continued equipment reliability rests with the operation, calibration and maintenance of the equipment. In order to assume maximum continued auxiliary equipment reliability, periodic on-line functional checks and calibrations must be performed. How can this be done? The only way is by convincing plant operations of the safety and reliability of the procedure for on-line checking. This can be reinforced during pre-commissioning, by including operators in functional testing checks and on-site training sessions, to show the function of the system. A site training course modified for the specific equipment would prove immensely valuable in achieving this.

Only by involving unit operators in the pre-start check-ups can it be hoped to establish a field functional checking procedure that will be utilized and followed through. Remember, a pressure switch of less than $400 in cost could cause equipment shutdown that could reduce on-site revenue by around one to two million USD per day. The pressure switch selected could be the best, the highest quality in the world, and be properly installed and set. If its calibration is not periodically checked, it could cause an unnecessary shutdown of equipment and result in this revenue loss.

Functional lube/seal system test procedure outline

Objective:	To confirm proper functional operation of the entire system prior to equipment start-up
Procedure format:	Detail each test requirement. Specifically note required functions/set points of each component. Record actual functions/set points and *all* modifications made.
Note:	All testing to be performed *without* the unit in operation.

I Preparation

A Confirm proper oil type and reservoir level
B Confirm system flush is approved and *all* flushing screens are removed
C Confirm all system utilities are operational (air, water, steam, electrical)
D Any required temporary nitrogen supplies should be connected
E All instrumentation must be calibrated and control valves properly set
F Entire system must be properly vented

II Test procedure

A Oil reservoir
  1. Confirm proper heater operation
  2. Check reservoir level switch and any other components (TIs, vent blowers, etc.)

B Main pump unit
1. Acceptable pump and driver vibration
2. Absence of cavitation
3. Pump and driver acceptable bearing temperature
4. Driver governor and safety checks (uncoupled) if driver is a steam turbine

C Auxiliary pump unit
Same procedure as item B above.

D Relief valve set point and non-chatter check

E Operate main pump unit and confirm all pressures, differential pressures, temperatures and flows are as specified on the system schematic and/or bill of material

F Confirm proper accumulator pre-charge (if applicable)

G Confirm proper set point annunciation and/or action of *all* pressure, differential pressure and temperature switches

H Switch transfer valves from bank ‘A’ to bank ‘B’ and confirm pressure fluctuation does not actuate any switches.

I Trip main pump and confirm auxiliary pump starts without actuation of any trip valves or valve instability.

Note: Pressure spike should be a minimum of 30% above any trip settings

J Repeat step I above but slowly reduce main pump speed (if steam turbine) and confirm proper operation

K Simulate maximum control oil transient flow requirement (if applicable) and confirm auxiliary pump does not start

L Start auxiliary pump, with main pump operating and confirm control valve and/or relief valve stability. Note: Some systems are designed to *not* lift relief valves during two pump operation

III Corrective action

A Failure to meet any requirement in Section II requires corrective action and retest

B Specifically note corrective action

C Sign-off procedure as acceptable to operate

Typical initial field functional lube/seal system test procedure baseline document

Item: Reference DWGS:	Turbine utility	P&ID _____
	Compressor utility	P&ID _____
Purpose:	To fully prove functional operation of entire lube/seal system, including all permissive, alarm and shutdown functions prior to initial operation of the unit.	
Note:	All testing to be performed without the unit in operation. When specified values are not satisfied correct and retest.	
Preparation:	Prior to testing of the system, confirm and sign off that the following has been checked:	

- Oil reservoir at proper level
- Specified oil is used
- Oil heater in operation
- Oil cooler water supply on
- System clean (all test screens out)
- Instrument air in operation at all instruments
- Temporary N_2 supply connected
- All instrumentation noted on attached list has been calibrated to specified values
- Steam lines to console blown
- Entire system vented

Item	Description	Specified value	Actual value	Witnessed by
1.	Record reservoir temp. rise in four (4) hours with heater on. Read on T.I.	Record actual ΔT	______	
2.	Check reservoir level switch	High level	High	
	setting Read on Annun.	14" from top. low level	______	
		45" from top	Low	
			______	
3.	Energize aux. lube pump check:			
3A.	Pump vibration	0.2 in/sec	______	
3B.	Motor vibration	0.2 in/sec	______	
3C.	Cavitation	None	______	
3D.	Pump/motor brg. temp	165°F	______	
4. (see note 1)	Block in aux. pump using pump discharge valve-set relief valve Valve chatter is not acceptable.	3130 kPa	______	
5.	Confirm the ‘Aux. pump running’ Annun. is actuated by switch By shutting off pump and restarting while reading pressure on PI	690 kPa rising	_____	
6.	Allow system to heat up to 49°C downstream of coolers and adjust the following items (if required) to attain specified values:			
6A.	Back press regulator Read value on PI	2262 Kpa	_____	
6B.	Lube oil supply valve PCV Read value on PI	124–138 Kpa	_____	
6C.	Control oil valve PCV Read value on PI	883 Kpa	_____	
6D.	LP case seal oil differential valve *PDCV*. Supply N_2 press, of *5 PSIG* at gas reference side of *PDT* and read differential press on PDI	241 Kpa	_____	
6E.	HP case seal oil differential valve. Supply N_2 pressure of *30 PSIG* at gas reference side of *PDT* and read differential press on PDI	241 kPa		

(Continued)

Item	Description	Specified value	Actual value	Witnessed by
7.	Record the following:			
7A.	Pump disch. press on PI	Approx. 2600	_____	
7B.	Oil temp. upstream on TI	Less than 65°C	_____	
7C.	Oil temp. downstream on TI	49°C		
7D.	Cooler/filter ΔP on PDI Switch transfer valve and record:	Less than 70 kPa	_____	
	Bank 'A' ΔP		_____	
	Bank 'B' ΔP			
7E.	Lube press at console on PI	*Approx. 283 kPa	_____	
7F.	Control press at console on PI	*1091 kPa	_____	
7G.	LP case on ΔP PDI	241 kPad	_____	
7H.	HP case on ΔP PDI	241 kPad	_____	
7I.	Lube press at unit on PI	124 to 138 kPa	_____	
7J.	Control press at unit PI	883 kPa	_____	
7K.	All 'sight' glasses show oil flow		_____	
7L.	Lube oil head tank is full	–	_____	
7M.	Turbine accumulator press on PI *Adjust as required to attain proper values at the unit	*Approx. 900 kPa	_____	
8.	Record seal oil drainer level for each drainer for one hour.			
	Drainer A	2 fills per hour	_____	
	Drainer B		_____	
	Drainer C		_____	
	Drainer D		_____	
9.	Switch transfer valve from 'A' to 'B' bank and observe press fluctuation on PI and confirm that PSL does not actuate.			
10.	Bleed low side of filter switch PD SH and confirm PDAH actuates at specified value.	241 kPa Rising	_____	
11.	Bleed pressure off PSL (low oil press) and confirm @ unit annun. *PAL* actuates at specified value. Read PI	90 kPa Falling		
12.	Increase temporary N_2 press on *PDSL* (L.P. case seal low ΔP alarm switch) and confirm Annun. *PDAL* actuates at specified value. Read PDI	207 kPa Falling	_____	
13. (see note 2)	Repeat above for *PDSL* (H.P. case seal low ΔP alarm) confirm Annun. *PDAL* actuates read on PDI	207 kPa Falling	_____	
14.	Bleed pressure off PSL (low oil trip switch) and observe: A. Annun. PALL functions Read PI B. T&T valve closes C. Valve rack closes	76 kPa Falling	_____	
15. (see Note 2)	Increase temporary N_2 press on *PDSLL* (L.P. case seal low ΔP trip switch) and observe that: A. Annun. *PDALL* functions. Read on PDI B. Action occurs as in 14 above	138 kPa Falling	_____	
16.	Repeat above for *PDSLL*(HP case seal low ΔP trip) and observe Annun. PDALL functions at specified value. Read PDI	138 kPa Falling	_____	
17.	Shut off aux. oil pump and confirm all valves are stable		_____	
18.	Time rundown of oil tank. Observe level switch *LSL* actuates Annun. *LAL*	Approx. 4" above Tang. Line	_____	
19.	A. Disconnect main lube oil pump from turbine B. Drain steam inlet line to main lube pump turbine C. Drain turbine D. Confirm turbine bearing cooling water is on E. Confirm trip is reset			
20.	Open inlet valve, gradually bring turbine up to speed and confirm the following:		_____	
	A. Rated speed (Strobe Tac)	3600 RPM		
	B. Inlet press PI	2274 kPa	_____	
	C. Inlet temp. TI *Temp*	327°C	_____	
	D. Exhaust press PI *Temp*	517 kPa	_____	
	E. Vibration (using metrix vibration instrument)	0.2 in/sec.		
21.	Disable governor and check overspeed trip 3 times.	4140 RPM	_____	
22.	Manually trip turbine using hand trip.			
23.	Check pump/turbine alignment and couple up main pump.			
24.	Slowly bring pump up to speed and check:			
	24A. Pump vibration	0.2 in/sec	_____	
	24B. Cavitation	None	_____	
	24C. Pump brg. temp.	165°F	_____	

Item	Description	Specified value	Actual value	Witnessed by
25.	Adjust Governor so all valves are as noted in step 7. Record speed RPM.		_____	
26.	Block in pump using pump discharge valve. Set relief valve *PSV.* Valve chatter is not acceptable.		_____	
27.	With turbine operating at speed noted in step 25 and disch. block valve open, manually trip turbine and observe:			
	A. Aux. pump starts.		_____	
	B. Min. press spike on PI lube	90 kPa		
	PI Control	T&T valve does not close	_____	
	PDI LP Case ΔP seal	207 kPa		
	PDI HP Case ΔP seal	207 kPa	_____	
	C. Alarm and trip switches connected with lube and seal oil are not actuated.		_____	
	D. All valves are stable.		_____	
28.	Restart pump turbine and dump control oil pressure using hand valve at turbine. Observe that no alarm or trip lights are actuated and that all valves remain stable.		_____	
29.	Start aux. oil pump with turbine operating and observe:			
	A. RVs do not lift		_____	
	B. All control valves are stable		_____	
	C. Oil pump turbine does not hunt (speed remains stable)			
30.	Stop aux. pump motor and observe:			
	A. All control valves remain stable.		_____	
	B. Min. press. spike on PI lube	90 kPa		
	PI Control	T&T valve does not trip	_____ _____	
	PDI LP Case ΔP	207 kPa	_____	
	PDI LP Case ΔP	207 kPa	_____	
31.	With turbine operating. Bleed press, from PSL and observe aux. pump starts and that all valves remain stable.		_____	
32.	Having satisfactorily completed all items above, secure both aux. and main pump and sign off as being acceptable for operation.		_____	
	1. NOTE At this point, elect one driver and continue to operate the console 24 hours a day.			
NOTE 1	RVs will continuously pass a small stream of oil. Actual setting will be that pressure at which stream volume increases. Observe by un-bolting FLGS at reservoir. Accumulation value is with pump discharge block fully closed.			
NOTE 2	Block out gas signal to diff. control valve during this step.			

Best Practice 7.27

Perform an oil system site audit if the subject system has caused critical machinery unit shutdowns in the past that have resulted in lost product revenue.

Use the original equipment manufacturer (OEM) or a qualified consultant to review past problems and conduct a site audit to cover the following items as a minimum:

- Determine the present operating condition of each system component
- Verify that all set points are as specified in the instruction manual
- Check component selection, where required and verify proper component sizing
- Define a remedial action plan which in addition may include:
- Off-line testing at the first opportunity
- Requests for additional component sizing information from the OEM
- Modifications to the operating procedure for this oil system

Lessons Learned

Continuing critical machinery unit oil system reliability issues need to be evaluated in terms of lost revenue costs to justify an oil system site audit to discover root causes and corrective action.

Benchmarks

I have performed site auxiliary system audits since 1990 for the resolution of long term reliability issues that have lost the end user large amounts of revenue. This procedure has resulted in oil system and machinery unit reliabilities of 99.7%+.

B.P. 7.27. Supporting Material

See B.P: 7.5 for oil system component audit information guidelines.

Best Practice 7.28

Be sure to require all oil system interconnecting supply and drain piping to have maximum oil velocities set to prevent excessive pressure drops, which will affect control valve sizing and expose bearing brackets to flooding (possible split line oil leaks).

The maximum velocity values are:

- Supply piping – Maximum 6 ft/second
- Drain piping – Maximum 1 f/second

Our best practice recommendation is therefore to include interconnecting oil piping design and manufacture in the machinery vendor scope of supply.

Oil system vendor backpressure and supply valve sizing is based on assumed interconnecting oil pressure drops.

If the interconnecting piping is not sized properly and has greater pressure drop than anticipated during the selection of the control valves, the control valves may not be properly sized.

Improper control valve sizing will affect valve transient response and stability leading to lower than anticipated unit reliability.

In addition, higher than anticipated interconnecting drain pipe pressure drop can raise the level of oil in the bearing bracket and cause split line oil leaks that can expose the machinery and environment to fires in hot service areas (steam turbine surfaces and pipes).

Lessons Learned

Lower than anticipated unit reliability and safety issues (plant fires) have resulted from improper sizing of interconnecting oil piping.

The use of smaller than required interconnecting oil supply pipes has resulted in backpressure (bypass) valve replacement due to the higher pressure necessary to force the oil up to the unit and corresponding low valve opening causing valve noise and/or wear and or instability.

Additionally, fires have been experienced in steam turbine applications on the steam inlet end, when bearing bracket oil leaks were caused by drain oil piping being smaller than required, which raised the oil levels in the bearing bracket to the split line.

Benchmarks

This best practice has been used since the mid-1980s when I was involved with the start-up of a large petrochemical plant. Allowing the contractor (EP&C) to select and manufacture the interconnecting supply oil piping resulted in having to throttle the backpressure control valve inlet block valve to prevent valve noise, wear and instability, until the valve could be resized for a smaller trim at the first turnaround (three years of exposure to unnecessary unit trips caused by improper valve response during transient conditions). Since that time, this best practice has been used on all new projects to ensure maximum unit reliability (99.7% or greater).

B.P. 7.28. Supporting Material

See B.P: 7.5 for supporting material.

Best Practice 7.29

Label oil system piping with colored tape to help personnel to understand system operation and how to take corrective action quickly to prevent machinery unit trips.

Color coded and identified oil system piping greatly increases site personnel awareness of oil system operation.

Using colored tape to define each individual line of the system (supply lines, return lines, bypass lines) involves personnel and promotes 'ownership', thus increasing system safety and reliability.

Lessons Learned

Many critical machine unit shutdowns are the result of not monitoring the local instrument and components in the system. Failure to properly label piping, instruments and components leads to neglect and corresponding low oil system reliability.

Benchmarks

This best practice has been used since 2000 to increase understanding of oil system function and reliability of these systems. Operations, engineering and maintenance personnel, in many cases, have all participated in the labeling and have shown increased interest in understanding the system components function and local monitoring of these components.

B.P. 7.29. Supporting Material

Types of lubrication systems

The classical lube system arrangement consists of a steam turbine-driven, main pump and motor-driven auxiliary pump. In this section, we will devote our attention to other system variations.

Shaft-driven positive displacement main pump, motor-driven auxiliary pump

Refer to Figure 7.29.1. In this system, a shaft-driven main pump is supplied. The purpose of a shaft-driven pump is to continuously provide some amount of lubrication to the system bearings while the critical equipment is in operation. The idea is that on coastdown or start-up, the system would never be without

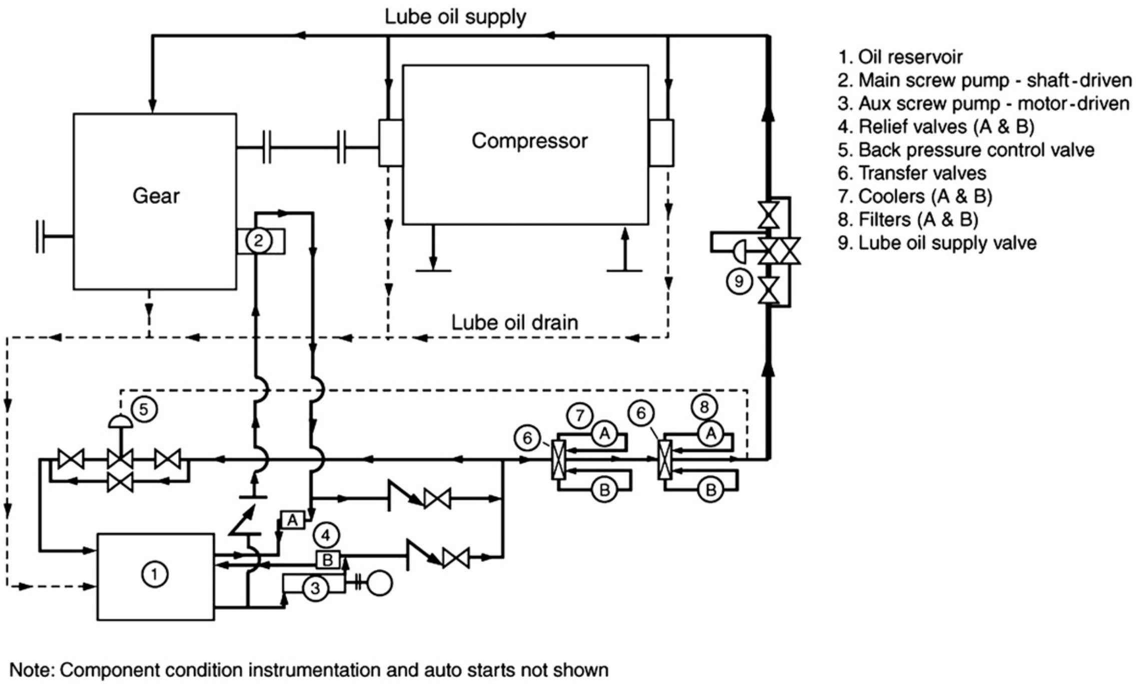


Fig 7.29.1 • Lube oil system main pump shaft-driven (Courtesy of Elliott Co.)

lubrication oil supply, even in the event of failure of the stand-by pump.

Such a system is useful in areas where an alternative form of energy is not available for main and spare pumps, as on platforms or in chemical plants where a utility steam system is not installed.

However, the system must still be designed to effectively lubricate all bearings prior to the main pump attaining full flow rate (speed). To facilitate this, the auxiliary pump is designed to start before the main critical equipment driver is started (permissive arrangement), and either automatically or manually shut off once the shaft-driven pump attains sufficient speed. On shutdown, the reverse occurs – the auxiliary pump will start on deceleration of the main pump. The signal to start the auxiliary pump can be critical equipment speed or system pressure. If a system is not designed for automatic shutoff of the auxiliary pump, care must be taken to ensure that it is designed for continuous two-pump operation. It is recommended that auxiliary pumps be shut off during critical equipment main pump operation.

A particular concern with this system design is to ensure priming of the main pump. Frequently, the main pump is at a significant height above the fluid reservoir. In this case, care must be taken to ensure proper priming of main pump prior to start-up. Many systems incorporate a priming line from the auxiliary pump to fill the main pump suction line. This requires a check valve (foot valve) in the main pump suction line and a self-venting device to ensure air is not entrained in the suction line of the main pump. Frequently, systems of this type do not incorporate a vent line and are susceptible to main pump cavitation on start-up.

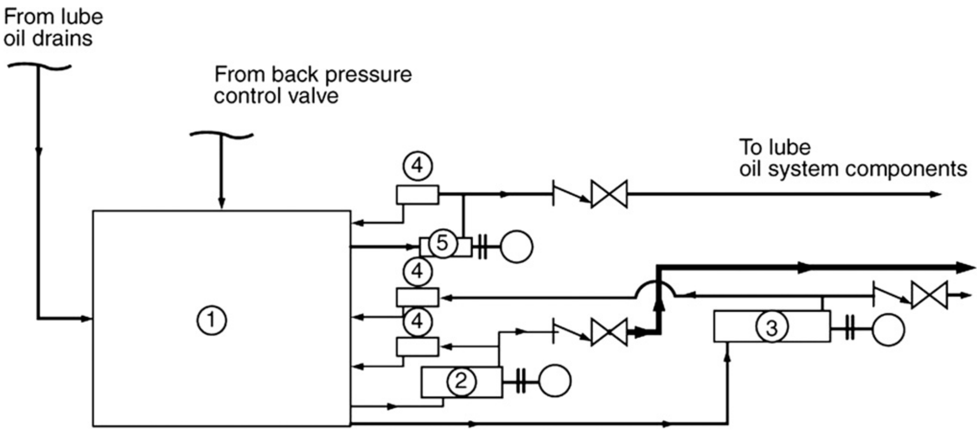
Referring to Figure 7.29.1, let's examine the bypass valve response in the event that the auxiliary pump does not shut off after the shaft-driven pump has attained full flow (speed). Upon initial auxiliary pump start-up, flow is supplied to the system and the bypass valve opens to control system pressure to a pre-set value, thereby bypassing excessive auxiliary pump (stand-by pump) flow. Upon starting of the critical equipment unit, the bypass valve will react to increasing main pump flow and will gradually open to the point that it would be at maximum stroke if the stand-by pump remained in operation. When the stand-by pump is shut down, the response of the bypass valve must be equal to the decrease of flow from the auxiliary stand-by pump such that system pressure does not drop below trip setting.

Refer back to the concepts of an equivalent vessel and orifice discussed in previous sections. In this case, the supply to the equivalent vessel (from the auxiliary pump) instantaneously drops while the demand from the critical equipment is constant thus causing an instantaneous drop in equipment vessel pressure. Prior to shutdown of the auxiliary pump, excessive supply was recirculated by the bypass valve. Upon rapid decrease of supply flow, the bypass valve must decrease demand at the same rate to ensure pressure in the system (equivalent vessel) is maintained at a constant value. Failure to do this can result in critical equipment shutdown.

Main and auxiliary A.C. motor-driven pumps, D.C. motor-driven emergency pump

Refer to Figure 7.29.2. This application would be used in facilities where steam systems are not available, such as chemical plants or in operations at remote locations or on offshore platforms. In this case, both pumps (main and stand-by pump), would shut down in the event of a power failure. If the critical equipment unit is also motor driven, it too will cease operation.

The design objective of this auxiliary system is to provide a sufficient flow, via an emergency pump driven by a DC source (direct current), for a sufficient time to promote the safe



1. Oil reservoir
2. Main oil pump - motor-driven
3. Aux oil pump - motor-driven
4. Relief valves
5. Emergency oil pump - DC motor-driven

Notes: Other major components not shown
Component condition instrumentation and auto start of aux and emergency pumps not shown

Fig 7.29.2 • Lube oil system, main pump shaft-driven (Courtesy of Elliott Co.)

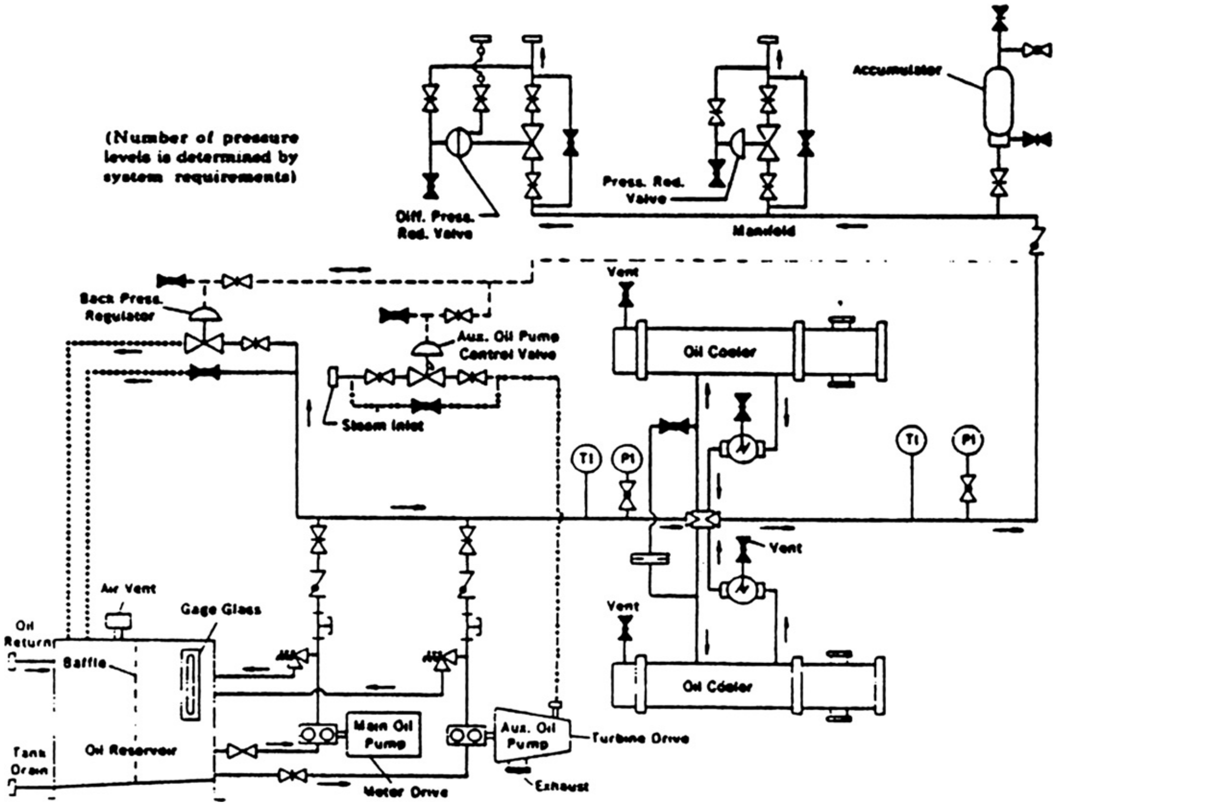


Fig 7.29.3 • A.C. motor-driven main pump, steam turbine auxiliary pump drive (Courtesy of Elliott Co.)

coastdown of the critical equipment unit. The DC power source can be obtained from a UPS (Uninterrupted Power Supply) system or other sources.

Coastdown times of the driven equipment vary between 30 seconds to in excess of 4 minutes. Sufficient emergency pump energy must be available. The emergency pump would be activated in this case on auxiliary system pressure drop. Note that the emergency pump is not a full capacity pump. It is sized to only provide sufficient flow during equipment coastdown to prevent auxiliary system component damage.

A.C. motor-driven main pump, steam turbine-driven auxiliary pump

Refer to Figure 7.29.3 for a diagram of this system. A system of this type would be used in a plant where dual pump driver power is available, but steam is at a premium. Therefore, the turbine driver is not desired to be used for continuous operation. This system requires continuous attention in terms of steam turbine steam conditions. The turbine must be capable of accelerating rapidly since a system accumulator is not included. Therefore, steam conditions at the turbine flange must be maintained as specified. That is, steam must be dry. A steam trap is recommended to be installed to continuously drain condensate from the steam. To ensure rapid turbine acceleration, the turbine inlet valve must be a snap open type (less than 1 second full open time) to minimize auxiliary pump start-up time. Experience has shown that this type of system is not reliable in terms of auxiliary pump starts. If used, this type of system should employ a large accumulator to provide sufficient flow to the system for approximately ten seconds as a minimum since the start-up time of a steam turbine is much slower than that of a motor-driven pump.

Centrifugal pump system

Figure 7.29.4 depicts a dual centrifugal pump system. The main pump is steam turbine-driven and the stand-by pump is motor-driven. Such a system can be used in areas of the world where large ambient temperature fluctuations are not present; Middle East, etc. Since the pumps are dynamic type, note that a bypass valve is not used in this system. This is because the flow rate of dynamic pumps are determined by system resistance. Therefore, the control valve in this application senses pressure in the system and adjusts system resistance at the pump discharge to supply the required flow.

Referring back to the concept of an equivalent vessel, let us examine the case of increased system demand as in the case of bearing wear. Bearing wear will gradually increase system demand and reduce system equivalent vessel pressure. Consider in this case the system to be the equivalent vessel. Since the control valve senses equivalent vessel pressure, a reduction in the equivalent vessel pressure will open the valve to provide greater pressure at the sense point. This action in turn will reduce pump discharge pressure. Referring to the characteristic curve of a centrifugal pump, reduced pump discharge pressure results in increased pump output flow, thereby compensating for the increased demand requirement.

In the event of a sudden system flow increase, a sudden change in the equivalent vessel pressure would be experienced.

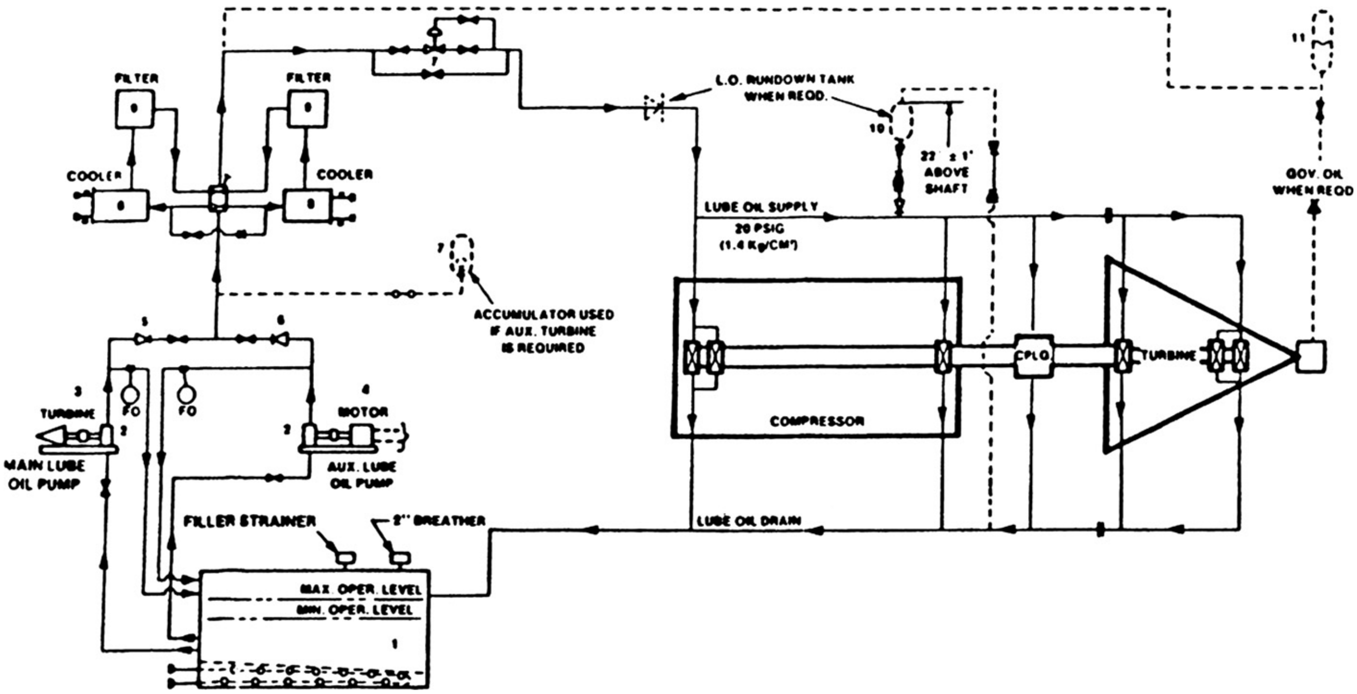


Fig 7.29.4 • Lubrication system – centrifugal pumps (Courtesy of Dresser-Rand)

This would result in a sudden drop of equivalent vessel pressure, resulting in a sudden opening of the control valve. However, in this case, the sudden pressure drop would initiate starting of the stand-by pump. As soon as the stand-by pump were to start, supply flow to the equivalent vessel would quickly increase, causing a corresponding increase in pressure. However, the control valve sense point would note the pressure increase and rapidly shut the control valve, thus increasing the system resistance to both pumps, and forcing pumps to a lower flow rate.

An important consideration in this system is that both main and auxiliary pumps operate at essentially the same speed and have essentially the same characteristic curves. (This would not be the case if one pump had excessively worn internals.) Therefore, the total output flow of the pumps would be a result of the combined curve of the pump output. Refer to Figure 7.29.5 for a view of the combined effect of both pumps

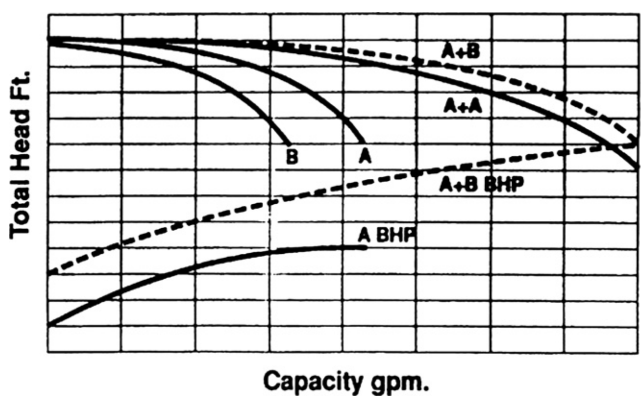


Fig 7.29.5 • Non-identical pumps with stable head curves

operating in parallel. From this example, the importance of ensuring correct similar characteristics of both main and auxiliary pumps in systems incorporating dynamic pumps can be seen. If one pump is steam turbine-driven, it is important to periodically check speed and correct the turbine governor setting if necessary. The motor and turbine speeds should be the same.

Dual motor-driven positive displacement pump system with a rundown tank

This system is similar to that shown in Figure 7.29.2 with the exception of the absence of an emergency pump. In this case, the user has elected to have a rundown tank in the system that will supply sufficient lube oil for rundown (refer to Figure 7.29.6).

The function of the tank is to provide oil to all bearings at a lower pressure than normal, but still sufficient to preclude bearing damage during emergency shutdown. In this case, the tank will drain when both main and auxiliary pumps cease to function. An important consideration is the material of the rundown tank since it represents a large vessel downstream of the filter in the system. Any rust scale or corrosion present in this tank will go directly into the bearings on shutdown and could cause a significant problem. Rundown tanks should also be sized to provide sufficient flow for the entire coastdown period. The calculation of the equipment coastdown time must include process system information. The lower the system pressure (resistance) during the coastdown period, the longer the equipment will continue to turn. The vendor and user must coordinate closely regarding the rundown tank sizing. It is recommended that a small amount of flow be continuously circulated through the rundown tank to prevent sediment accumulation and maintain operating viscosity (in cold regions).

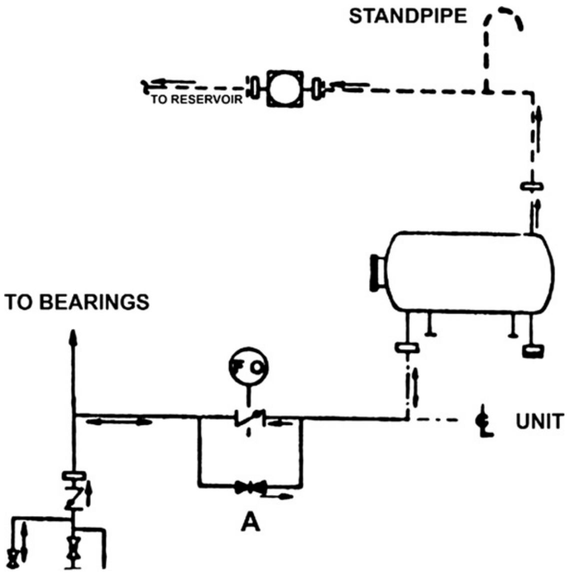


Fig 7.29.6 • Lube oil rundown tank system with continuous overflow (Courtesy of Elliott Co.)

This is accomplished by installing a return line from the tank to the oil drain line that incorporates an orifice to regulate flow (approximately 8 LPM [2 GPM]).

One final comment concerning lube systems with rundown tanks or emergency pumps. It is strongly recommended that in the case of equipment shutdown, even though equipment is furnished with rundown tanks and emergency pumps, bearings should be checked for wear upon coastdown. Failure to do so could result in catastrophic damage to the equipment if bearings were failed at shutdown. Additionally, in systems employing gears, gear box inspection covers should be removed and the gear mesh checked since spray elements, if clogged, may not distribute sufficient lubricating flow to the gear mesh in emergency conditions.

Centrifugal shaft-driven main pump, motor-driven positive displacement auxiliary system pump

Refer to Figure 7.29.7. This system combines two types of pumps, dynamic and positive displacement. An important consideration is that the maximum pressure of the auxiliary (positive displacement) pump be limited below the maximum pressure of the shaft-driven (dynamic) pump. If this is not done, the shaft-driven pump output pressure, when operating with the stand-by pump, will be less and its output flow will be reduced to zero. Continued operation in this mode will result in overheating of the shaft-driven pump and failure. The stand-by pump discharge pressure setting is regulated by a bypass valve or a relief valve in this case. The setting of this valve must be checked to be sure it is less than the dynamic pumps discharge pressure at the dynamic pump minimum flow point.

Arrangement options

In this section we will briefly discuss a few arrangement options available for lubrication systems. As mentioned, the arrangement of auxiliary systems directly determines system reliability since arrangement determines accessibility to component parts that must be serviced and calibrated while critical equipment is operating. Attention must be drawn to particular applications and the need to maximize component accessibility. It is recognized that certain applications contain minimal space for auxiliary equipment and that equipment must be arranged for the available spaces.

Integral auxiliary systems

Refer to Figure 7.29.8. Such a system incorporating the lube oil system in the baseplate of the critical equipment is used in remote applications and frequently on platforms since space is at

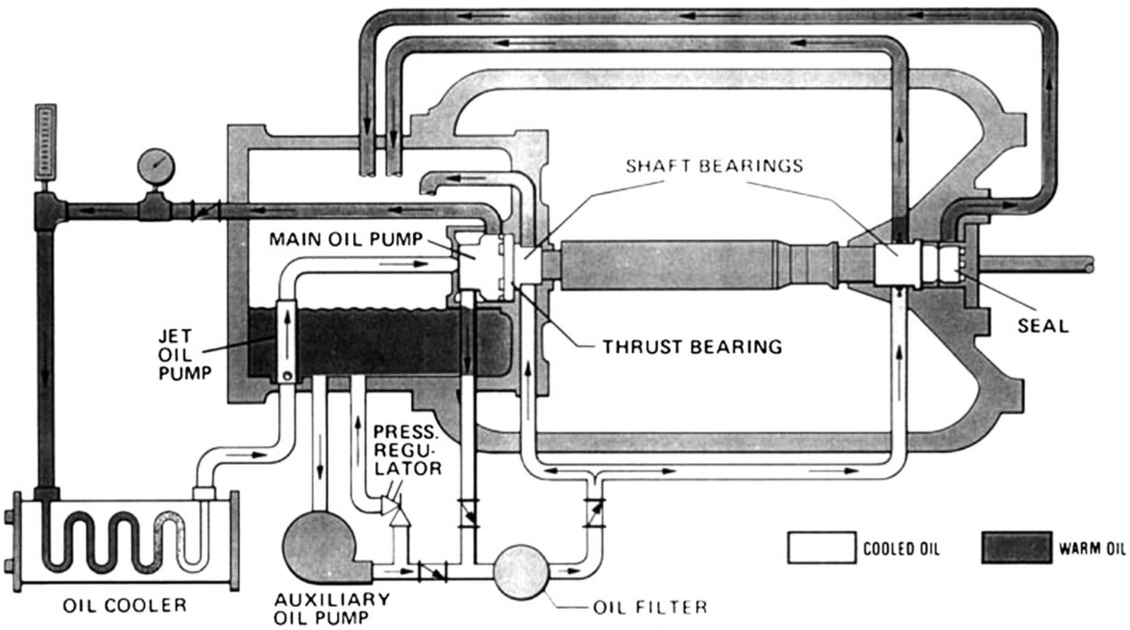


Fig 7.29.7 • Shaft-driven centrifugal pump, motor-driven auxiliary pump (Courtesy of York International Corp.)

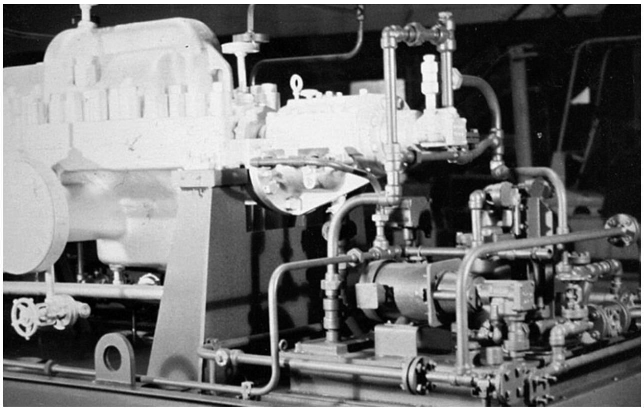


Fig 7.29.8 • Modularized oil console arrangement (Courtesy of Fluid Systems, Inc.)

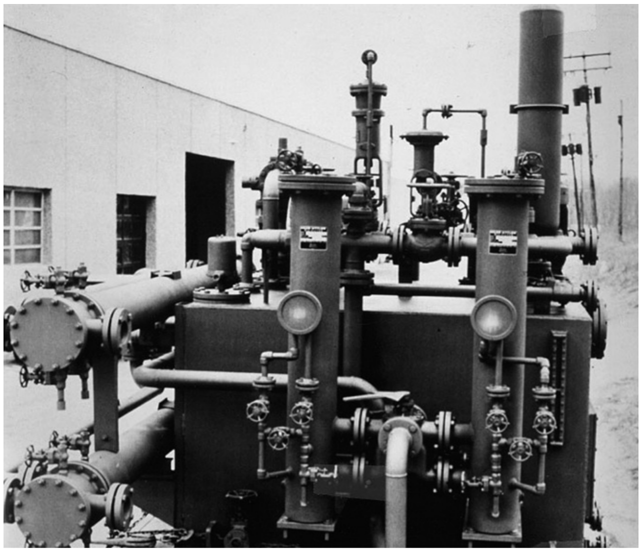


Fig 7.29.10 • Modularized oil console arrangement (Courtesy of Fluid Systems, Inc.)

Fig 7.29.9 • Horizontal oil console arrangement (Courtesy of G.J. Oliver, Inc.)

a premium. This system should be reviewed thoroughly in the design phase to optimize accessibility. Note that even though space is minimal, major components can be arranged such that accessibility is possible.

Horizontal console (no components on the reservoir)

Please refer to Figure 7.29.9. This console arrangement is fairly typical for a critical equipment lubrication system. Note the positions of components affording ample space for maintenance and equipment calibration. In addition, note the placement of interconnecting piping, thus allowing for maximum mobility on the console.

Reservoir integral with component

Refer to Figure 7.29.10. This arrangement is frequently used in restricted space locations or could be the result of an attempt by the original equipment vendor to minimize cost. Careful scrutiny of arrangement design early during the project can avoid problems. Note from Figure 7.29.10 that even though all components are mounted on the reservoir, accessibility is maximized to components. In this case, the reservoir was mounted approximately three feet below grade, thus allowing all components to be within easy maintenance reach.

Best Practice 7.30

Adjust critical equipment shaft vibration alarm settings to activate as low as possible (+50% of initial reading) to detect rotor condition change before component damage occurs.

Present industry practice (2010) is to high speed balance rotors which results in low initial field shaft vibration readings (typically less than 15 microns peak to peak).

Normal shaft vibration alarm settings are 50–60 microns which can amount to a 400% increase in shaft vibration before an alarm is activated.

Setting the shaft vibration alarm at 50% of the initial field value will allow early detection of rotor condition change, and initiate investigation and an action plan for corrective action before a rotor or component failure occurs.

Once the cause of change is known and accepted, the alarm setting can be increased to an increment of the new reading (suggested increment of 50% of new reading).

Lessons Learned

Failure to detect early rotor condition change has resulted in damage to compressor and turbine internals which has caused long shutdown periods (weeks or months) to repair stationary internals which are usually un-spared.

Recent experience (2010) with a generator exciter failure showed that shaft vibration increased from 10 microns to 30 microns, but was undetected since the alarm was set at 60 microns. Shortly thereafter the vibration exceeded 100 microns – at which point the exciter disintegrated.

Benchmarks

This best practice has been recommended and used on new equipment since the 1990s, to closely monitor shaft vibration condition changes and to prevent rotor and/or component failure.

B.P. 7.30. Supporting Material

In a recent case (2010), threshold shaft vibration and phase angle alarms (+50% vibration and +/−10% phase angle) were set on a compressor that contained prototype impellers of high head and stress, in order to warn of initiated cracks and to allow an orderly shutdown before internal compressor damage could occur.

Control Oil System Best Practices

Best Practice 7.31

Check the control oil accumulator every three months to ensure desired transient response, in order to prevent a steam turbine trip on low control oil pressure.

In order to ensure proper transient response, an accumulator must maintain the proper nitrogen pre-charge pressure and have the bladder intact (rupture free).

Accumulators cannot be checked for pre-charge and bladder condition on-line, since the nitrogen pre-charge pressure will be in equilibrium with the accumulator oil pressure.

In order to check the pre-charge and bladder condition, the accumulator must be isolated and drained. Nitrogen is then introduced at the correct pre-charge pressure to confirm fittings are leak free and the bladder is in good condition.

After the accumulator check and with the accumulator pre-charge as specified, the isolation valve must be slowly opened to ensure that the control oil pressure does not fall below the low control oil pressure setting that will trip the turbine.

FAI accumulator best practice is to install a bypass valve, with an orifice around the isolation valve to prevent rapid oil pressure drop when the accumulator is introduced back to the system.

Lessons Learned

Failure to periodically check the accumulator for pre-charge and bladder condition has resulted in many steam turbine unit trips during a transient condition (rapid load change: the surge valve opens rapidly, process control system rapidly changes turbine speed).

There has been a reluctance to check accumulators in some plants, because the check has caused a low control oil pressure trip on occasions when the accumulator isolation valve was opened too quickly.

Benchmarks

This best practice has been recommended since 1990, resulting in critical equipment train maximum reliability above 99.7%.

B.P. 7.31. Supporting Material

Accumulators

Referring to concepts previously discussed in this section, the equivalent vessel concept exactly defines an accumulator function. An accumulator is simply a vessel which compensates for rapid, short term flow disturbances in the auxiliary system. Most accumulators contain bladders (see Figure 7.31.1). It is important to remember that transient disturbances are on the order of micro seconds and usually less than five seconds in duration.

A schematic for a pre-charged accumulator is shown in Figure 7.31.2.

The pre-charge pressure is set at the pressure that the volume of the accumulator flow is required in the system. (This value is usually around 60–70% of the normal header pressure in which the accumulator is installed.) The quantity of oil available from a pre-charged accumulator is extremely low.

As an example, a system with a flow capacity of 120 GPM has a motor driven auxiliary pump that requires three seconds to attain full speed when started by a pressure switch or transmitter at 140 PSIG. Normal header pressure equals 160 PSIG. Determine the amount of oil that is required to prevent the pump header pressure from falling below 100 PSIG and the number of pre-charged 10 gallon accumulators required. (See Figure 7.31.3.)

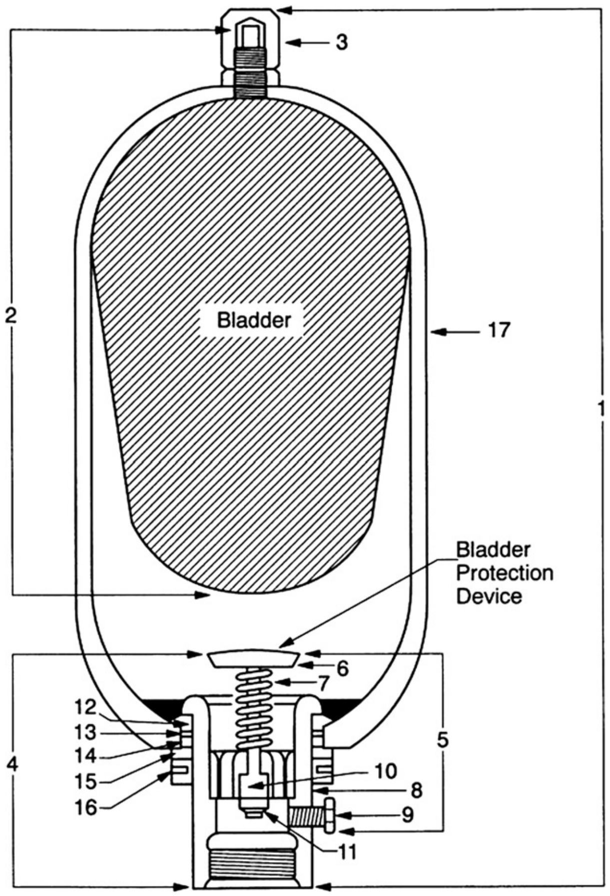


Fig 7.31.1 • Typical oil system accumulator (Courtesy of Greer)

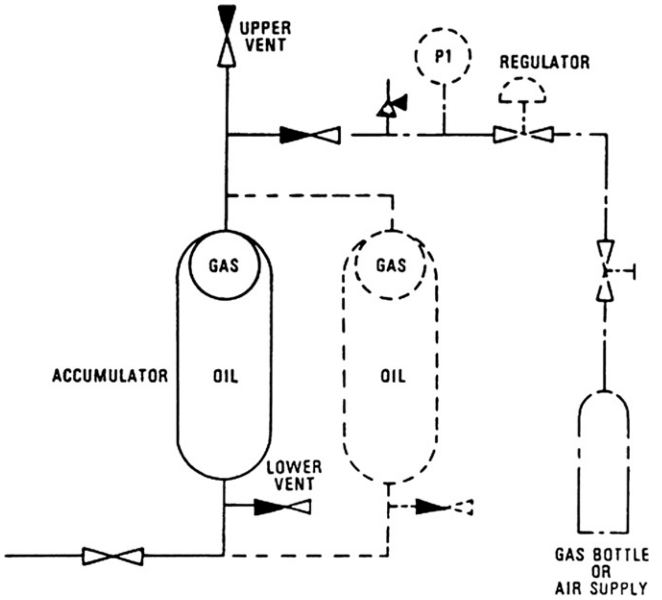


Fig 7.31.2 • Accumulator precharging arrangement (Courtesy of Elliott Co.)

Often an accumulator is improperly sized because of the misconception that its stated size is in fact the capacity contained therein. Actual capacity in any accumulator is equal to the internal volume minus the gas volume over the liquid volume. Typically these values are 50% of the stated capacity or less.

System reliability considerations

Concerning auxiliary system control and instrumentation, a number of reliability considerations are worthy of mention.

Control valve instability

Control valve instability can be the result of many factors. To name a few: improper valve sizing, improper valve actuators, air in hydraulic lines or water in pneumatic lines. Control valve sensing lines should always be supplied with bleeders to ensure that liquid in pneumatic lines or air in hydraulic lines is not present. Presence of these fluids will usually cause instability in the system. Control valve hunting is usually a result of improper controller setting on systems with pneumatic actuators. Attention is drawn to instruction books to ensure that proper settings are maintained. Frequently direct acting control valves exhibit instabilities (hunting on transient system changes). If checks for air prove inconclusive, it is recommended that a snubber device mentioned previously be incorporated in the system to prevent instabilities. Some manufacturers install orifices which sufficiently dampen the system. If systems suddenly act up where problems previously did not exist, any snubber device or orifice installed in the sensor line should be checked immediately for plugging.

Excessive valve stem friction

Control valves should be stroked as frequently as possible to ensure minimum valve stem friction. Excessive valve stem friction can cause control valve instabilities or unit trips.

Given:

- System required flow = 120 GPM
- System pressure at accumulator (at which accumulator effect is desired) = 140 PSIG – 154.7 PSIA (P2)
- Gas precharge pressure (pressure at which accumulator oil flow ceases, assuming system pressure does not fall below this level) = 110 PSIG = 124.7 PSIA (P_1)
- Volume of accumulator = 9 gallons (V_a) (accounts for volume of internal parts)

Determine:

- Amount of oil required
- Number of 10 gallon accumulators required

Amount of oil required:

- System flow per second $= \dfrac{120 \text{ Gal/Min}}{60 \text{ Sec/Min}}$
  $= 2$ Gal/Sec
- Oil required = 3 Sec. $\times$ 2 Gal/Sec
  = 6 Gallons
- Volume of oil entering system for each 10 gallon accumulator

$$V_{oil} = (V_a)\left[1 - \left(\frac{P_1}{P_2}\right)\right]$$

$$= (9 \text{ Gal})\left[1 - \left(\frac{124.7}{154.7}\right)\right]$$

$= 1.75$ Gal. per accumulator

- Number of 10 gallon accumulators required

$$\text{Number of 10 gallon accumulators} = \frac{\text{Oil quantity required}}{\text{Quantity available per accum:}}$$

$$= \frac{6 \text{ Gal:}}{1{:}75 \text{ Gal:}}$$

= 3.42 accumulators required
= four 10 Gal. accumulators

This is a large number of accumulators and is caused by: the conservative setting of P2 and the neglect of the effect of system control valves and partial auxiliary pump flow during pump acceleration. Let's set P2 just (1 PSIG) below the normal header setting and recalculate the number of accumulators required.

$$V_{oil} = (9 = (9 \text{ Gal})\left[1 - \left(\frac{124.7}{175.7}\right)\right]$$

= 2.6 Gal/accumulator
= 3 accumulators required

The above example demonstrates the importance of properly sizing an accumulator.

Fig 7.31.3 • Accumulator sizing

Control valve excessive noise or unit trips

Squealing noises suddenly produced from control valves may indicate valve operation at low travel (C_v) conditions. Valves installed in bypass functions that exhibit this characteristic may be signaling excessive flow to the unit. Remember the concept of control valves being crude flow meters. Observation of valve travel periodically during operation of the unit will indicate any significant flow changes.

Control valve sensing lines

Frequently, plugged or closed control valve sensing lines can be a root cause of auxiliary system problems. If a sensing line that is dead ended is plugged or closed at its source, a bypass valve will not respond to system flow changes and could cause a unit shutdown. Conversely, if a valve sensing line has a bleed orifice back to the reservoir (to ensure proper oil viscosity in low temperature regions), plugging or closing the supply line will cause a bypass valve to fully close rendering it inoperable and may force open the relief valve in a positive displacement pump system.

Valve actuator failure modes

Auxiliary system control valve failure modes should be designed to prevent critical equipment shutdown in case of actuator failure. Operators should observe valve stem travel and pressure gauges to confirm valve actuator condition. In the event of actuator failure, the control valve should be designed for isolation and bypass while on line.

This design will permit valve or actuator change out without shutting down the critical equipment. During control valve on line maintenance, an operator should be constantly present to monitor and modulate the control valve manual bypass as required.

Accumulator considerations

Concerning accumulators, checks should be made when unit is shut down for accumulator bladder condition if supplied with bladders. One area which can cause significant problems in auxiliary systems is accumulators supplied with a continuous charge. That is, charge lines (nitrogen or air) that come directly from a plant utility system. Any rupture of a diaphragm will provide a means for entrance of charge gas directly into the lube system. Most plant utility lines contain pipe scale that could easily plug systems and cause significant critical equipment damage.

In addition, the following reliability factors should be noted (refer to Figure 7.31.2):

- Be sure to install a check valve upstream of the accumulators to ensure all accumulator oil is delivered to the desired components.
- Accumulators should be checked periodically (monthly) for proper pre-charge and bladder condition by isolating and draining the accumulator. Note that the accumulator pre-charge pressure cannot be determined while on line.
- When refilling the accumulators, care must be taken not to suddenly open the supply valve. Best practice is to install an orificed bypass valve to be used for filling the accumulator.
- Best practice is also to install two (2) full size accumulators to ensure that one accumulator is always on line during monthly checks.

This concludes this chapter dealing with system controls and instrumentation. As we proceed, details concerning controls and instrumentation as related to specific systems will be covered.

Best Practice 7.32

Use multiple solenoid control oil dump valves in parallel and in series, with on-line solenoid test facilities for each valve, to ensure optimum reliability of the trip system.

One solenoid valve installed in the control oil system exposes the turbine to a spurious trip if the solenoid should open, and to catastrophic damage if the valve fails to open on command.

Installing two solenoid valves in parallel significantly increases the reliability of the system since only one valve has to open to trip the turbine on command. However if one valve opens when not required, a trip can occur.

Installing two solenoid valves in parallel with another valve in series in each loop (total four (4) solenoid valves) will result in the highest trip system reliability, since this will provide optimum assurance that one valve will open on command in addition to preventing an incidental solenoid valve opening.

In addition, having an on-line solenoid test for each valve every three months will provide optimum assurance that the solenoid trip system will function as necessary.

Lessons Learned

Malfunctioning solenoid valve trip control systems have been responsible for catastrophic turbine failures and spurious trips on low control oil pressure due to solenoid valve failures.

The majority of older control oil trip systems used neither parallel trip valves nor on-line test facilities.

Benchmarks

This best practice has been used since 1997, when research into the causes of turbine trips showed that the highest cause of turbine trips were improper functioning of the solenoid trip valves in single and parallel trip systems.

B.P. 7.32. Supporting Material

Protection

The function of the steam turbine protection system is often confused with the control system, but the two systems are entirely separate. The protection system only operates when any of the control system set point parameters are exceeded and the steam turbine will be damaged if it continues to operate. Figure 7.32.1 defines the typical protection methods.

The protection system monitors steam turbine total train parameters and ensures safety and reliability by the following action:

- Start-up (optional) provides a safe, reliable fully automatic start-up and will shut down the turbine on any abnormality
- Manual shutdown
- Trip valve exerciser allows trip valve stem movement to be confirmed during operation without shutdown
- Rotor overspeed monitors turbine rotor speed and will shut down turbine when maximum allowable speed (trip speed) is attained
- Excessive process variable signal monitors all train process variables and will shut down turbine when maximum value is exceeded

Fig 7.32.1 • Protection

A schematic of a multi-valve, multi-stage turbine protection system is shown in Figure 7.32.2.

This system incorporates a mechanical overspeed device (trip pin) to shut down the turbine on overspeed (10% above maximum continuous speed). Centrifugal force resulting from high shaft speed will force the trip lever which will allow the spring loaded handle to move inward. When this occurs, the port in the handle stem will allow the control oil pressure to drain and drop to zero. The high energy spring in the trip and throttle valve, normally opposed by the control oil pressure will close suddenly (less than 1 second). In this system there are two other means of tripping the turbine (reducing control oil pressure to 0 PSI):

- Manually pushing spring loaded handle
- Solenoid valve opening

The solenoid valve will open on command when any trip parameter set point is exceeded. Solenoid valves are designed to be normally energized to close.

In recent years the industry has required a parallel and series arrangements of solenoid valves to ensure increased steam turbine train reliability. Figure 7.32.3 shows two popular methods of overspeed protection used in the past.

Most speed trip systems now incorporate magnetic speed input signals and two-out-of-three voting for increased reliability. Figure 7.32.4 presents the devices that trip the turbine internally. That is, they directly reduce the control oil pressure causing a trip valve closure without the need of a solenoid valve (external trip method).

Two popular types of steam turbine shutoff valves are shown in Figure 7.32.5. Both types use a high spring force, opposed by control oil pressure during normal operation, to close the valve rapidly on loss of control oil pressure.

It is very important to note that the trip valve will only close if the spring has sufficient force to overcome valve stem friction. Steam system solid build-up, which increases with system pressure (when steam systems are not properly maintained), can prevent the trip valve from closing.

To ensure the trip valve stem is free to move, all trip valves should be manually exercised on-line. The recommended frequency is once per month.

All turbine trip valves should be provided with manual exercisers to allow this feature. Figure 7.32.6 presents facts concerning manually exercising a turbine while on line.

Protection system philosophies have tended to vary geographically with steam turbine vendors, as indicated in Figure 7.32.7.

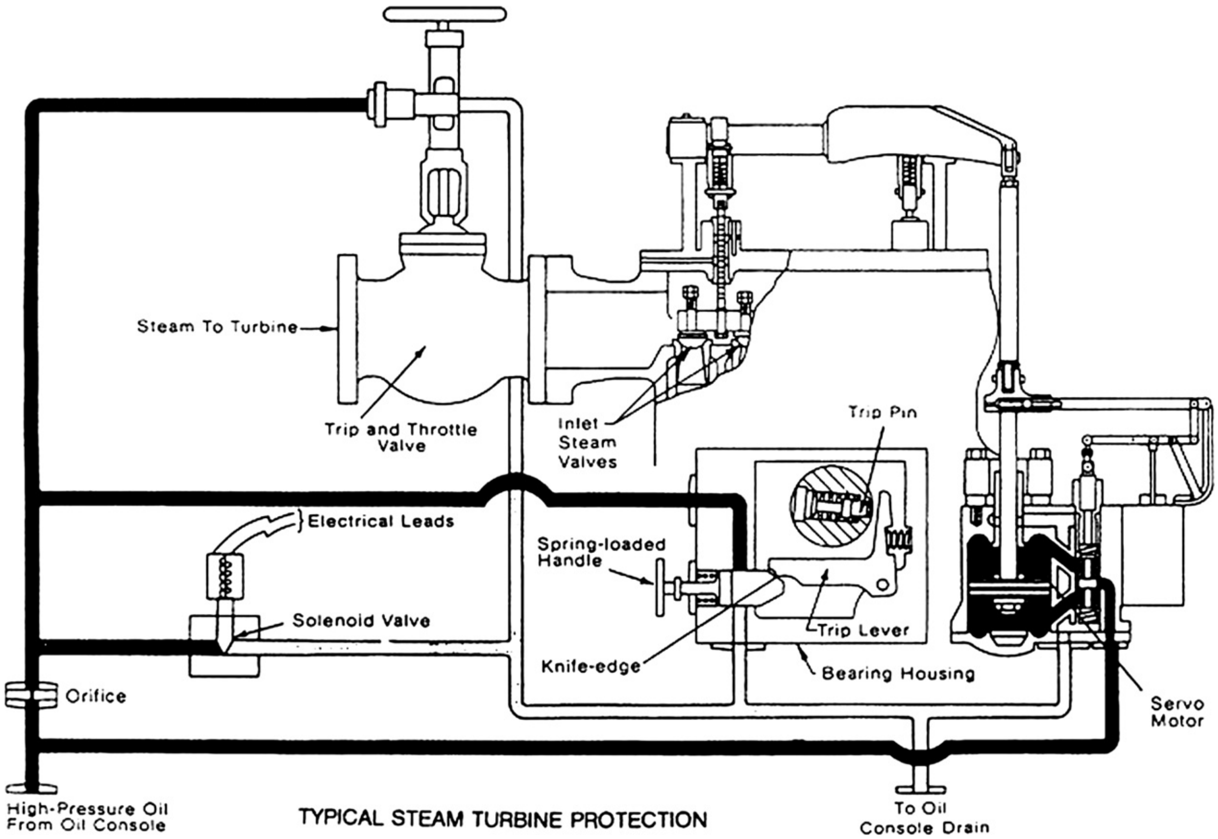


Fig 7.32.2 • Typical steam turbine protection (Courtesy of Elliott Co.)

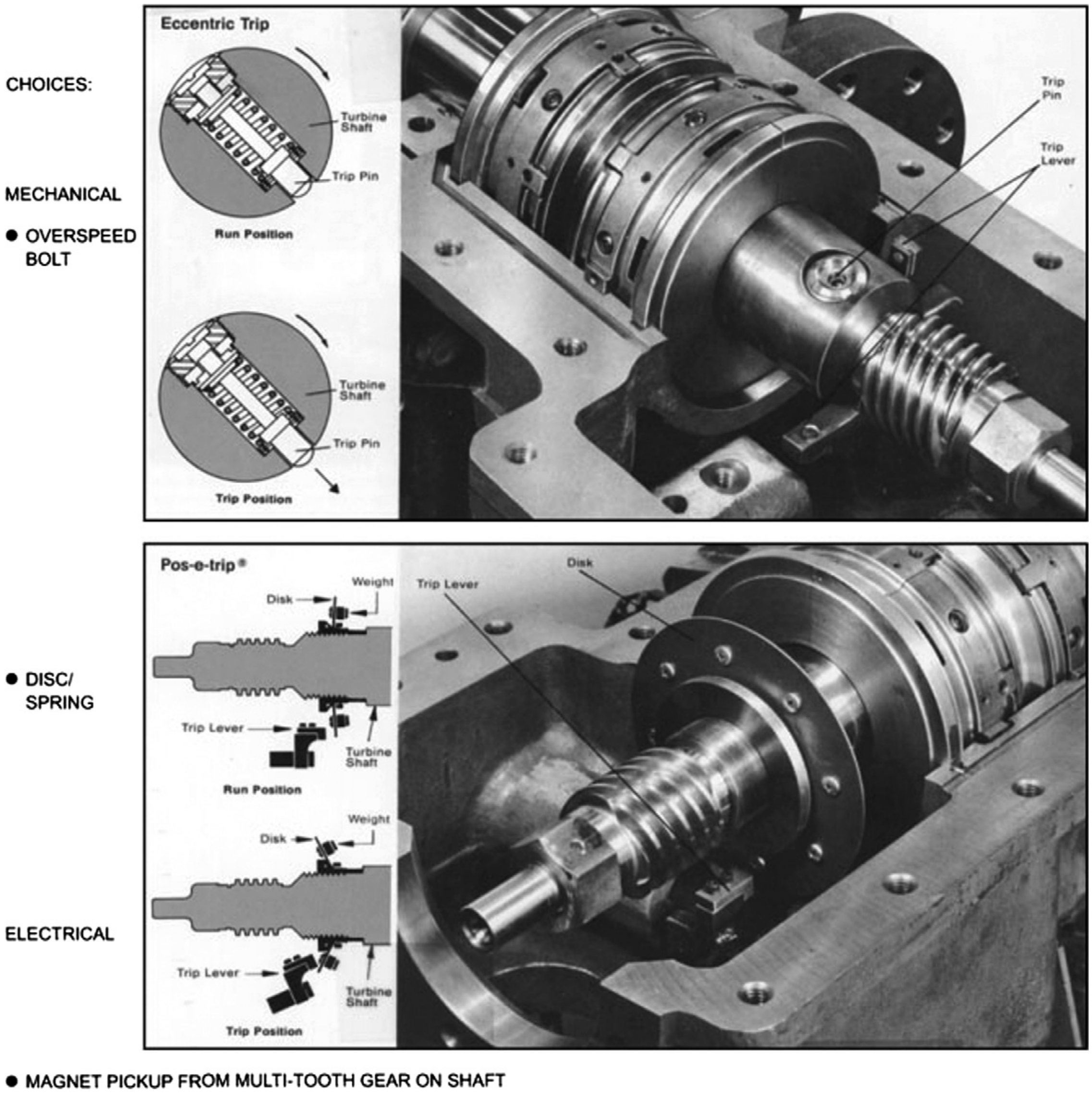


Fig 7.32.3 • Overspeed detection (Courtesy of Elliott Co.)

- Loss of control oil pressure

Spring force automatically overcomes oil force holding valve open (approximate set point 50–65% of normal control oil pressure)

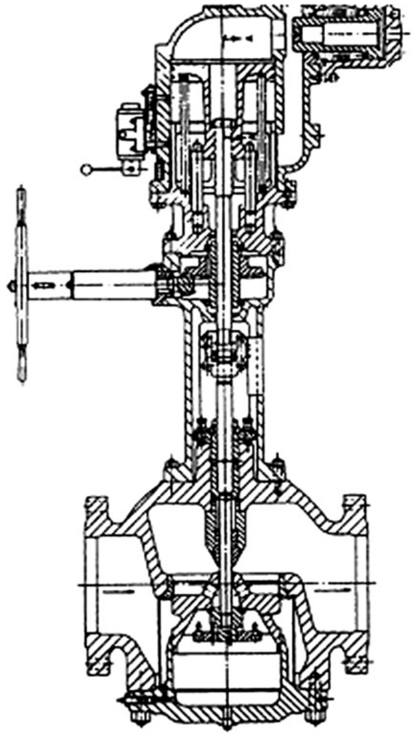
- Manual trip (panic button)

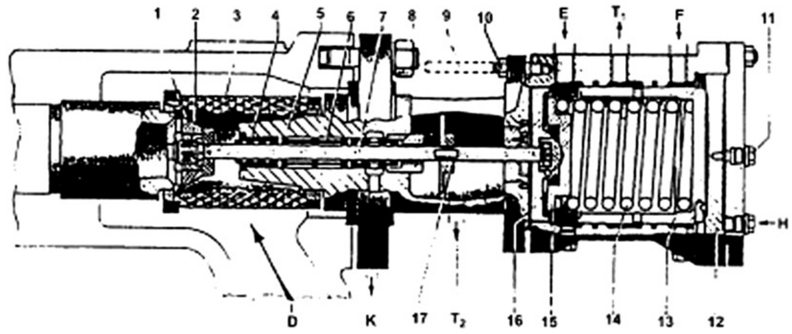
Manually dumps control oil on command

- Optional

Turbine excessive axial movement

Fig 7.32.4 • Internal protection





- FUNCTION RAPIDLY TO CUT OFF STEAM TO TURBINE ON OVERSPEED OR ANY DESIGNATED UPSET. STRONG SPRING FORCE RAPIDLY (ONE (1) SECOND) CLOSES VALVE.

Fig 7.32.5 • Steam turbine shut-off valves. Left: Trip and throttle (Courtesy of Gimple Corp.). Right: Trip (Courtesy of Siemens)

- Trip valve is only as reliable as valve to move
- Should periodically (minimum one per month) exercise valve to ensure movement
- Exercisers will not trip turbine
- If valve does not move, must be remedied immediately

Fig 7.32.6 • On-line manual exercise of trip valve

- Most domestic vendors rely only on trip valve to shut off steam supply. (Throttle valves remain open.)
- European vendors close both trip and automatic throttle valve on trip signal.

Fig 7.32.7 • Protection system philosophies

Best Practice 7.33

Exercise all critical service trip valves (un-spared turbines) to ensure movement a minimum of once a month assuming that the steam system is properly maintained and treated.

All critical service steam turbine trip valves are closed by spring force when the opposing control oil force is suddenly reduced allowing the spring force to close the valve in one second or less.

However, the effective spring force to close the trip valve is a function of the closing spring force minus any trip valve packing friction force.

The trip valve packing is effectively a filter with a differential pressure equal to the inlet steam pressure minus atmospheric pressure. The higher the steam pressure, the greater the driving force to lodge any steam contaminants into the packing and to increase the packing friction force. Also, the higher the steam pressure the more difficult boiler treatment becomes and the greater the possibility of steam contaminants (silica, calcium, etc.).

Especially during the start-up of a new plant with very high pressure steam (over 100 barg), consider the plant experience and boiler treatment equipment in operation to determine if a monthly trip valve exercise is appropriate. FAI has, in some cases, recommended daily exercise in new plants with a gradual weekly incremental increase to one month valve exercises.

Please remember that we are concerned with a trip valve exercise, not a test! If the trip valve fails to move, global safety practices require this un-spared turbine to be immediately shut down using the inlet steam isolation valve with exposure to the corresponding revenue losses until the valve problem is corrected. It is better to err on the conservative side if there is any question about the condition of the steam system.

Finally, confirm with the vendor during the design phase that it is impossible to cause a trip during the trip valve exercise. Then demonstrate to each operating shift that the exercise of the trip valve during commissioning and start-up with the vendor representative present that a trip and/or upset of the steam turbine is not possible during a trip valve exercise.

Lessons Learned

It seems that the majority of operating plants do not exercise the trip valves periodically, causing unscheduled shutdowns (when an exercise was performed without valve stem movement) or catastrophic damage to the turbine and/or driven equipment.

Benchmarks

This best practice of including more frequent exercise intervals for new plants, has been followed since the mid-1980s to achieve maximum critical steam turbine driven train reliability (greater than 99.7%).

B.P. 7.33. Supporting Material

See B.P. 7.32 for supporting material.

Compressor Liquid Seal Best Practices

Liquid seal systems have generally not been used since the acceptance of dry gas seal systems by all end users in the late 1980s. However, most compressor trains designed before 1985 still use liquid seal systems. These systems have many potential weak points that can significantly reduce the reliability of the critical compressor trains that they service. This section therefore presents the best practices used by the author to optimize the safety and reliability of these systems.

Best Practice 7.34

Use flow-through type bushing seals when contact seal surface speeds exceed 100 m/min (300 ft/min), and/or when suction pressures in the compressor are greater than 1,000 kpa (147 psi).

High seal surface speeds and/or low atmospheric bushing flow during the start-up of high suction pressure compressors with non-flow-through bushing seals will result in high bushing seal surface temperatures. These can exceed the atmospheric side seal Babbitt material limit of 150°C (300°F), which will result in premature wear of the seal, which will in turn cause the auxiliary seal oil pump to start with exposure to a seal oil system shutdown.

Oil seal flow through the atmospheric seal is a direct function of the compressor suction pressure.

Compressors with suction pressures of greater than 10 barg (145 psig) are typically started up with lower suction pressures for reasons of low starting torque and/or nitrogen purge pressure system limits in the plant.

If a flow-through option in the seal is not employed, the flow through the atmospheric bushing will be considerably less in proportion to the square root of the starting suction pressure divided by the normal operating pressure.

An oil seal flow-through design will ensure that the proper amount of seal oil cooling is provided at all times during start-up conditions. In addition, seals with high surface speeds will have a higher MTBF if the additional cooling of a flow through design is provided.

Note: Many non-flow-through seal systems can be modified to a flow-through option during a turnaround, with proper advance planning and OEM involvement.

Lessons Learned

Failure to have sufficient oil seal cooling during all phases of operation has resulted in lower than expected oil seal MTBF and shutdowns due to excessive atmospheric bushing wear.

Benchmarks

The best practice has been used during design phases and field auxiliary system audits since 1990 to recommend modification to a flow through bushing seal design. All compressor vendors have the design capability to modify these seals. Of course, since the mid-1990s most new compressor designs use dry gas seal system which eliminates the need for this best practice. See chapter 9 for dry gas seal best practices.

B.P. 7.34. Supporting Material

There are numerous types of fluid seal systems since the types of seals utilized, sealing fluids and sealing pressures vary widely. Regardless of the type of seal used, the function of a critical equipment seal system is as follows: '*To continuously supply clean fluid to each specified seal interface point at the required differential pressure, temperature and flow rate*'. A typical seal system for a centrifugal compressor is shown in Figure 7.34.1.

The system shown is for use with clearance bushing seals. Let's examine this figure by proceeding through the system from the seal oil reservoir through the compressor shaft seal and back through the reservoir. As previously discussed, the concept of sub-systems can be useful here. The seal oil system shown can be divided into four major sub-systems:

A The supply system
B The seal housing system
C The atmospheric drain system
D The seal leakage system

A The supply system

This system consists of the reservoir, pumping units, the exchangers, transfer valves, temperature control valves, and filters. The purpose of this sub-system is to continuously supply clean, cool sealing fluid to the seal interfaces at the correct differential pressure.

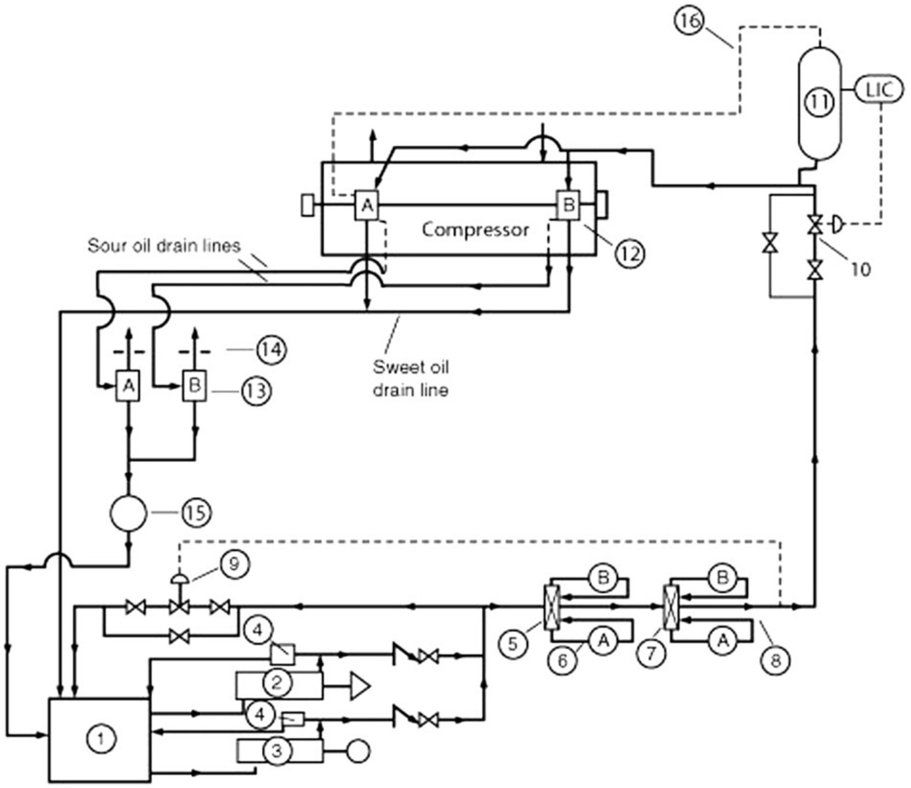
B The seal housing system

This system is comprised of two different seals. A gas side bushing, and an atmospheric bushing. The purpose of the seal housing system is to positively contain the fluid in the compressor and not allow leakage to the atmosphere. The seal fluid is introduced between both seal interfaces, thus constituting a double seal arrangement. Refer to Figure 7.34.2 for a closer examination of the seal.

The purpose of the gas side bushing seal is to constantly contain the reference fluid and minimize sour oil leakage. This bushing can be conceived as an equivalent orifice. This concept is similar to bearings previously discussed, with the exception that the referenced downstream pressure of the gas side bushing can change. In order to ensure a constant flow across this 'orifice', the differential pressure must be kept constant. Therefore, every compressor seal system is designed to maintain a constant differential against the gas side seal. The means of obtaining this objective will be discussed as we proceed.

The other seal in the system is the atmospheric bushing, whose purpose is to minimize the flow of seal liquid at an amount that will remove frictional heat from the seal. This bushing can be conceptualized as a bearing, since the downstream pressure is usually atmospheric pressure. In systems that directly feed into a bearing, the atmospheric bushing downstream pressure will be constant (approximately 138 kPa [20 PSI]). However, the upstream supply pressure will vary with the pressure required by the sealing media in the compressor.

As an example, if a seal system is designed to maintain a constant differential of 34.5 kPa (5 PSI) between the compressor process gas and the seal oil supply to the gas side bushing, the supply pressure with 0 kPA (0 PSIG) process gas pressure, would be 34.5 kPa (5 PSI) to both the gas side bushing and atmospheric bushing. Therefore, gas side bushing and the atmospheric bushing differential would both be equal to 34.5 kPa (5 PSI). If the process gas pressure were increased to 20 psi,



1. Oil reservoir
2. Main screw pump - turbine-driven
3. Aux screw pump - motor-driven
4. Relief valves
5. Transfer valve
6. Oil coolers (A & B)
7. Transfer valve
8. Oil filters (A & B)
9. Back pressure control valve
10. Level control valve
11. Overhead seal oil tank (typically 15 ft above compressor centerline)
12. ISO sleeve seals (also called liquid bushing seals)
13. Seal oil drainers (A & B)
14. Drainer orifice vents (A & B)
15. Degassing tank
16. Reference gas line to seal oil tank

Note: component condition instrumentation and autostarts not shown

Fig 7.34.1 • API 614 lube/seal oil system for ISO-sleeve seals (Courtesy of Elliott Co.)

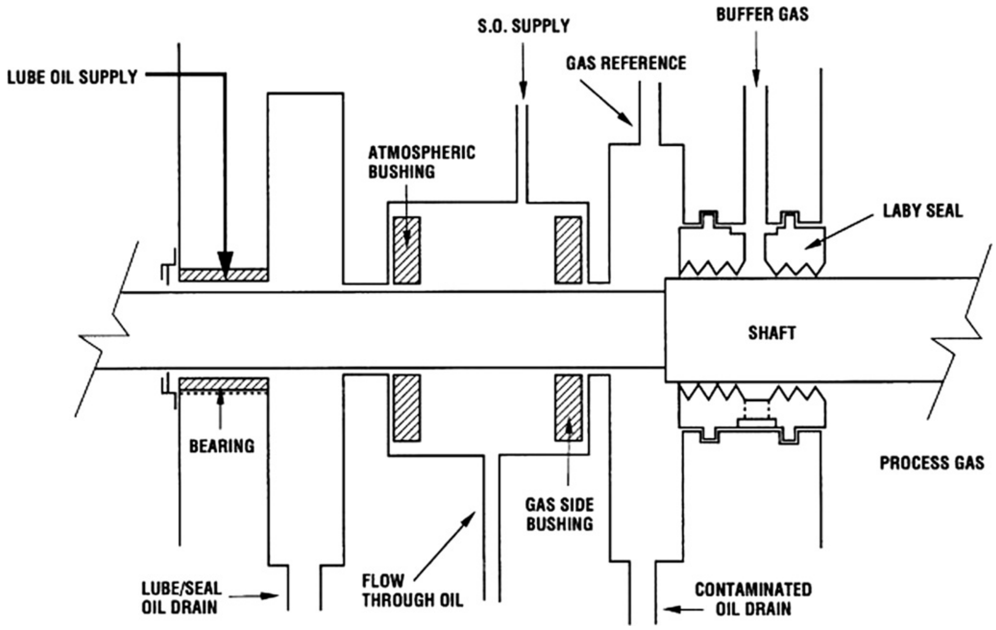


Fig 7.34.2 • Bushing seal schematic (Courtesy of M.E. Crane, Consultant)

the seal oil system would maintain a differential of 34.5 kPa (5 PSI) across the gas side seal, and the supply pressure to the gas side bushing and atmospheric bushing would be 172 kPA (25 PSI). In this case, the differential across the gas side bushing would remain constant at 34.5 kPA (5 PSI), but the atmospheric bushing differential pressure would increase from 34.5 to 172 kPA (5 to 25 PSI). As a result, a primary concern in any seal liquid system is the assurance that the atmospheric bushing receives proper fluid flow under all conditions. After the seal fluid exits the seal chamber, it essentially returns through two additional subsystems.

C The atmospheric draining system

The flow from the atmospheric bushing, if it does not directly enter the bearing system, will return to the seal oil reservoir. In addition, flow from any downstream control valve will also return through the atmospheric drain system to the seal oil reservoir. Both these streams should be gas free since they should not come in contact with the process gas.

D The seal leakage system

The fluid that enters the gas side bushing is controlled to a minimum amount such that it can be either discarded or properly returned to the reservoir after it is degassed. Typically, this amount is limited to less than 77 liters (20 gallons) per day per seal. Since this liquid is in contact with the high speed shaft it is atomized and combines with sealing gas to enter the leakage system. This system consists of:

- An automatic drainer
- A vent system
- Degassing tank (if furnished)

The function of each component is as follows:

The drainer

The drainer contains the oil-gas mixture from the gas side seal. The liquid level under pressure in the drainer is controlled by an internal float or external level control valve to drain oil back to the reservoir or the de-gassing tank, as required.

The vent system

The function of the seal oil drainer vent system is to ensure that all gas side seal oil leakage is directed to the drainer. This is accomplished by referencing the drainer vent to a lower pressure than the pressure present at the gas side seal in the compressor. The drainer vent can be routed back to the compressor suction, suction vessel or a lower pressure source.

The degassing tank

This vessel is usually a heated tank, with ample residence time (72 hours or greater) to sufficiently de-gas all seal oil such that it will be returned to the reservoir and meet the seal oil specification (viscosity, flashpoint, dissolved gasses, etc.). These items will be discussed in detail later.

We will now proceed to discuss each of the major subsystems in detail, defining the function of each such that the total operation of a seal system can be simplified.

The supply system

Referring to definition of a seal oil system, it can be seen that its function is identical to that of a lube oil system, with one exception. The exception is that the seal fluid must be delivered to the seals at the specified differential pressure. Let's examine this requirement further.

Refer again to Figure 7.34.2. Notice that the atmospheric bushing downstream pressure is constant (atmospheric pressure). However, the gas side bushing pressure is referenced to the compressor process pressure. This pressure can and will vary during operation. If it were always constant, the requirement for differential pressure control would not be present in a seal system and would be identical to that of a lubricating system. Another way of visualizing the systems is to understand that the lube system utilizes differential pressure control as well, but the reference pressure (atmospheric pressure) is constant and consequently all control valves need only control lube oil pressure. However seal systems require some means of constant differential pressure control (reference gas pressure to seal oil supply pressure). This objective can be accomplished in many different ways. Referring back to Figure 7.84 it can be seen that the supply system function is identical to that of a lube oil system with the exception that the liquid is referenced to a pressure that can vary and must be controlled to maintain a constant differential between the referenced pressure and the seal system supply pressure. The sizing of the seal oil system components is also identical to that of the lube oil system components. Refer back and observe the heat load and flow required of each seal is determined in a similar way to that of the bearings. Seals are tested at various speeds and a necessary flow is determined to remove the heat of friction under various conditions. The seal oil flow requirements and corresponding heat loads, are then tabulated and pumps exchangers, filters, and control valves are sized accordingly.

The seal oil reservoir is sized in exactly the same way as lube oil reservoir in our previous example. The only major difference between the component sizing of a seal and a bearing is that the seal flows across the atmospheric bushing will change with differential pressures. Every liquid compressor seal incorporates a double seal arrangement. The gas side seal differential is held constant by system design. The atmospheric side seal differential varies with varying seal reference (process) pressure. Therefore, the total flow to the seals will vary with process pressure and must be specified for maximum and minimum values when sizing seal system components. Remembering the concept of an equivalent orifice, a compressor at atmospheric conditions will require significantly less seal oil flow than it will at high pressure 1,380 kPa (200 PSI) conditions. This is true since the differential across the atmospheric seal and liquid flow will increase from a low value to a significantly higher value, while the gas side bushing differential and liquid flow will remain constant provided seal clearances remain constant.

Many seal system problems have been related to insufficient seal oil flow through the atmospheric bushing at low suction pressure conditions. Close attention to the atmospheric drain

cavity temperature is recommended during any off design (low suction pressure condition) operation.

The seal housing system

Regardless of the type, the purpose of any seal is to contain the fluid in the prescribed vessel (pump, compressor, turbine, etc.). Types and designs of seals vary widely. Figure 7.34.3 shows a typical mechanical seal used for a pump.

Since the contained fluid is a liquid, this seal utilizes that fluid to remove the frictional heat of the seal and vaporize the liquid, thus attaining a perceived perfect seal. A small amount of vaporized liquid constantly exits the pump across the seal face. It is a fact that all seals leak. This is the major reason that many pump applications today are required to utilize seal-less pumps to prevent emission of toxic vapors. The following is a discussion of major types of seal combinations used in centrifugal compressor seal applications.

Gas seals

A typical gas seal is shown in Figure 7.34.4. Gas seals are used almost exclusively today (2010) since their supply systems appear to be much simpler than those of a traditional liquid seal system.

Since gas seals utilize the sealed gas or a clean buffer gas, a liquid seal system incorporating pumps, a reservoir and other components, is not required. However, one must remember that the sealing fluid still must be supplied at the proper flow rate, temperature and cleanliness. As a result, a highly efficient, reliable source of filtration, cooling, and supply must be furnished. If the system relies upon inert buffer gas for continued operation, the supply source of the buffer gas must be as reliable as the critical equipment itself. Gas seal configurations vary and will be discussed in detail in the next section. They can take the form of single, tandem (series), or multiple seal systems. The principle of operation is to maintain a fixed minimum clearance between the rotating and non-rotating face of the seal. The seal employed is essentially a contact seal with some type of lifting device to maintain a fixed minimum clearance between the rotating faces. It is essential that the gas between these surfaces be clean since any debris will quickly clog areas and reduce the effectiveness of the lifting devices, consequently resulting in rapid damage to the seal faces.

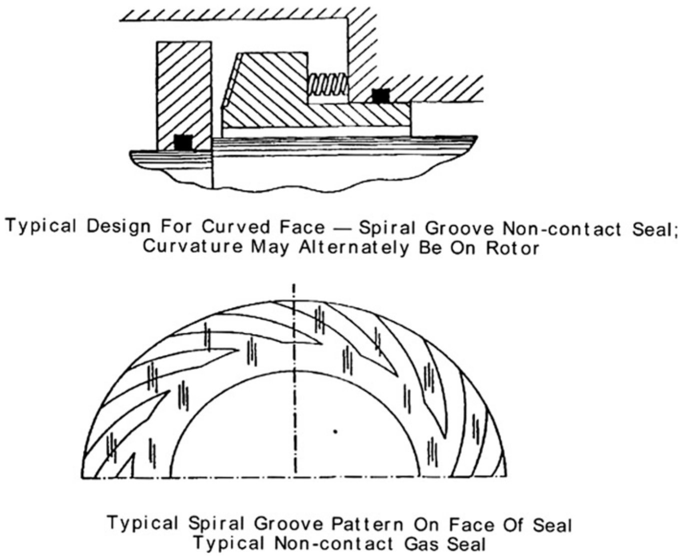


Fig 7.34.4 • Typical gas seal (Courtesy of John Crane Co.)

Liquid seals

Traditionally, the type of seal used in compressor service has been a liquid seal. Since the media that we are sealing against is a gas, a liquid must be introduced that will remove the frictional heat of the seal and ensure proper sealing. Therefore, all compressor liquid seals take the form of a double seal. That is, they are comprised of two seals with the sealing liquid introduced between the sealing faces. Refer to Figure 7.34.5.

To ensure proper lubrication of both the gas side (inboard) and atmospheric side (outboard) seals, the equivalent 'orifices'

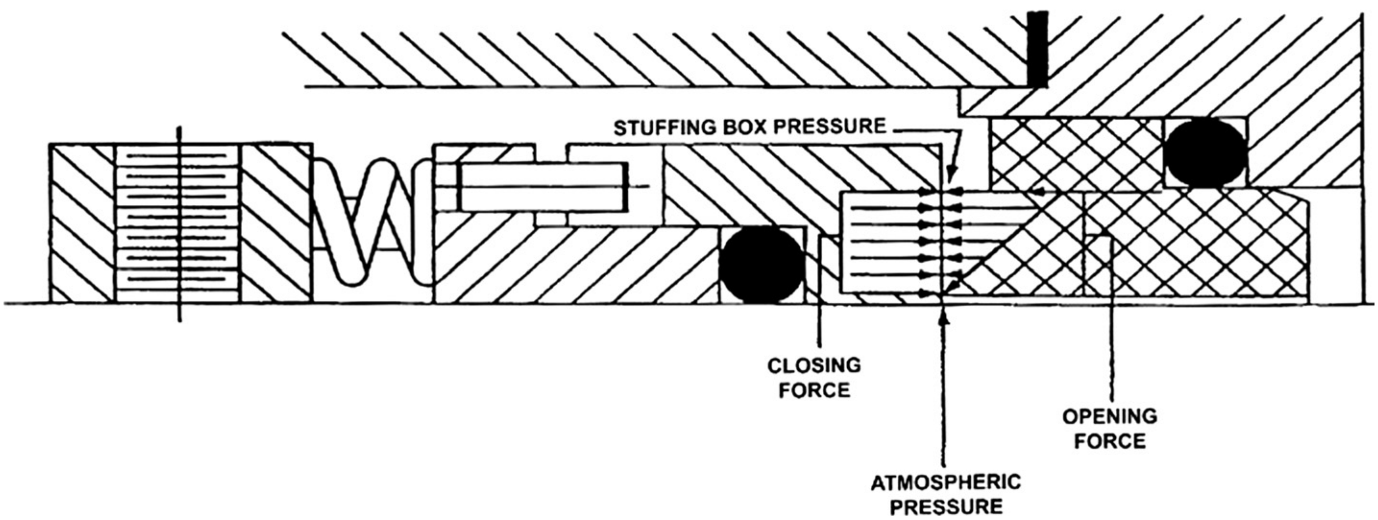


Fig 7.34.3 • Typical pump single mechanical seal

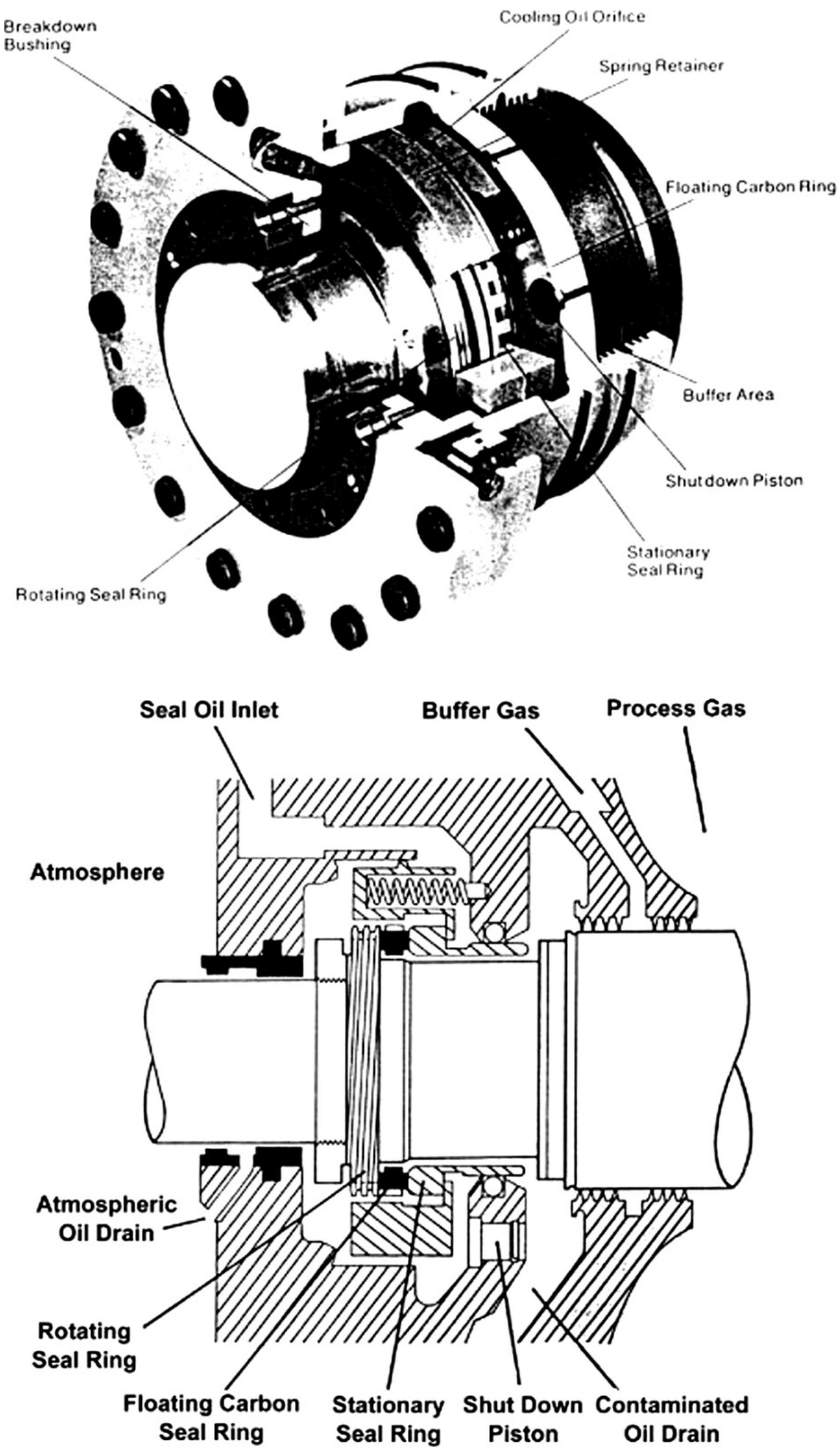


Fig 7.34.5 ● ISO carbon seal (Courtesy of Elliott Co.)

of each seal must be properly designed such that the differential pressure present provides sufficient flow through the seal to remove the heat of friction at the maximum operating speed. The type of gas side seal used in Figure 7.34.5 is a contact seal similar to that used in most pump applications. This seal provides a minimum of leakage 20 to 40 liters (5 to 10 gallons) per day per seal and provides reliable operation (continuous operation for three or more years). As will be discussed below, the specific types of seals used in the double seal (liquid) configuration can vary.

Liquid bushing seals

A liquid bushing seal can be used for either a gas side or an atmospheric side seal application. Most seals utilize a liquid bushing seal for an atmospheric bushing application. A typical bushing seal is shown in Figure 7.34.6.

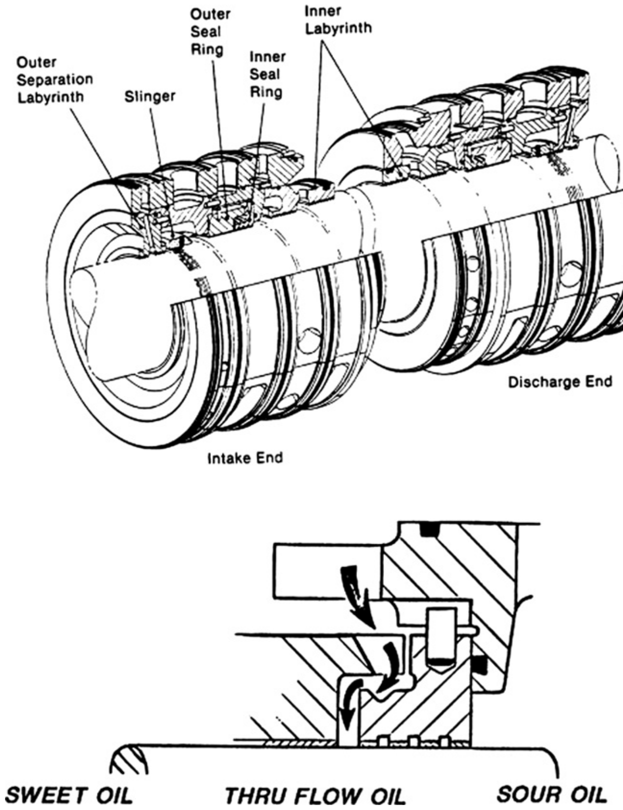


Fig 7.34.6 • Bushing seal – top: oil film seal: bottom: seal oil flow (Courtesy of Dresser-Rand)

The principle of a bushing seal is that of an orifice. That is, a minimum clearance between the shaft and the bushing surface to minimize leakage. The bushing seal is designed such that the clearance is sufficient to remove all the frictional heat at the maximum power loss condition of that bushing with the available fluid differential across the bushing. It is important to realize that while acting as a seal, the bushing must not act as a bearing. That is, it must have degrees of freedom (float) to ensure that it does not support the load of the rotor. Since its configuration is similar to a bearing, if not allowed freedom of movement, it can act as an equipment bearing and result in a significant change to the dynamic characteristics of equipment with potential to cause damage to the critical equipment. In order to achieve the objectives of a bushing seal, clearances are on the order of 0.0005" diametrical clearance per inch of shaft diameter.

Liquid bushing seals are also used for gas side seals, however, their leakage rate will be significantly larger than that of a contact seal since they are essentially an orifice. When used as a gas side bushing, therefore, the system must be designed to minimize the differential across the bushing. As a result, the differential control system utilized must be accurate enough to maintain the specified oil/gas differential under all operating conditions. The typical design differential across a gas side bushing seal is on the order of 35 to 70 kPad (5 to 10 psid). The accurate control of this differential is usually maintained by a level control system.

Referring back to Figure 7.34.6, one can see that functioning of the bushing seal totally depends on maintaining a liquid interface between the seal and shaft surface. Failure to achieve this results in leakage of gas outward through the seal. It must be fully understood that all bushing seals must continuously maintain this liquid interface to ensure proper sealing. All systems incorporating gas side bushing seals must have the seal system in operation whenever pressurized gas is present inside the compressor case. If a liquid interface is not maintained, gas will migrate across the atmospheric bushing seal and proceed through the system returning back to the supply system. There have been cases in such system designs where failures to operate the seal system when the compressor is pressurized have resulted in effectively turning the gas side bushing into a filter for the entire process gas system! This resulted in the supply side of the seal oil system being filled with extensive debris that required lengthy flushing and system cleaning operations prior to putting the unit back into service. Remember, any system incorporating a gas side bushing seal must be designed such that the entrance of process gas into the supply system is prohibited at all time. This can be accomplished by either:

- Continuous buffer gas supply
- A check valve installed as close as possible to seals in the seal oil supply header
- Rapid venting and isolation of the compressor case on seal system failure

In the second and third cases above, supply seal oil piping must be thoroughly checked for debris prior to re-start of the compressor. It is our experience that many bushing seal system problems have resulted from improper attention to the above facts.

Contact seals

Figure 7.34.7 shows a typical compressor contact seal. As mentioned, these seals are similar in design to pump seals. In order to remove the heat of friction for this type of seal, a sufficient differential pressure above the reference gas must be maintained. Typical differentials for contact seals vary between 240 and 345 kPad (35 and 50 psid). Leakage rates with a properly installed seal can be maintained between five to ten gallons per day per seal.

Shaft speed is a limitation in the use of contact seals. Since the contact seal operates on a surface perpendicular to the axis of rotation, the rubbing speed of the seal surface is critical. As a result, contact seals are speed limited. Typical maximum speeds are approximately 12,000 revolutions per minute. Above those speeds, bushing seals are used, since the sealing surface is maintained at a lower diameter and correspondingly lower rubbing speed. The maximum limit of differential pressure across contact seals is controlled by the materials of construction and is approximately 1,380 kPad (200 psid). As a result, contact seals are usually used for gas side seal applications. They are very seldom utilized for atmospheric seal applications since they are differential pressure limited.

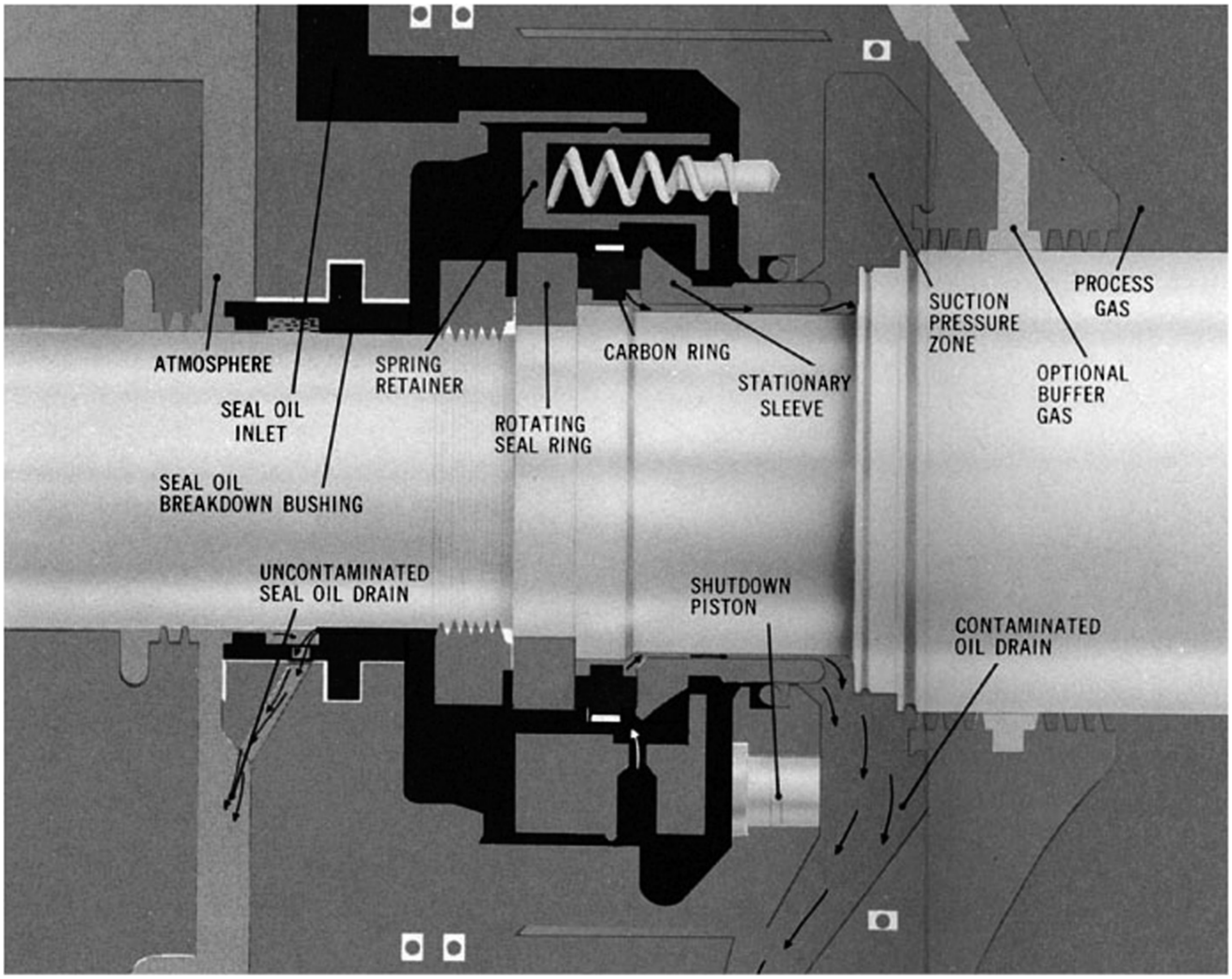


Fig 7.34.7 • Compressor contact seal (Courtesy of Elliott Co.)

Since the differential pressure required across the seal face is relatively high as compared to a bushing seal, contact seals utilize differential pressure control as opposed to level control for most bushing seals. This fact will be discussed in the next section.

Restricted bushing seals

The last type of seal to be discussed is a restricted bushing type seal. This type of seal is shown in Figure 7.34.8.

This particular type of restricted bushing seal utilizes a small pumping ring in the opposite direction of bushing liquid flow to compensate for the relatively large leakage experienced with bushing seals by introducing an opposing pumping flow in the opposite direction. Seals of this type can be designed for practically zero flow leakages. However, it must be pointed out that in variable speed applications, the pumping capability of the trapped seal ring must be calculated for both minimum and maximum speeds. Failure to do so can result in the actual pumping of gas from the compressor into the sealing system. It is recommended that such seals be designed to leak a small amount at maximum operating speed. Any retrofits of equipment employing this type of seal should be investigated when higher operating speeds are anticipated. A restricted bushing seal is used exclusively for gas side service.

In summary, the basic types of liquid seals used for compressor applications can be either: open bushing types, contact types, or restricted bushing types. Contact types are used primarily on the gas side. Liquid bushing types are used on either the gas or atmospheric side. Restricted bushing types are used exclusively on the gas side.

We will now investigate various seal system designs using various seal combinations employing the types of seals that have been discussed in this section.

Seal supply systems

As can be seen from the previous discussion, the type of seal system will depend on the type of seal utilized. We will now examine five different types of seal system, each utilizing a different type of main compressor shaft seal system. As we proceed through each type, the function of each system will become clear.

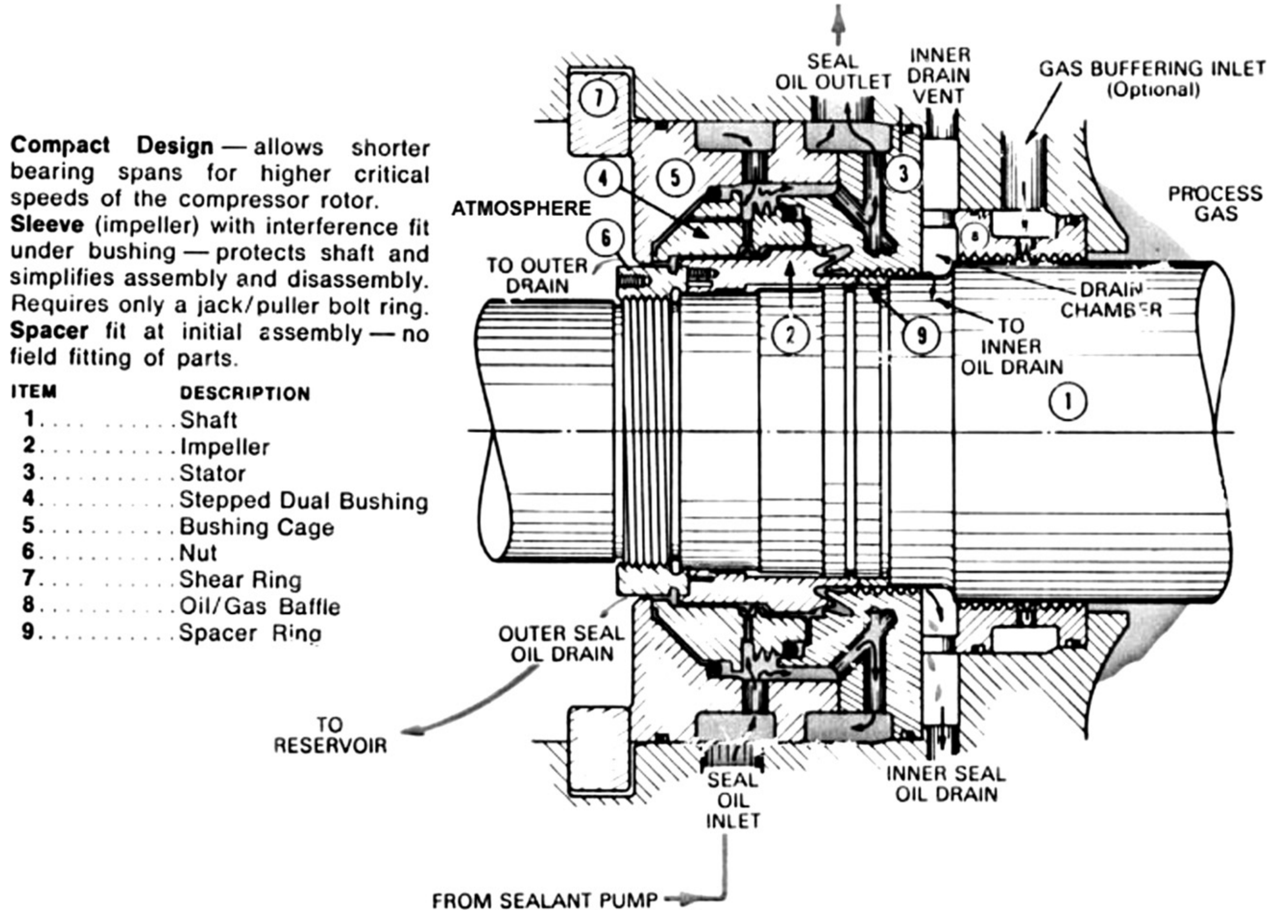


Fig 7.34.9 • Compressor shaft seal (Courtesy of IMO Industries)

Example 1: Contact type gas side seal – bushing type atmospheric side seal with cooling flow ('flow-through type')

This system incorporates a contact seal on the gas side and a bushing seal on the atmospheric side of each end of the compressor. The inlet pressure of the seal fluid on each end is referenced to the suction pressure of the compressor. It should be noted that some applications employ different reference pressures on each end of the compressor. The reference pressure should be taken off the balance drum end, or high pressure end of the compressor, to ensure that the oil to gas differential pressure is always at a minimum acceptable value. Therefore, the low pressure end may experience a slightly higher oil to gas differential then the reference end of the seal. Refer to Figure 7.34.9.

Proceeding through the seal, the seal oil supply, which is referenced to the gas reference pressure, enters the seal chamber. The differential across the gas side contact seal is maintained by a differential pressure control valve located downstream of the seal. Seal oil flows in three separate directions:

- Through the seal chamber (cooling flow)
- Through the gas side contact seal: 38–77 liters/day (10–20 gallons/day)
- Through the atmospheric seal

Let's examine the variants of flows across the equivalent orifice of each portion of this configuration.

The gas side contact seal will experience a constant flow, that for purposes of discussion can be assumed to be zero gallons per minute (since the maximum flow rate will usually be on the order of 38 liters [10 gallons] per day).

The atmospheric side bushing seal flow will vary based upon the referenced gas pressure. At low suction pressure conditions, this flow will be significantly less than it will be under high pressure conditions. The seal system design must consider the maximum reference pressure to be experienced in the compressor case to ensure that sufficient seal oil flow is available at maximum pressure conditions.

The seal chamber through flow in this seal design is used to remove any excess frictional heat of the seals and is regulated by the downstream control valve. As an example, let us assume the following values were calculated for this specific seal application.

1. *Gas side seal flow* = 0
2. *Atmospheric side seal flow*

Reference pressure = 0 kPa (PSIG)
Seal flow = 19 LPM (5 GPM)
Reference pressure = 1,380 kPa (200 PSIG)
Seal flow = 46 LPM (12 GPM)

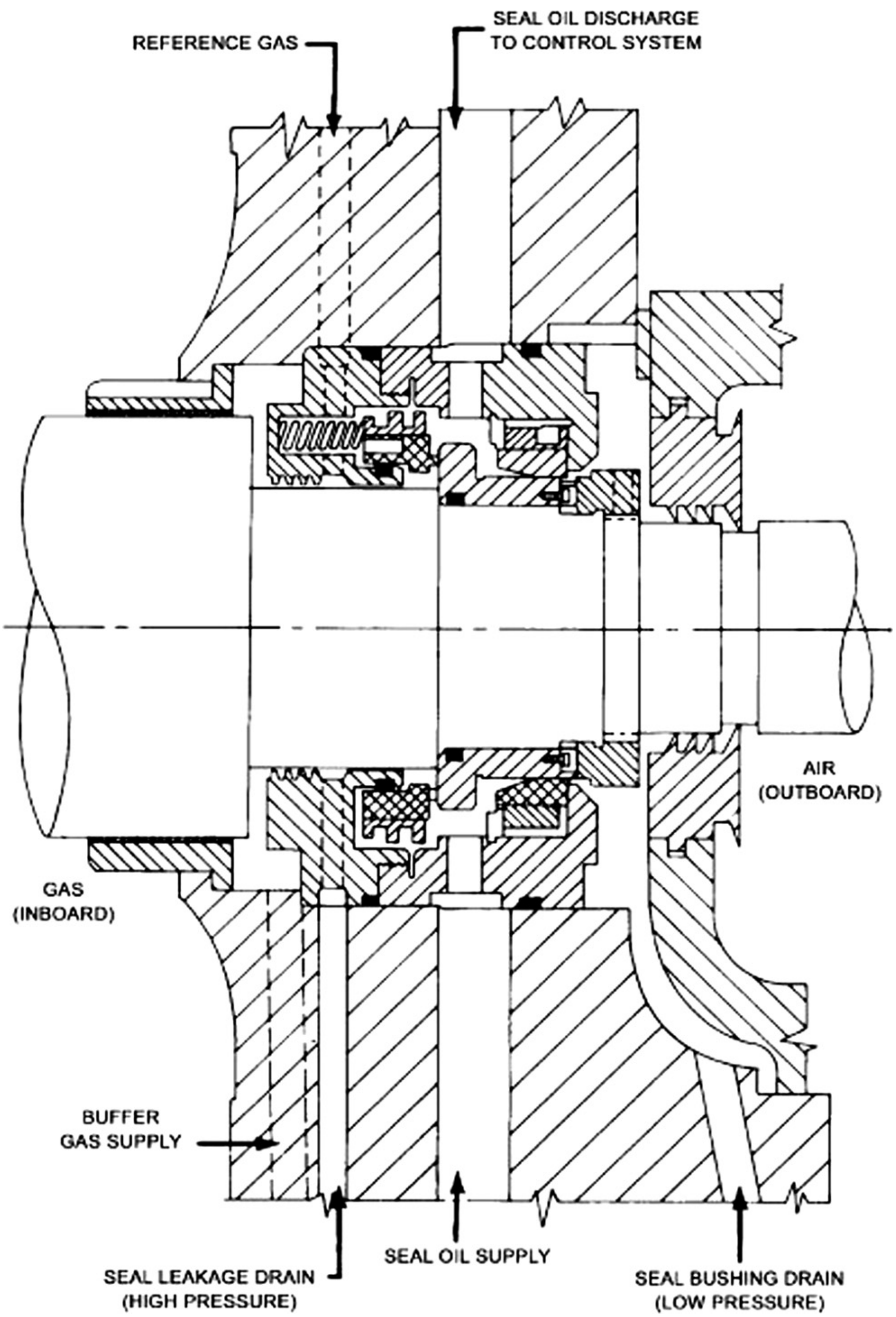


Fig 7.34.9 • Compressor shaft seal
(Courtesy of IMO Industries)

3. *Flow-through flow*

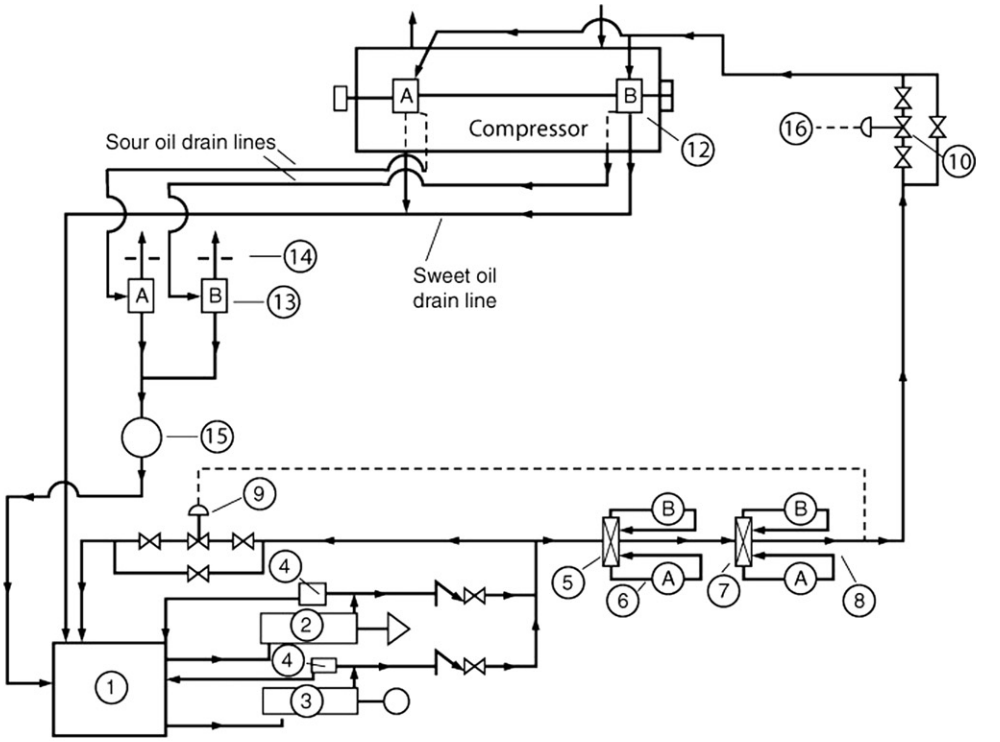
Minimum = 12 LPM (3 GPM) (occurring at high ATM bushing flow = 46 LPM [12 GPM])
Maximum = 46 LPM (12 GPM) (occurring at low ATM bushing flow = 12 LPM [3 GPM])

4. *Seal oil supply flow* in both cases = 58 LPM (15 GPM)

As shown in the previous example, the required seal oil supply at maximum operating speed required to remove frictional heat is 58 LPM (15 GPM). At start-up, low suction pressure conditions, the control valve must open to allow an additional ten gallons a minute flow through to the seal chamber. At maximum operating pressure, however, the valve only passes a flow of three gallons a minute since 46 LPM (12 GPM) exit through the atmospheric bushing. This type of system is less sensitive to low suction pressure operation since flow through oil will remove frictional heat around the atmospheric bushing.

Example 2: Contact type gas side seal – bushing type atmospheric seal with orificed through flow

The only difference between this type of system and the previous one is that the back pressure is maintained constant by a permanently installed through flow orifice. As a result, the differential pressure control valve is installed on the inlet side of



1. Oil reservoir
2. Main screw pump - turbine-driven
3. Aux screw pump - motor-driven
4. Relief valves
5. Transfer valve
6. Oil coolers (A & B)
7. Transfer valve
8. Oil filters (A & B)
9. Back pressure control valve
10. Differential pressure control valve
11. Overhead seal oil tank (typically 15 ft above compressor centerline)
12. ISO sleeve seals (also called liquid bushing seals)
13. Seal oil drainers (A & B)
14. Drainer orifice vents (A & B)
15. Degassing tank
16. Reference gas line

Note: component condition instrumention and autostarts not shown

Fig 7.34.10 • API 614 Lube-seal oil system for ISO-carbon seals (Courtesy of Elliott Company)

the system. The process gas reference is still the same as before, that is, to the highest pressure side of the compressor. Figure 7.34.10 shows this type of system.

Let us examine the previous example case for this system and observe the differences.

1. Gas side seal flow = 0
2. Atmospheric side seal flow
   Reference pressure = 0 kPa (PSIG)
   Seal flow = 19 LPM (5 GPM)
   Reference pressure = 1,380 kPa (200 PSIG)
   Seal flow = 46 LPM (12 GPM)
3. Flow through flow (orifice)
   Minimum = 2 LPM (0.5 GPM)
   Maximum = 12 LPM (3 GPM)

As can be seen, this system is more susceptible to high temperature atmospheric bushing conditions at low suction pressures and must be observed during such operation to ensure integrity of the atmospheric bushing. In this system, the control valve will sense supply oil pressure to the seal chamber and control a constant set differential, approximately 240 kPad (35 psid), between the reference gas pressure and the supply pressure. If continued low pressure operation is anticipated with

such a system, consideration should be given to a means of changing the minimum flow and maximum flow orifice for various operation points. Externally piped bypass orifices could be arranged such that a bypass line with a large orifice for minimum suction pressure conditions could be installed and opened during this operation. It is important to note, however, that the entire supply system must be designed for this flow condition and control valve must be sized properly to ensure proper flow at this condition. In addition, the low pressure bypass line must be completely closed during normal high pressure operation.

Example 3: Bushing gas side seal – bushing atmospheric side seal with no flow through provision

Figure 7.34.1 shows this type of seal system. In this type of system, the differential control valve becomes a level control valve sensing differential from the level in an overhead tank and is positioned upstream of the unit. Both bushings can be easily conceived as equivalent orifices. The gas side bushing flow will remain constant regardless of differential. The atmospheric bushing flow will vary according to seal chamber to atmospheric pressure differential.

Therefore, the atmospheric bushing must be designed to pass a minimum flow at minimum pressure conditions that will remove frictional heat and thus prevent overheating and damage to the seal. Since a gas side bushing seal is utilized, a minimum differential across this orifice must be continuously maintained.

Utilizing the concept of head, the control of differential pressure across the inner seal is maintained by a column of liquid.

As an example, if the required gas side seal differential of oil to gas is 34.5 kPad (5 psid) by the liquid head equation:

$$\frac{\text{Head} = 0.102\ (34.5)\ \text{kPad})}{.85} = 4.1\ \text{meters}$$

$$\text{Head} = \frac{2.311 \times 5\ \text{psid}}{.85} = 13.6\ \text{ft.}$$

Therefore maintaining a liquid level of 4.1 m (13.6 ft) above the seal while referencing process gas pressure will ensure a continuous 34.5 kPad (5 psid) gas side bushing differential. In this configuration, the control valve which senses its signal from the level transmitter, will be sized to continuously supply the required flow to maintain a constant level in the overhead tack. As an example, consider the following system changes from start-up to normal operation.

In this example a change from the start-up to operating condition will increase gas reference pressure on the liquid level in the overhead tank and would tend to push the level downward. Any movement of the level in the tank will result in an increasing signal to the level control valve to open, thus increasing the pressure (assuming a positive displacement pump) to the overhead tank and reestablishing the preset level.

In the above example, at 1,380 kPa (200 psi) reference pressure, the bypass valve would close considerably. To increase the pressure supply of the seal oil from 34.5 to 1,410 kPa (5 to 205 psi), the difference of bypass flow through the valve 27 LPM (7 GPM) is equal to the increased flow through the atmospheric bushing at this higher differential pressure condition. Utilizing the concept of equivalent orifices, it can be seen that the additional differential pressure across the atmospheric bushing orifice is compensated for by reducing the effective orifice area of the bypass control valve. This is accomplished by sensing the level in the head tank and maintaining it at a constant value by opening the seal oil supply valve.

As in the case of the orificed through flow example above, this configuration is susceptible to high atmospheric bushing temperatures at low suction pressures and must be monitored during this condition. Repeated high temperatures during low suction pressure conditions should give consideration to resizing of atmospheric bushing clearances during the next available turnaround. The original equipment manufacturer should be consulted to ensure correct bushing sizing and supply system capability.

Item	Start-up condition	Normal operation
Compressor suction pressure	0 kPa (PSIG)	1,380 kPa (200 PSIG)
Overhead tank reference pressure	0 kPa (PSIG)	1,380 kPa (200 PSIG)
Gas side seal bushing flow	0 LPM (GPM)	0 LPM (GPM)
Atmospheric side seal bushing flow	19 LPM (5 GPM)	46 LPM (12 GPM)
Seal pump flow	77 LPM (20 GPM)	77 LPM (20 GPM)
Bypass valve flow	58 LPM (15 GPM)	31 LPM (8 GPM)

Fig 7.34.11 • Seal System Flow

Example 4: Gas side bushing seal – atmospheric side bushing seal with through flow design

Refer to Figure 7.34.12. The only difference between this system and the previous example is that a through-flow option is added to allow sufficient flow through the system during changing pressure conditions. The bypass valve in the previous system is replaced in this system by a level control valve referenced from a head tank level transmitter, and is installed downstream of the seal chamber. This system functions in exactly the same way as the system in Example 1. The only difference being that a level control valve in this example replaces the differential control valve in the previous example. Both valves have the same function, that is, to control the differential in the seal chamber between the seal oil supply and the referenced gas pressure. A level control valve is utilized in this example, however, since a bushing seal requires a significantly lower differential between the seal oil supply pressure and the gas reference pressure.

Consider the following example. Assume that a differential control valve would be used as opposed to a level control valve for the system in Figure 7.34.12. For the start-up case, the differential control valve would have to maintain a differential of 34.5 kPa (5 psi) over the reference gas. When the reference gas pressure were 0, the oil upstream pressure to the valve would be approximately 34.5 kPa (5 psi). For the operating case,

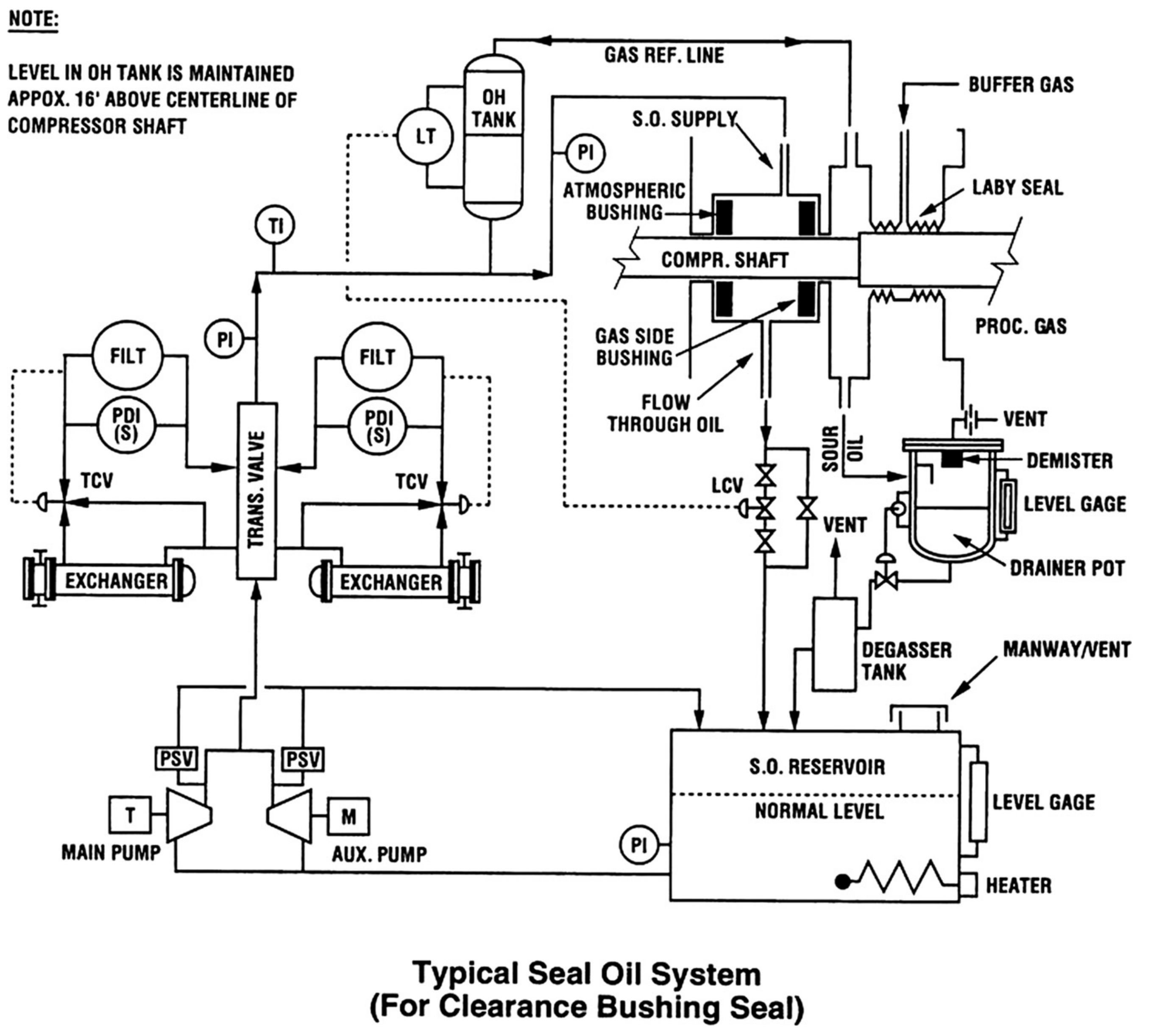


Fig 7.34.12 • Typical seal oil system (Courtesy of M.E. Crane, Consultant)

maintaining the same 34.5 kPa (5 psi) differential, the upstream pressure across the valve would be approximately 1,410 kPa (205 psi) instead of 34.5 kPa (5 psi). Consequently, the valve position would change significantly, but still would have to control the differential accurately to maintain 34.5 kPa (5 psi). Reduction of this pressure in any amount below 34.5 kPa (5 psi) could result in instantaneous bushing failure. However, if a level control valve were installed, the accuracy of the valve would be measured in mm (inches) of oil instead of kPa (psi). Any level control system could control the level within 50 mm (2"), which would be only a 0.4 kPad (0.06 psid) variation in pressure differential!

This example shows that the accurate means of controlling differential pressure for systems requiring control of small differential values, is to use level instead of differential control. This system would be designed such that the combination of the atmospheric flow and the through flow through the seal would be equal to the flow from the pump.

Example 5: Trapped bushing gas side seal: atmospheric side bushing seal with flowthrough design

This system would follow exactly the same design as the system described in Example 4. The only difference would be in the amount of flow registered in the seal oil drainer. A trapped bushing system is designed to minimize seal oil drainer pot leakage. Typical values can be less than 19 l/day (5 gal/day).

Seal supply system summary

All of the above examples have dealt with a system incorporating one seal assembly. It must be understood that most systems utilize two or more seal system assemblies. Typical multi-stage compressors contain two seal assemblies per compressor body and many applications contain upwards of three compressor

bodies in series, or six seal assemblies. Usually each compressor body is maintained at the suction pressure to that body, therefore three discreet seal pressure levels would be required and three differential pressure systems would be utilized. The concepts discussed in this section follow through regardless of the amount of seals in the system. Sometimes, the entire train, that is, all the seals referenced to the same pressure. In this case, one differential seal system could be used across all seals.

In conclusion, remembering the concept of an orifice will help in understanding the operation of these systems. Remember, the gas side bushing is essentially zero flow, the atmospheric side bushing flow varies with changing differential across the seal and any seal chamber through flow will change either as a result of differential across a fixed orifice or the repositioning of the control valve.

Seal liquid leakage system

This seal system sub-system's function is to collect all of the leakage from the gas side seal and return it to the seal reservoir at specified seal fluid conditions. Depending upon the gas condition in the case, this objective may or may not be possible. If the gas being compressed has a tendency to change the specification of the seal oil to off specification conditions, one of two possibilities remain:

- *Introduce a clean buffer gas* between the seal to ensure proper oil conditions
- Dispose of the seal oil leakage

In most cases, the first alternative is utilized. Once the seal oil is in the drainer, a combination of oil and gas are present. A vent may be installed in the drainer pot to remove some of the gas, or a degassing tank can be incorporated.

This concludes the overview section of seal oil systems. As can be seen from the above discussion, it is evident that the design of a seal oil system follows closely to that of a lube system. The major difference is that the downstream reference pressure of the components (seal) varies, whereas in the case of a lube system it does not. In addition, the collection of the expensive seal oil is required in most cases and a downstream collector, or drainer system, must be utilized. Other than these two exceptions, the design of the seal system is very similar to that of a lube system and the same concepts apply in both cases.

Best Practice 7.35

Do not use dynamic gas side (inner side) bushing seals unless absolutely necessary since these types of seals typically have lower MTBFs than higher seal oil leakage straight bushing seals and can cause gas leakage to the atmosphere through the atmospheric bushing (outer side).

Dynamic gas side seals employ devices to act as reverse flow pumps to minimize inner seal oil leakage to the seal oil drainers.

These devices are speed and clearance sensitive and operate at higher temperatures than straight through bushing seals. This reduces the life of these seals, to less than 12 months in some cases.

In many cases the reverse pumping action will bring gas into the port between the inner and outer (atmospheric) bushing seal, resulting in gas entering the outer seal and being transported outside the compressor.

Vendors and non-OEMs offer dynamic inner bushing seals as modifications to reduce seal oil leakage into the drainers and/or the compressor.

It is recommended that prior to accepting a dynamic inner seal modification, end users perform a life cycle cost analysis. This should compare the savings of the reduced requirement for seal oil to the drainers with the cost of one shutdown for a seal change-out during a run (4–5 years). A typical seal change will take a minimum of five days resulting in lost revenue for critical service compressors.

Lessons Learned

The MTBF of dynamic bushing inner oil seals is less than straight though bushing seals and the cost of unscheduled shutdowns during a continuous critical service compressor run will usually exceed the savings gained by having less oil leakage to the drainers.

Note that if the reason for change-out to a dynamic bushing is the entry of oil into the compressor, especially in refrigeration applications where entrained oil in the process will affect chiller cooling effectiveness, there is another modification that can be made to the seal oil system to correct this problem (and that option is not to use dry gas seals). See B.P. 7.42.

Benchmarks

This practice has been used since the mid-1980s to maximize the MTBF of oil bushing seals from all vendors. The use of straight through or static inner bushing seals has resulted in oil seal MTBFs in excess of 120 months and control of inner seal oil leakage to manageable values where clean buffer gas and proper system modifications are implemented.

B.P. 7.35. Supporting Material

See B.P: 7.34 for supporting material.

Best Practice 7.36

Use clean, sweet, buffer gas whenever process gas is sour and/or can contain debris to optimize oil seal MTBF (in excess of 120 months).

The entrance of sour gas and/or gas with debris into the oil seal system will expose the seal oil system to the following issues that can reduce seal MTBF, cause unscheduled shutdowns and pose a safety hazard (gas release to the plant environment):

- Mechanical or iso carbon inner seal hang up and high seal oil leakage
- Blockage of internal seal oil drainer orifices causing ingestion of seal oil into the compressor
- Contamination of overhead seal oil tanks with oil sludge, eventually blocking internal seal oil ports and exposing the seals to low or zero seal oil to gas differential and possible gas release to the plant

The use of a sweet and clean buffer gas will positively eliminate the above problems to optimize the life of the seals and the reliability of the seal system and corresponding unit (99.7% +).

Lessons Learned

Failure to design for and use a sweet and clean buffer gas in oil seal systems has resulted in low seal MTBFs (less than 12 months) and has caused gas releases in plants that have caused fires and personnel harm.

I have known of a case of loss of life that was caused by an internal valve oil drainer remaining open in sour gas service, which released H_2S to the plant while an operator was checking seal oil leakage from this drainer.

Benchmarks

This best practice has been used and recommended for new projects and plant modifications, since the above incident occurred in the late 1970s, to ensure plant safety and optimize seal oil system reliability.

B.P. 7.36. Supporting Material

The function of any buffer fluid system is to continuously supply clean buffer fluid to each specified point at the required differential pressure, temperature and flow rate. There are different types of fluid buffer systems:

A. Compressor buffer systems (Figure 7.36.1)
B. Steam buffer systems (Figure 7.36.2)
C. Liquid buffer systems or pump flush systems (Figure 7.36.3)

In this section, we will concentrate only on compressor buffer systems.

Compressor buffer gas system overview

Types of system

There are various types of compressor buffer gas systems for different applications. Refer to Figure 7.36.4 for a listing of system requirements.

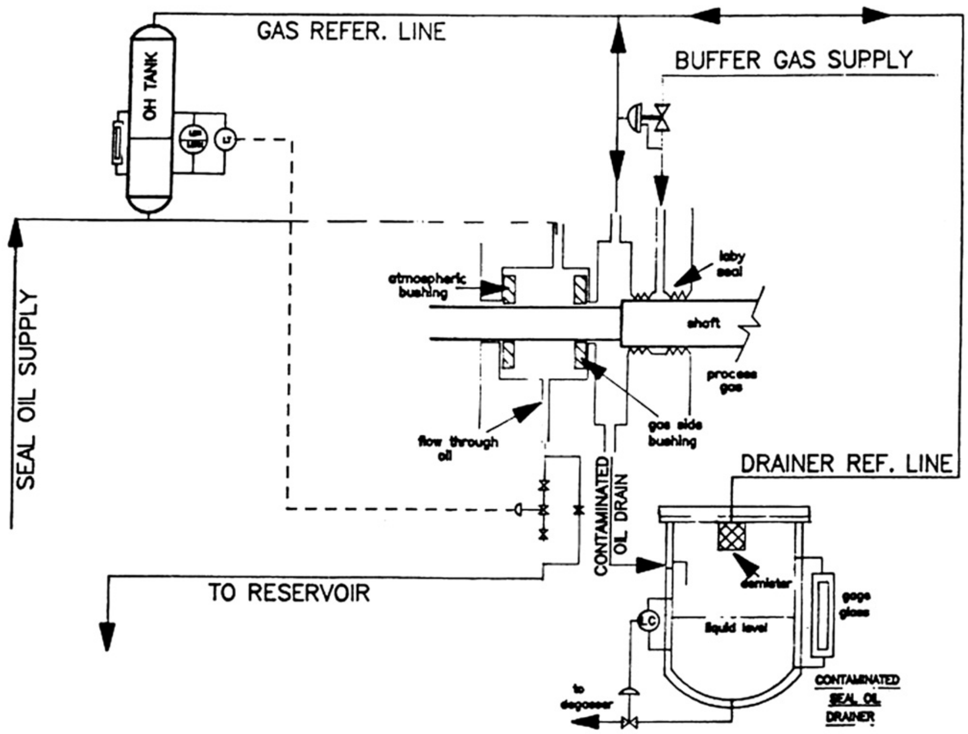


Fig 7.36.1 • Compressor buffer system (Courtesy of M.E. Crane, Consultant)

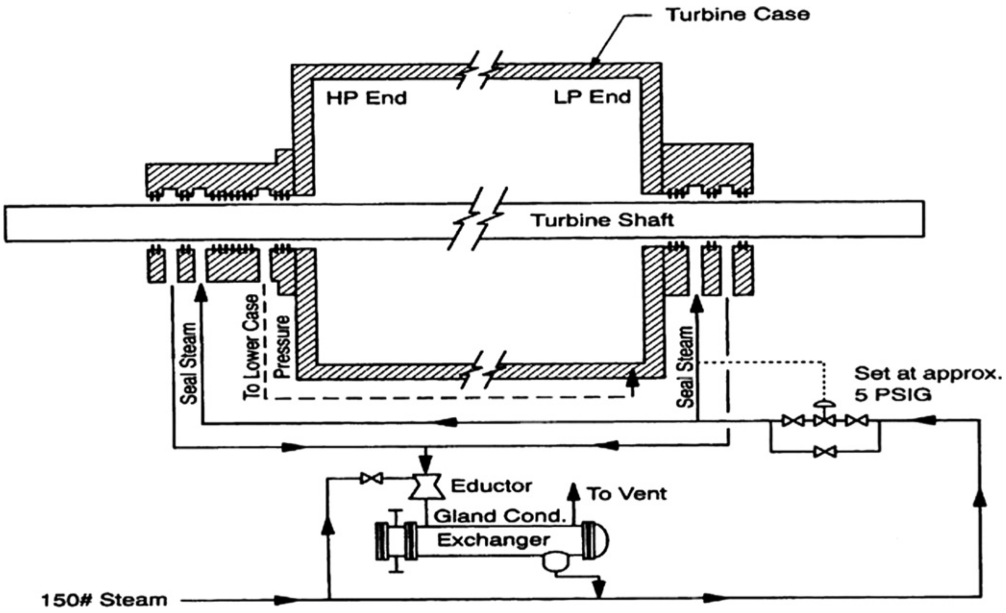


Fig 7.36.2 • Steam turbine gland seal system (Courtesy of M.E. Crane, Consultant)

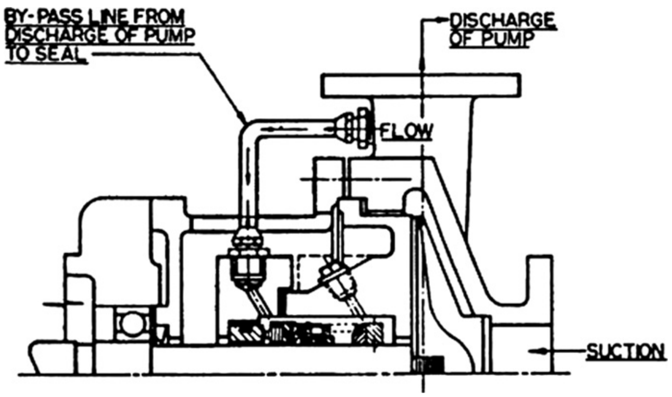


Fig 7.36.3 • Liquid buffer systems or pump flush system (Courtesy of John Crane Co.)

System operation

Refer to Figure 7.36.5 for a simple buffer gas system application. This application would be used for a process gas retaining system. Buffer gas is supplied to the system from an external source. Note that the reliability of the external source will directly influence the reliability of this buffer gas system. The gas flow rate is differential pressure controlled by control valve CV-1, which senses the compressor case reference pressure (on the balance drum or highest casing pressure end). Note that in this single valve design, the reference pressure closest to the balance drum must be used, since this pressure will be slightly higher than the suction pressure as a result of the balance line pressure drop. Referencing in this manner ensures that the differential of buffer gas on both ends will be at least the minimum amount set. The suction end buffer gas differential pressure will therefore be slightly higher.

It is important to note that differential pressure control is historically used for most buffer gas systems. However, when one considers the concept of an equivalent orifice, it can be seen that this valve is actually referencing flow in order to provide a constant flow across the buffer gas system. The flow then enters both ends of the compressor downstream of the buffer gas valve. Buffer gas flow then encounters two (2) equivalent orifices on each end of the compressor. The labyrinth towards the inboard side of the compressor, being equivalent orifices 1A and B and the labyrinth towards the seal ends being equivalent orifices 2A and B. Note that wear of either orifice will result in increased flow through that orifice. Wear can be the result of vibration, liquid entrainment, abrasive material or other causes.

Type	Function
1. Dead ended or seal chamber referenced system	■ Prevent contamination of seal oil by process gas
2. Dead ended type	■ Warm suction end of compressor
3. Vented trap system to flare or atmosphere (where acceptable environmentally and safety)	■ Prevent contamination of seal oil by process gas ■ Prevent seal oil from entering the compressor
4. Vented trap system returned to compressor	■ Prevent contamination of seal oil by process gas
5. Vented trap system	
	■ Prevent seal oil in compressor ■ Warm suction end of refrigeration Compressor ■ Prevent seal oil in compressor
6. Flow ratio controlled vented trap system	■ Positively prevent contamination of seal oil by process gas while ensuring seal oil does not enter compressor

Note: Systems 1–5 are differential pressure (buffer gas to process gas) controlled

Fig 7.36.4 • Buffer gas types and requirements

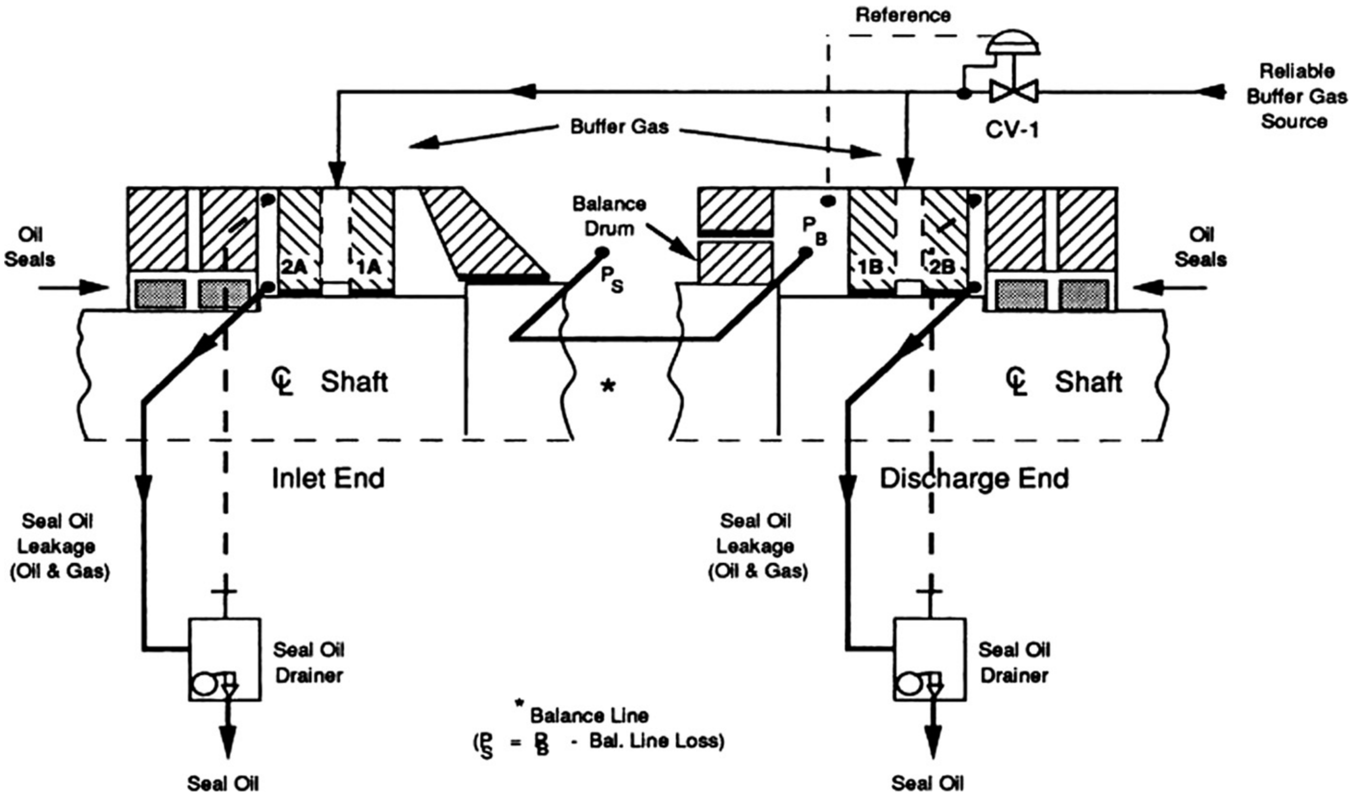


Fig 7.36.5 • Typical buffer gas system

The flow on each end is then distributed back to the compressor and towards the seal. This action ensures that the seal interface is always in contact with buffer gas, which is a controlled gas that meets unit seal cleanliness requirements. Depending upon the oil trap configuration, buffer gas may flow back to the unit. In this case, it will flow to the compressor.

It is important to note that this system only ensures that the process gas will not come in contact with the seal. Greater flow towards the compressor than towards the sour oil traps could result in seal oil ingestion into the machine. If this is not acceptable, a modification to this system must be made. This will be discussed below.

Typical buffer gas velocities towards the compressor end seal are 6–9 m/sec (20–30 ft/sec) but should take into account maximum (worn) labyrinth clearances.

Sources of buffer gas

The sources of buffer gas can vary with the application. Buffer gas can be obtained directly from the compressor if the gas is compatible with the seal oil, or from any external source that meets buffer gas specifications. When buffer gas is provided from an external source, one must consider the reliability of the source in assessing total buffer gas system reliability. Note that when buffer gas is taken from the compressor, it should be superheated to a sufficient value to prevent condensation across the labyrinth. Consideration should also be given to applications that are using water and/or solvent injection to prevent fouling.

Preventive maintenance

Referring to Figure 7.36.5, typical preventive maintenance of buffer gas systems should include the following:

Periodic valve movement

Some buffer gas systems usually are quite stable, since differential pressure of buffer gas valve and buffer gas opening can remain relatively constant. The valve should be stroked periodically to ensure freedom of movement. Care must be taken to ensure this action does not result in a unit trip since the compressor seal oil or seal gas system differential pressure is referenced to the buffer gas pressure.

Switch calibration

Periodic switch or transmitter calibration is essential to ensure proper operation of all devices and acceptable set points.

Flow meter measurement and calibration

As mentioned, the system is actually a flow control system. Differential pressure control has historically been used because of its accuracy over flow control. However, there are systems today that very accurately control flow. In order to assess the operation of buffer gas system and the deterioration of any system components, buffer gas flow measurement is essential. Flow meter values should periodically be recorded by operators and flow meters should be calibrated. Flow meters should be

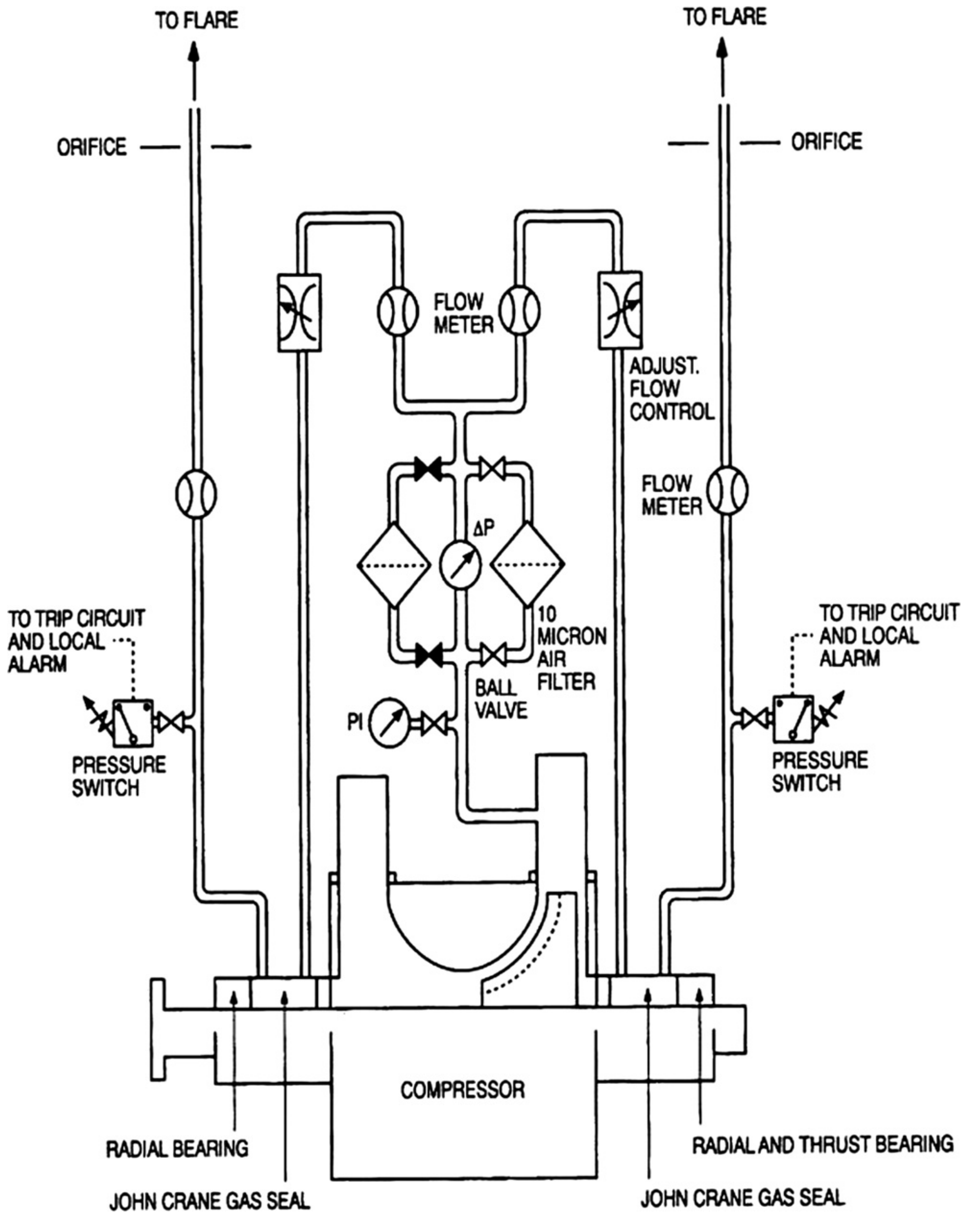


Fig 7.36.6 • Typical gas seal system
(Courtesy of John Crane Co.)

installed in the inlet and the outlet (in a flow through system) or any buffer gas system. By comparing the flow into the system vs. the flow through the outlet of the system, the inner flow to the compressor can be known at all times. An increase of inlet flow at the same differential pressure conditions will indicate wear of the inner labyrinth and may result in necessary maintenance. Understanding the condition of the buffer seal labyrinths can be very important in planning shutdowns. Change of buffer gas flows suddenly with corresponding sudden changes or usages of seal oil will indicate damage to the buffer gas sealing labyrinths. This can be the result of a number of upsets occurring inside the compressor:

A. Surge
B. High vibration (due to unbalance)
C. High vibration (due to liquid entrainment)
D. High vibration (due to operation at critical speeds)
E. Labyrinth erosion due to saturated gas flow

Buffer gas valve stability

Buffer gas valve stability should be checked to ensure stable operation. Controllers should be reset if any instability is present. Valve instability can lead to compressor alarm and shutdown since the seal oil or gas differential controls are referenced off of buffer gas pressure.

If a flow meter is not present in the system, the buffer gas control valve can be used as a rough flow meter, provided differential pressure across the valve and valve travel are known. Referring to valve sizing data for the specific valve, the approximate valve C_v can be obtained. Solving for flow in the valve coefficient (C_v) equation can lead to conclusions concerning changes in flow in the system.

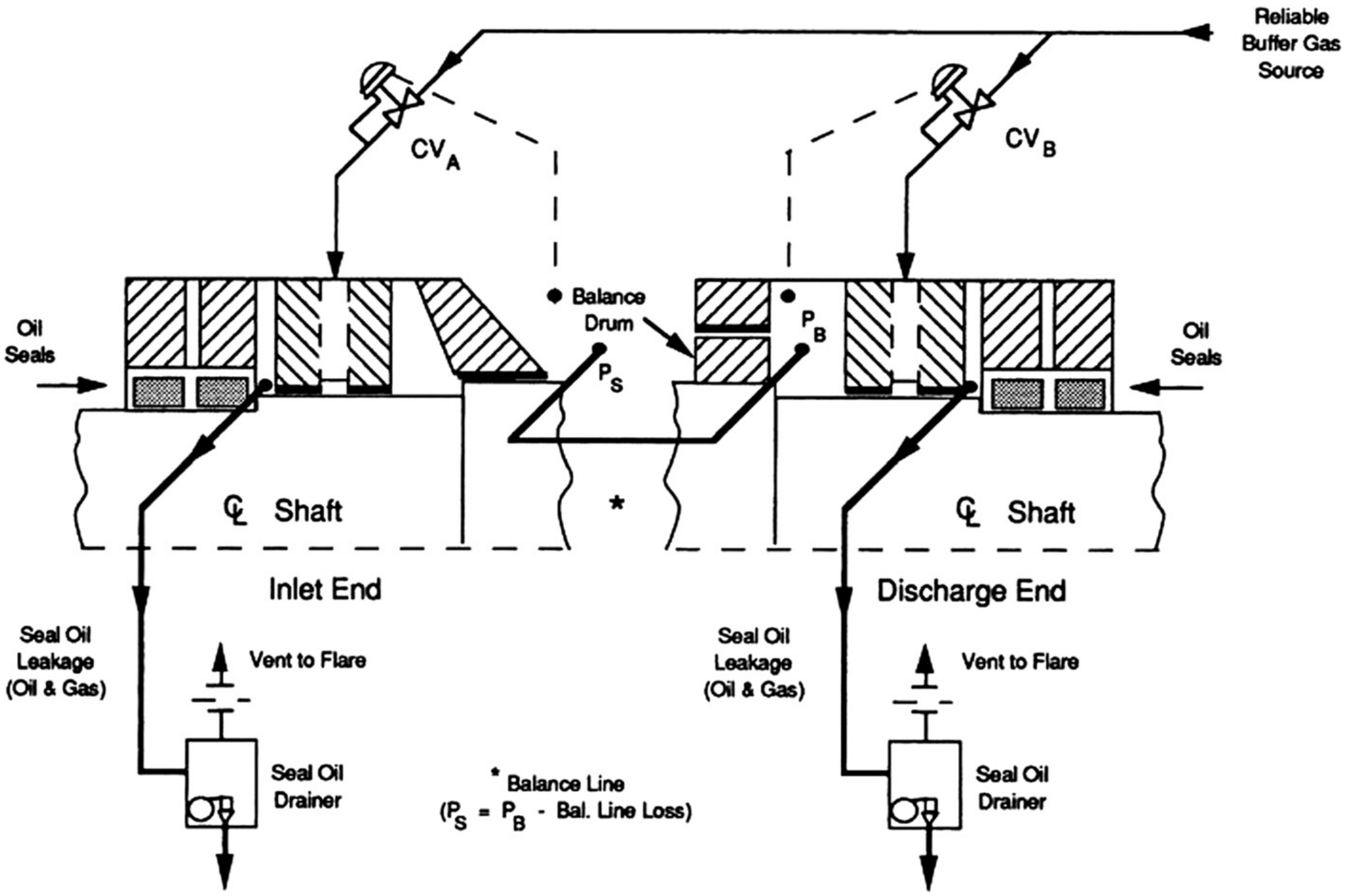


Fig 7.36.7 • Dedicated ΔP control buffer

Gas seal applications

Refer to Figure 7.36.6. This application ensures that the gas in contact with the seal will be buffer gas. In this example, the buffer gas originates from the compressor discharge, but must be conditioned and filtered (ten microns) to meet gas seal requirements to preclude the possibility of gas seal passage fouling. In this case, oil ingestion is not a problem since a gas seal is used. Also, the differential pressure does not need to be regulated since the seal can withstand high differential pressures.

Multiple control valves, vented traps

Figure 7.36.7 shows a design which utilizes a dedicated differential pressure control valve to each seal. This design is utilized to ensure proper pressure differential (buffer gas to process gas) at each seal. In this application, traps are vented which ensures a flow from the buffer gas system gas labyrinths down through the traps. Proper sizing of the vent orifices can eliminate the possibility of oil ingestion.

System reliability factors

Oil ingestion

As mentioned above, oil ingestion can be a serious problem. Processes that require dry gas, utilize exchangers, or incorporate reactors, usually cannot tolerate any large amounts of oil in the system. Oil carryover can necessitate an entire unit shutdown and extensive cleaning time. Care must be taken to select a system that precludes this possibility.

Buffer gas reference

Systems incorporating buffer gas use the buffer gas supply as a reference for differential pressure control of the liquid seals. The valve selection of all buffer gas systems must be carefully made to prevent unstable valve operation. Valve hunting will cause proportional swings of the seal differential reference pressure. High swings, i.e., increasing pressure will result in decreasing seal oil to reference gas differential pressure which can lead to unit shutdowns.

Single buffer gas valve, vented traps

Figure 7.36.8 shows this system.

Oil ingestion prevention systems

In the case that oil ingestion is prohibited, the following systems can prevent the possibility of oil carryover into the compressor if properly designed and maintained.

By properly sizing orifices in the vents of each contaminated seal oil drainer, a flow will be initiated to positively ensure all seal oil leakage is brought to the traps. Remember that in the

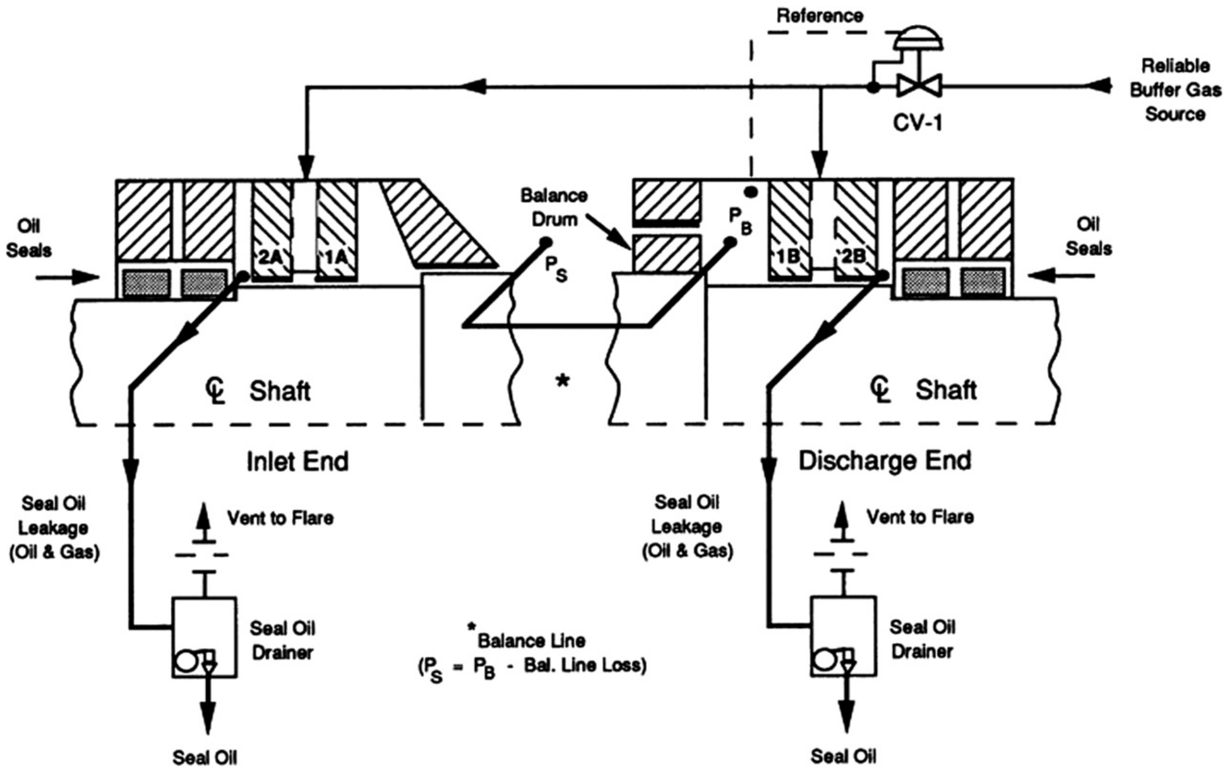


Fig 7.36.8 • Single buffer gas △P control, vented drainers

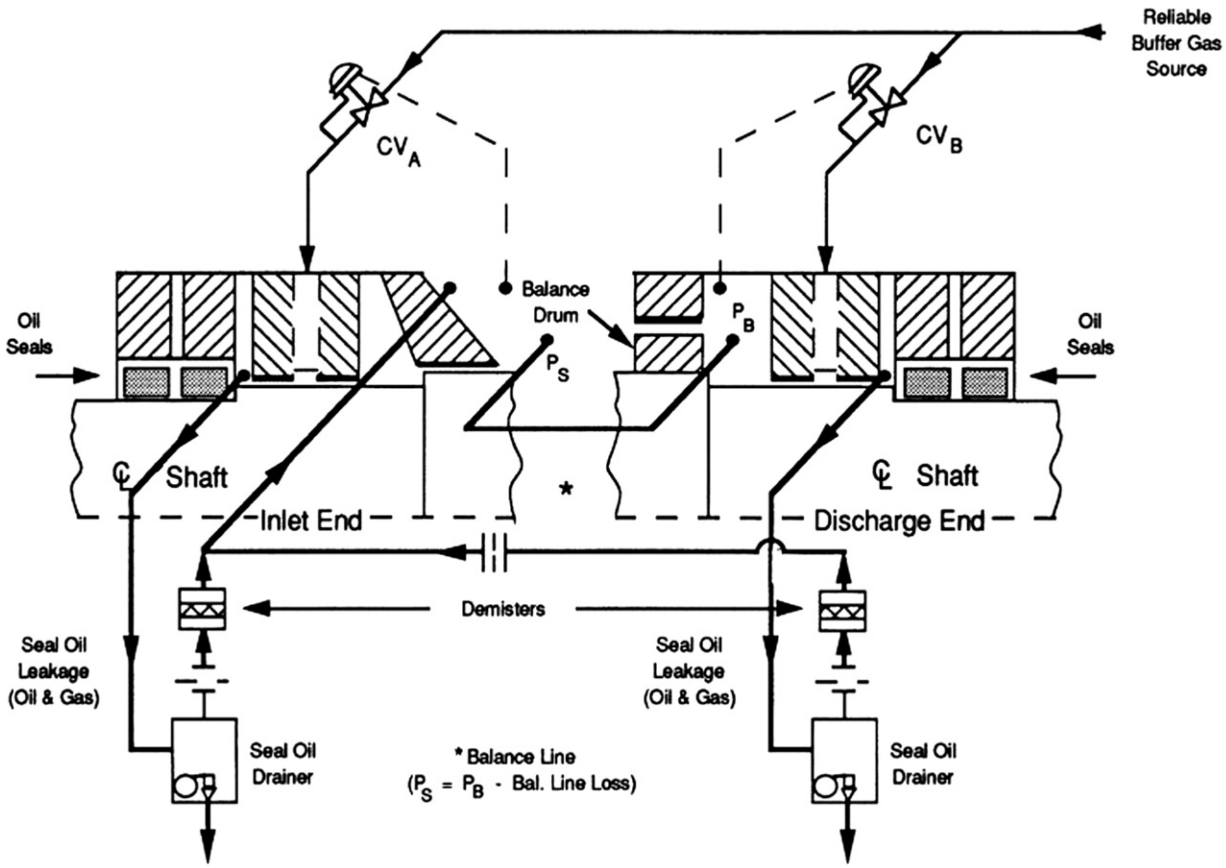


Fig 7.36.9 • Dedicated △P control buffer gas system, vents returned to compressor suction

seal chamber, the leakage is atomized since the unit is operating. As a result, the velocity of the gas to the drain traps must be equal or greater than the velocity of gas towards the compressor to ensure all oil is drained through the traps.

Multiple buffer gas valves, vents returned to the compressor suction

Refer to Figure 7.36.9 which shows this system.

As previously stated, differential valves are used in this example to ensure proper differential or (flow control) to the buffer gas labyrinth. In addition, drainer trap vents are returned to the compressor suction to avoid loss of process gas. In this example, a properly sized, efficient demister is required to prevent the possibility of any oil mist carryover from the drainers to the compressor suction. The demister should be equipped with an oil level glass and checked frequently. In remote operations, a level alarm and automatic drainer should be installed.

Oil ingestion prevention systems do offer the possibility of process gas contamination. If the velocity to the traps is greater than the velocity towards the machine, process gas can be mixed with the buffer gas which can contaminate the seal oil leakage. This type of system should incorporate periodic checks of oil leakage (oil flash point and viscosity) to ensure that seal oil is within spec before it is returned to the seal oil reservoir.

Warming systems

Refrigeration gas compressors, ethylene, propylene, etc., frequently incorporate suction warming systems to prevent seal cavity freeze-up. Discharge pressure is cycled directly to the inlet side of the compressor. The temperature rise in the compressor is usually sufficient to adequately warm this end of the unit. A typical warming system is shown in Figure 7.36.10. Its design is very similar to that of a gas seal system previously discussed.

Process gas and oil ingestion preventive systems

We have previously discussed both process gas retaining systems and oil ingestion preventive systems. There are applications where both these systems are required, that is, buffer gas must prevent both oil ingestion, and ensure the seal is not in contact with process gas. An effective system to use in this application is shown in Figure 7.36.11.

The objective of any buffer gas system is flow-related. In order to meet the objective of ensuring buffer gas is in contact with the seal and precluding oil ingestion, there must be equal gas flow in each direction; inward towards the compressor and outward towards the seal. The system shown achieves that objective by measuring flow to the buffer gas labyrinths and

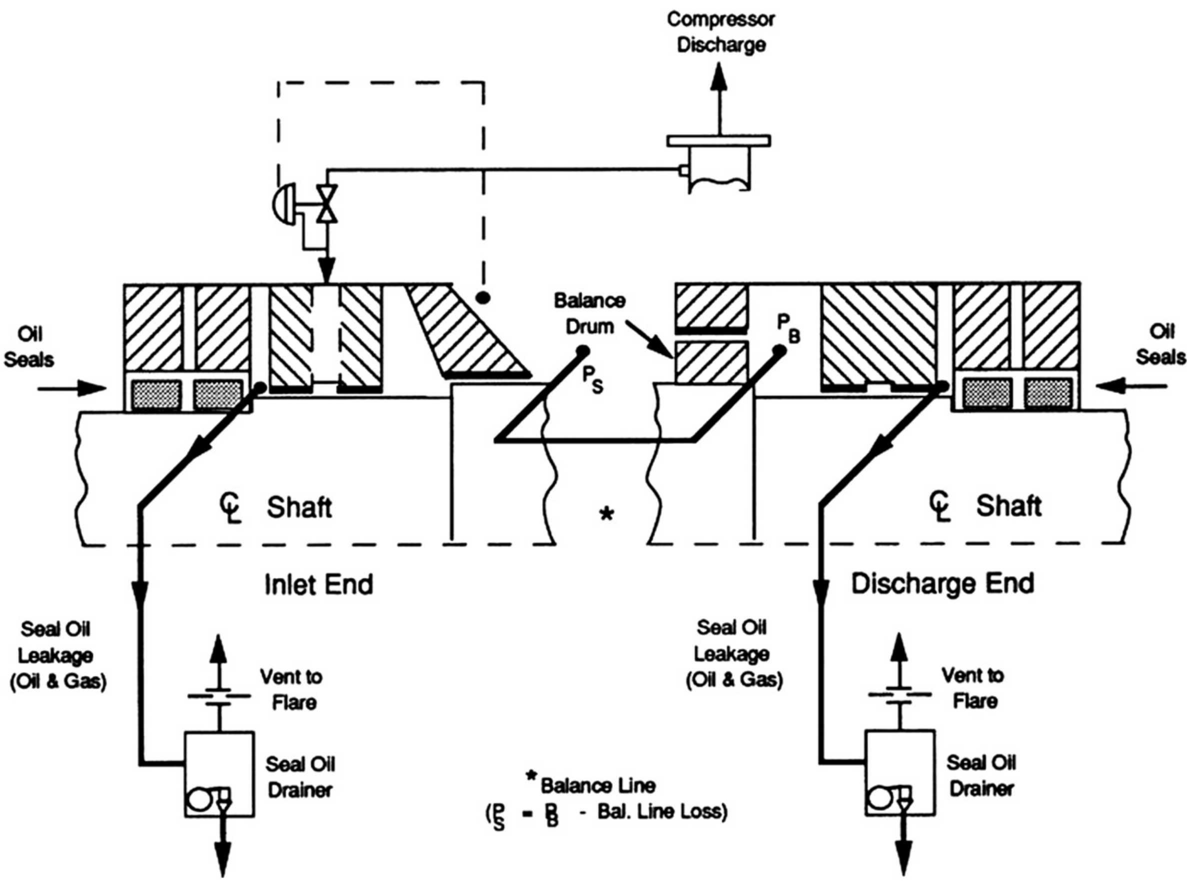


Fig 7.36.10 • Typical compressor warming system

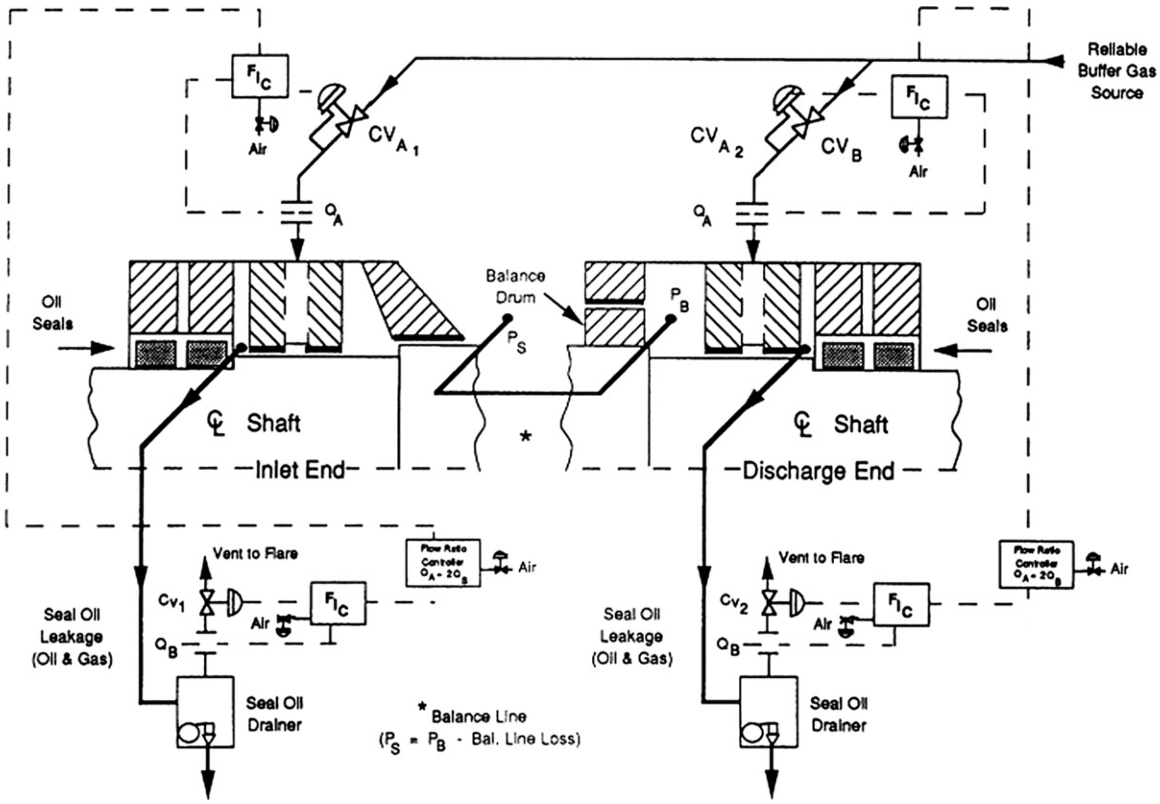


Fig 7.36.11 • Flow ratio buffer gas control system

measuring flow out of the vent trap. A ratio controller ensures that the flow into the system is always twice the flow out of the system through the trap. This will ensure that one-half of the buffer gas flow migrates towards the compressor and one half migrates towards the traps.

Many seal oil systems in operation have experienced problems with oil ingestion into the process. This type of system will positively prevent these problems by providing a variable orifice (control valve) on each drainer vent to ensure contaminated seal oil will be returned to the drainers. A cost effective alternative to this system, when buffer gas is not provided, is to install a bypass needle valve around each orifice and to open this valve (to increase gas velocity to the drainer) when seal oil is detected in the compressor loop or when seal oil reservoir level falls.

Pressure balancing of multi-stage compressor trains

In order to simplify the seal oil system, all the seal cavities of a multi-case compressor train are frequently balanced in reference to the same pressure. This is achieved by reducing pressures from the higher case to the lower case in a labyrinth system. While this design is simple and achieves the objectives, care must be taken to know the characteristics of the particular gas involved. There have been instances where large pressure drops will cause liquid (hydrates) to drop out of the gas, which will then result in rapid erosion of labyrinth teeth and increase of clearances. Applications of this type are solved by heating the gas prior to injection into the compressor.

Best Practice 7.37

Require separate drains with sight glass and TI in each atmospheric seal drain line, to enable the monitoring of the atmospheric seal condition and ensure seal maintenance is only performed during a scheduled turnaround.

Failure to monitor atmospheric seal flow (by visual means) and temperature during start-up when atmospheric seal oil pressure may be low, or during upsets will expose the plant to unscheduled shutdowns and possible gas releases to the plant environment.

Lessons Learned

Failure to require a separate drain for each atmospheric seal and to install sight glasses and temperature instruments in the drain lines has led to un-scheduled plant shutdowns and gas releases when excessive atmospheric seal wear went undetected.

Benchmarks

This best practice has been used since the late 1960s to ensure that the atmospheric seal could be monitored during operation, and to provide valuable information to the turnaround planning team to relegate the required seal change out to a planned turnaround.

B.P. 7.37. Supporting Material

See B.P: 7.34 for supporting material.

Best Practice 7.38

Require that the seal oil system is on before gas is introduced to the compressor case for bushing seals to prevent a gas release to the plant.

Lack of design information disclosed to plant operation personnel and/or those responsible for writing the plant operating procedures has led to the false belief that starting the oil system prior to introducing gas into the compressor will flood the compressor with seal oil.

All compressor vendors design their units with shaft steps or other means to greatly minimize the entrance of oil into the compressor.

Since the only means to prevent entrance of process gas into a bushing seal is a positive seal oil to seal gas pressure differential, failure to commission the seal oil system prior to admitting process gas into the compressor will:

- Result in a gas release to the plant environment
- Trap any debris in the process gas stream into the inner bushing seal thus failing the seal and possibly distributing debris through the seal oil system up to the outside of the filter cartridge.

Lessons Learned

Introducing gas into the compressor before the seal oil system is commissioned will cause a gas release into the plant which can be life threatening if the gas is sour, and will drastically reduce seal MTBF.

I have been in plants where sour gas bushing seal applications, without sweet buffer gas, allowed process gas to be introduced into the compressor before the seal oil system was put into operation.

Benchmarks

This best practice has been used since the mid-1980s when a plant had a history of bushing seal low MTBFs (less than 12 months), because gas was introduced into the compressor before the oil system was commissioned and the operating procedure was written to allow this practice. Since that time, this best practice has been used to ensure optimum bushing seal MTBFs (in excess of 100 months).

B.P. 7.38. Supporting Material

See B.P: 7.34 for supporting material.

Best Practice 7.39

Oil bushing seals without a through-flow provision should have a false signal reference pressure, to ensure adequate bushing flow to provide cooling for the seal during low process gas pressure conditions.

Since the flow through the atmospheric bushing is directly proportional to the reference gas pressure, low reference gas pressures during start-up will result in low cooling flow through the atmospheric bushing and corresponding bushing wear due to overheating.

A means of ensuring the proper atmospheric bushing flow during low reference pressure conditions is to provide a false reference gas signal via a pressure control valve and check valve arrangement in the reference line to duplicate the pressure drop across the atmospheric bushing during normal operating conditions.

It is true that during this time the differential across the inner (gas side) bushing seal will be excessive, but this is a temporary condition that will not affect compressor operation.

Lessons Learned

Failure to provide sufficient seal oil pressure to the atmospheric bushing during low reference gas pressure conditions exposes the atmospheric bushing to overheating and low MTBFs (less than 12 months).

Benchmarks

This best practice has been used since 1995; mostly as a recommendation to correct long term bushing seal low MTBFs, to maximize oil seal reliability and to achieve oil seal MTBFs in excess of 100 months.

B.P. 7.39. Supporting Material

Figure 7.39.1 shows a typical reference gas 'false signal' arrangement that will provide sufficient pressure to ensure adequate atmospheric bushing oil flow during low reference gas pressure (start-up) conditions.

KEY ITEM	DESCRIPTION
1.	Pressure control valve to maintain constant clean gas supply pressure to seal oil tank (set for 200 kPa below normal reference pressure).
2.	Pressure transmitter to monitor 'false signal' set pressure.
3.	High signal selector device (ball check valve) will open to provide inert gas to seal oil tank whenever PCV set pressure is greater than reference pressure and close whenever reference pressure is greater than PCV set pressure.
4.	Check valve to prevent clean gas supply from entering compressor.
5.	Overhead seal oil tank.
6.	Overhead seal oil tank level controll to maintain constant seal oil/gas differential.

Fig 7.39.1 • Typical reference gas "false signal" arrangement

Best Practice 7.40

Design overhead seal oil tanks with easy access inspection flanges to check internal condition at each turnaround.

Since the majority of overhead seal oil tanks do not employ a separation bladder between the reference gas and the seal oil, any debris contained in the reference gas will enter the oil seal every time the compressor is shut down, since there cannot be a filter between the overhead seal oil tank and the seals.

Most seal oil systems that use an overhead seal oil tank to provide the required seal oil to seal gas differential pressure do not have provision for easy inspection of the tank during a turnaround.

If the seal oil overhead tank is found to contain excessive debris, a clean buffer gas system should be installed to continuously provide a clear source of reference gas pressure to ensure that the bushing seal will never be exposed to debris and/or chemical compounds which can render the bushing seal useless.

Lessons Learned

Many oil seal failures are caused by debris from the process system entering the small clearances of the oil seals via the reference gas connection on top of the seal oil tank.

Oil seal systems do not allow condition monitoring of the overhead seal oil tank, which is exposed to contamination from the reference gas connection.

As a result, the internal condition of the overhead seal oil tank should be inspected during each turnaround to determine if it should be flushed during the turnaround.

Benchmarks

This best practice has been recommended since 1990, to ensure optimum bushing seal MTBFs and to prevent gas releases to the plant environment.

B.P. 7.40. Supporting Material

Please refer to material in B.P: 7.23.

Best Practice 7.41

Only use seal oil drainers with external level control valves, to prevent drainer blockage or friction binding of the internal drainer float valve.

Seal oil drainers with external level control valves positively eliminate the possibility of drainer blockage and open drainer valves that will allow process gas to enter the degassing tank in large quantities and eventually enter the oil reservoir exposing the plant to a safety hazard.

Seal oil drainers equipped with an internal float valve mechanism are prone to blockage (due to the small orifice inserted in the valve seat to control the rate of oil drainage and to prevent excessive amounts of process gas escaping before the float valve reseats). They are also prone to friction binding of the internal float mechanism if any process gas debris is present in the drainer.

Lessons Learned

Internal float type drainers have been responsible for plant safety issues (oil reservoir explosions) when the internal mechanism has become friction bound and stayed open, thus allowing large quantities of gas to enter the oil reservoir. In addition, blocked internal float type drainers have been responsible for large amounts of oil entering the process system.

A recent (2010) case of internal float drainer blockage was experienced in a petrochemical plant. The blocked drainer was responsible for large amounts of oil entering the ethylene refrigeration system and freezing on the chiller coils, requiring the entire plant to be shut down for 5 days, with a revenue loss of over $5MM.

Benchmarks

This best practice has been used since 1980 for all projects using seal oil systems and in auxiliary system audits to achieve optimum safe and reliable operation of all seal oil systems.

B.P. 7.41. Supporting Material

Please refer to material in B.P: 7.23.

Best Practice 7.42

Install needle valve bypasses around drainer vent orifices to positively prevent entrance of seal oil into the compressor and the process.

If the seal oil drainer vent system (orifices or valves) are not sufficient to exert a greater force than the force of the impeller on the seal oil/gas mixture, oil will enter the compressor.

The force of the impeller on the seal oil/gas mixture is determined by the clearance of the shaft end labyrinths which can increase due to shaft vibration and/or hydrate erosion.

Therefore, the shaft end labyrinths are actually variable orifices which can and will change clearances with time.

Installation of a needle valve around the drainer vent fixed orifice renders this orifice variable as well and will allow adjustment when the shaft end labyrinth clearance increases to ensure that seal oil will be directed to the drainers and not the compressor.

Lessons Learned

The primary cause of seal oil entry into the compressor is the failure to have drainer orifices of large enough size when the shaft end labyrinth clearance increases.

This lesson learned has led to many retrofits to dry gas seal from oil seal systems in the 1990s and more recently. This retrofit is very costly and risky, since the compressor rotor critical speeds will be affected when a dry gas seal with much lower damping than an oil seal is installed. Using B.P. 7.42 in lieu of dry gas seals saves considerable costs and ensures positive results.

Benchmarks

This best practice has been used since 1986 to alleviate all problems concerning entry of seal oil into the compressor resulting in optimum seal MTBFs (greater than 100 months) and compressor reliability.

B.P. 7.42. Supporting Material

Please refer to material in B.P: 7.23.

Best Practice 7.43

Use a dedicated vacuum degasser unit installed in the degassing tank to transfer seal oil back to the oil reservoir.

Degassing tanks, even with the API specified 72 hour retention time, nitrogen bubbling and heating are not totally effective in ensuring that oil reservoirs in seal oil systems will be gas free.

Using a dedicated, slip steam, vacuum degassing unit to service the retained seal oil in the degassing tank will positively prevent the entrance of entrained process gases into the oil reservoir.

Lessons Learned

Oil samples in seal oil reservoirs (oil − low flash point) show that even properly sized and designed degassing tanks do not positively eliminate entrained process gas from the oil reservoir which exposes the plant to safety issues.

Benchmarks

This best practice has been used since the mid-1980s to ensure that oil reservoirs in seal oil services are free of entrained gases, since their presence can lower the oil flash point below the operating temperatures of the system, and hence expose the plant to safety issues. Using a dedicated vacuum degassing unit in the degassing tank has resulted in maximum critical unit reliability in compressor trains using seal oil systems (99.7% +).

B.P. 7.43. Supporting Material

Oil reclamation units

In cases where the degassing tanks have proven not to be effective, or are inadequately sized, the use of an oil reclamation unit should be considered. All oil from drainers should be collected or directly piped to an oil reclamation unit. Considering the cost of typical mineral oil (approximately $25 per gallon), a standard compressor with two seals can use $1,000 of oil per day if oil cannot be returned to the reservoir.

Figure 7.43.1 shows a typical oil reclamation unit which has the capability to degas all oil entering the unit. In large installations, this unit may be justified for direct installation downstream of the drainers taking suction from the degassing tank and returning oil to the seal oil reservoir. For smaller systems, the purchase of one unit should be considered for the site. Gas entrained liquids can then be collected and transported to the unit for reclamation at specified intervals or the unit can be temporarily installed between the degassing tank and the seal oil reservoir.

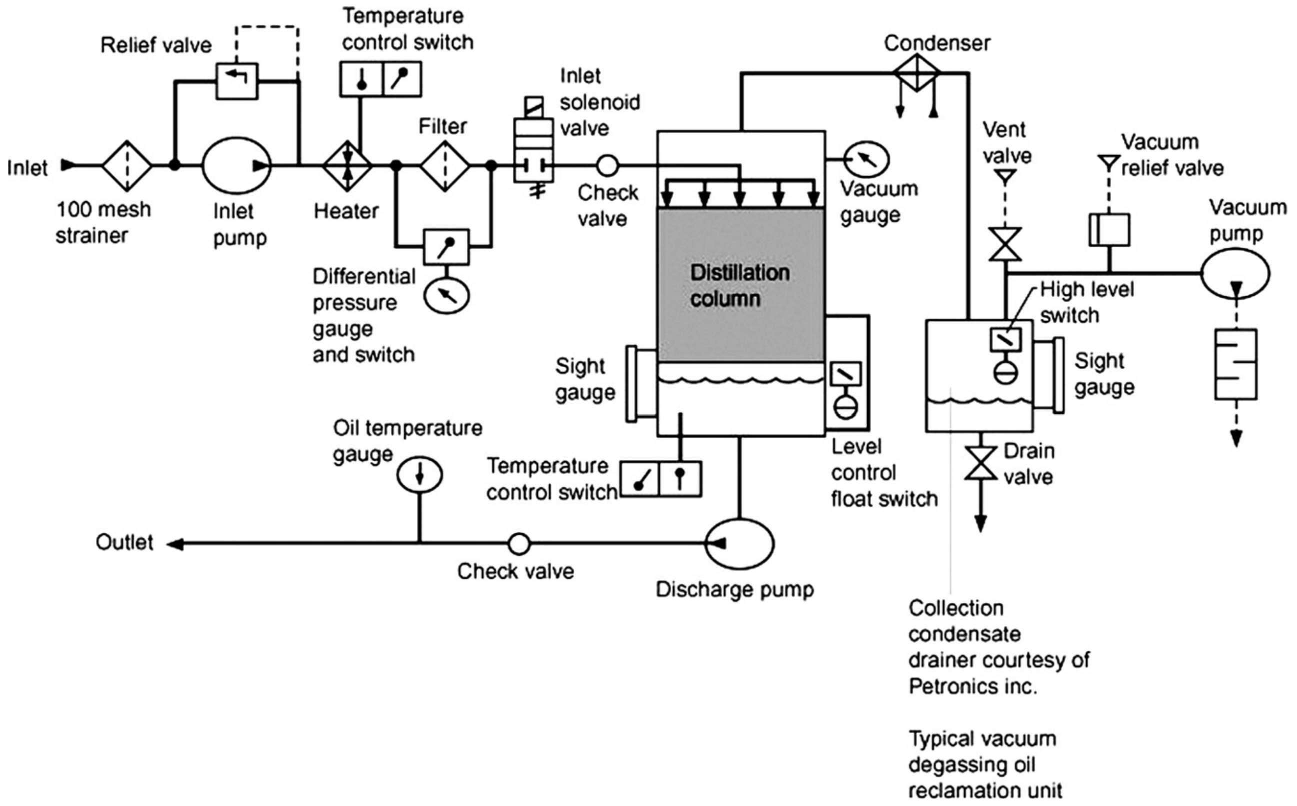


Fig 7.43.1 • Oil reclamation unit schematic (Courtesy of Petronics, Inc.)

8

Pump Mechanical Seal Flush Best Practices

Experience shows that pump MTBFs (mean time between failures) are directly related to mechanical seal reliability. It is well known that pump mechanical seal MTBF is the lowest of all machinery components.

This chapter will therefore present best practices for pump mechanical seals in the hope that these proven Best Practices will significantly increase your plant's mechanical seal and pump MTBFs.

Best Practice 8.1

Confirm the actual seal operating conditions on the data sheet (PT, vapor pressure, S.G., P1 and P2) to optimize mechanical seal MTBFs.

Of all machinery components, mechanical seals are the most affected by the process conditions.

Accurate definition of the following parameters on the pump and seal data sheet will go a long way towards assuring optimum pump mechanical seal MTBFs:

- Fluid vapor pressure
- Fluid specific gravity and viscosity
- Fluid pumping temperature (PT)
- Suction pressure
- Cooling medium temperatures (if a flush cooler will be used)
- External flush fluid pressures, temperature, vapor pressure, viscosity and specific gravity

Lessons Learned

Failure to properly specify the correct process conditions on the pump and/or mechanical seal data sheet will result in lower than optimum seal MTBFs.

Most 'bad actor' seals (seals with more than one failure per year) result from improper specification of process conditions on the data sheets.

Always check all operating seal process conditions against the data sheet values when investigating a 'bad actor' seal.

If the proper instrumentation (pressure gauge, etc.) is not installed, have the appropriate instruments installed at the time of seal replacement.

Benchmarks

This best practice has been used since 1990 to troubleshoot mechanical seal problems and to ensure maximum mechanical seal MTBFs (greater than 100 months).

B.P. 8.1. Supporting Material

Confirm process conditions are actually as stated in data sheets. If the 'bad actor' pump seal reliability, when operating in the EROE (see B.P. 2.7 for EROE details), does not significantly improve, a complete check of seal fluid conditions in the seal chamber is required. Figure 8.1.1 shows the process variables that influence seal reliability.

- Seal chamber temperature
- Seal chamber pressure
- Seal fluid characteristics:
  - Cleanliness
  - Vapor pressure
  - Viscosity
  - Specific heat
  - Specific gravity

Fig 8.1.1 • Seal reliability and pump fluid conditions

Temperature monitoring

Since seal chamber or seal flush line pressure gauges are not usually installed on new pumps, the first place to start is with seal chamber temperature. Flush plans with coolers (21, 23, 41, etc.) have an option for a thermometer downstream of the cooler which can be used to determine the seal chamber temperature. This value must be compared to the PT (pumping temperature) value listed on the data sheet. If the measured value does not agree with the data sheet, consult with operations first to see if process changes can be made. If they cannot be made, discuss the measured temperature with the seal vendor representative. Alternative options for measuring seal fluid temperature are noted in Figure 8.1.2.

- Temperature gun close to seal gland – be sure to monitor off of a non-reflective surface
- Surface contact thermometer on flush line – add 5°C to measured value if pipe and use recorded value if SS tubing
- Install a thermometer in the flush line
- Use an infrared camera to record temperature and temperature profile

Fig 8.1.2 • Alternative methods to measure seal fluid temperature

Pressure monitoring

Unless the plant assigned machinery specialist is 'world class' there are usually no pressure gauges in the flush line or seal chamber. Our recommendations concerning seal chamber pressure monitoring are presented in Figure 8.1.3. The measured seal chamber pressure must be compared to the seal vendor's assumed value, which should be on the seal layout drawing. If this value is not present, the seal vendor should be consulted. Note that the seal vendor assumed seal chamber pressure is the value that was used in the calculation of the seal PV. If the measured value does not agree with the data sheet, consult with operations first to see if process changes can be made. If process changes cannot be made, discuss this fact with the seal vendor representative.

- Install a pressure gauge in the seal chamber
- Modify the seal flush line for installation of a pressure gauge.

Fig 8.1.3 • Seal chamber pressure monitoring guidelines for 'bad actor' seals

Seal fluid characteristics

If all of the above mentioned items are in accordance with the seal design (EROE, seal chamber temperature and seal chamber pressure), a check of the seal fluid characteristics is required. Figure 8.1.4 presents guidelines for checking the seal fluid characteristics.

- Take a sample to determine if debris is present
- If seal fluid is not water or product fluid (which must meet product specifications) have sample analyzed
- Send sample results to seal vendor

Fig 8.1.4 • Seal fluid check guidelines

If the measured fluid sample parameters are not as stated on the data sheet, consult with operations first to see if process changes can be made. If process changes cannot be made, discuss this with the seal vendor representative.

Variance in seal chamber pressures and temperatures, if the pump is operating in the EROE, will most likely be caused by malfunction of components in the flush system. Figure 8.1.5 defines those flush system components which can affect seal chamber pressures and temperatures.

- Orifice condition
- Seal gland condition
- Throat bushing clearance
- Cooler condition
- Strainer condition
- Cyclone separator condition
- Buffer fluid condition (tandem seal)
- Buffer fluid level (tandem seal)
- Pumping ring condition
- Buffer fluid pressure (tandem seal)

Fig 8.1.5 • Seal reliability as a function of flush system component condition

Based on the seal flush plan used, the flush system components should be checked in the logical order – starting with the beginning of the flush system and ending with the throat bushing. (The exception is flush plan 13 which begins with the throat bushing and ends at the pump suction pipe.) FAI 'seal maintenance best practice' requires that the flush system be completely checked (including the throat bushing clearance) each time a seal is changed. Note that this recommendation applies to all seal configurations: single, tandem (dual un-pressurized) or double (dual pressurized).

Best Practice 8.2

Pre-select mechanical seal and flush system design during the pre-FEED project phase, based on plant, company and/or industry experience, to optimize mechanical seal MTBF.

Use plant, company and industry lessons learned to properly select a flush system that will result in optimum seal life – in your plant!

Frequently, the flush system is selected by the process licensor or the EP&C (contractor) and does not reflect actual plant conditions.

Be proactive early in the project design (pre-FEED phase) to convince project management of the proper flush systems to apply for all pump services.

Lessons Learned

The use of the following flush system components where they are not warranted has resulted in low mechanical seal MTBFs and has exposed plants to safety issues:

- **Flush line strainers**
- **Cyclone separators**
- **Flush line coolers**

Flush line strainers can expose plants to seal failure and safety issues, since they are not monitored in the control room and can result in flush line blockage, which will fail the mechanical seal and can expose the plant to significant safety issues in hydrocarbon applications.

Benchmarks

This best practice has been used since the mid-1970s, when I became involved with pump selection. Since that time, prohibiting the use of flush line strainers and cyclone separators has resulted in optimum mechanical seal safety and reliability during field mechanical seal reliability optimization audits for all projects.

B.P. 8.2. Supporting Material

Optimal seal plans for various applications

We know the design aspects and how to monitor different flush systems, but when would certain flush systems be better than others? Figure 8.2.1 lists the parameters required for a reliable flush source.

Mechanical seal flush must possess the following qualities for optimal seal life:

- Cool
- Clean
- Approximately 345 kPa (50 psi) above vapor pressure (psia)
- Most importantly it must be cost effective!

Fig 8.2.1 • Optimal flush plans for various applications – overview

Therefore, if you can say that the flush system for an application can provide all the qualities (most important is cost effectiveness) listed in Figure 8.2.1, then you have selected the optimal flush system.

The following points outline different seal flushing scenarios, listing considerations specific to each. Note that these are general points, and should not be taken as being definitive for every application; however they can aid in flush plan selection. Ultimately, the seal vendor should be consulted when selecting the seal and flush plan; the more input they have, the higher seal reliability your plant should see as a result.

HC service with no known solid particles and temperature under 150°C (300°F)

- Assuming satisfactory vapor margin (approximately 345 kPa [50 psi] above vapor pressure) in seal chamber a Plan 11 would be the optimal choice.
- A vertical pump application would require venting of the seal chamber back to suction if possible (Plan 13 or 14).
- A Plan 52, 53 or 54 (53 or 54 preferred) would be recommended in low S.G. (< 0.7) services.

Fig 8.2.2 • Optimal flush plans for various applications: option 1

Clean HC service (e.g. #2 FO) between 150 and 230°C (300 and 450°F)

- Plan 23 is most efficient in cooling, however proper installation and venting is required.
- Plan 21 will be sufficient in most cases and will be easier to operate... note that the orifice sizing is critical here as this determines the velocity of the fluid through the heat exchanger.

Fig 8.2.3 • Optimal flush plans for various applications: option 2

Hot oil service above 230°C (450°F [typically tower bottoms])

- A single seal with the use of a Plan 32 is the most reliable/cost effective if a reliable source is available nearby... note that if the source is a product that has to be reprocessed it may not be cost effective.
- A dual pressurized seal would be the best option (Plan 53 or 54) if a Plan 32 is deemed not feasible.

Fig 8.2.4 • Optimal flush plans for various applications: option 3

Acid service (e.g. H_2SO_4)

- Reliable Plan 32 can be used, however it is very critical for it to be operating at times when pump is installed in field (standby and startup situations).
- A dual pressurized seal, whether contacting (Plan 53/54) or non-contacting (Plan 74) will give the optimal seal life with ease of operation... note that process side seal components will need to be constructed of materials that are corrosion resistant to the particular acid.

Fig 8.2.5 • Optimal flush plans for various applications: option 4

Dirty service (containing suspended solid particles)

- A cyclone separator (Plan 31) can be effective, however it must be sized correctly and particle size must not fluctuate greatly.
- A clean external flush source (Plan 32 for single, Plan 54 for dual) will isolate the seal faces from solids that can cause premature failure due to abrasion.

Fig 8.2.6 • Optimal flush plans for various applications: option 5

Best Practice 8.3

A cartridge seal design should be used whenever possible to ensure proper seal assembly and optimum mechanical seal MTBFs.

The use of cartridge mechanical seals significantly minimizes installation errors.

While cartridge seals are more expensive than component seals, the additional cost can be justified in many cases based on the material costs and revenue losses of component seals.

Lessons Learned

Modifying existing component mechanical seals for cartridge seal assemblies, based on a life cycle cost analysis, has resulted in significantly higher seal MTBFs.

Many clients have standardized on cartridge seals for all pumps that can accommodate them, justified by a life cycle cost analysis which compares the material, maintenance and lost revenue costs of component seals to the additional costs of cartridge seals.

Benchmarks

This best practice has been used since 2000 for new projects, and recommends modifications to cartridge seals for plant 'bad actor' seals. This best practice has resulted in seal MTBFs of greater than 48 months, compared with previous MTBFs below 12 months.

B.P. 8.3. Supporting Material

Pusher vs. non-pusher

Mechanical seals are typically categorized into two major types; pusher and non-pusher.

A pusher-type seal consists of a primary sealing ring assembled with an ‘O’ ring and springs (can be one or multiple). The purpose of this is to force the sealing fluid across the face and keep it from leaking to the ID (atmospheric) side of the seal. The dynamic ‘O’ ring is designed to move axially (be pushed) along the shaft or sleeve (in a cartridge seal). The surface underneath the dynamic ‘O’ ring must therefore be very smooth (< 32 RMS) to allow for this axial movement. If solids are abundant in the sealing fluid, they can build up on the ‘O’ ring and prevent this axial movement (hang up).

A non-pusher type seal consists of a bellows assembly. The bellows is a component that acts as both the load element (like a spring in a pusher type) and a secondary sealing element (like an ‘O’ ring in a pusher type). Because the bellows prevents any leakage to the atmospheric side of the seal, and has a large clearance between itself and the shaft or sleeve, it can move freely in the axial direction (no dynamic ‘O’ ring), reducing the potential for hang up.

Refer to Figure 8.3.1 for details on these two types of seals.

Pusher type seals are used more commonly in low S.G. (<0 .7) services.

Pusher Seal	Non-Pusher Seal
■ Closing force supplied by spring(s)	■ Closing force supplied by bellows (no dynamic ‘O’ ring)
■ Used in low temp. services	■ Can be used in high temp services (metal bellows)
■ ‘O’ ring secondary seals	■ Metal bellows use ‘grafoil’ secondary seals to handle high temperature
■ Used in light end services (ethylene, propane, methane, butane, etc.)	

Fig 8.3.1 • Single seal (pusher vs. non-pusher)

Referring back to the previous discussion on balance ratio (Figure 8.3.2), which is balance of the ratio of closing area to opening area of the seal.

Our depiction (Figure 8.3.2) shows the concept of balance ratio in a pusher type seal. In a bellows seal, the secondary sealing element (bellows) generally has a larger diameter than a pusher seal, therefore the closing area is less. Since the closing area is larger and the width of the primary ring face is limited (cannot be too large or it won’t fit in the bellows assembly), the balance ratio cannot be varied as much as in a pusher seal. With light S.G. fluids, it is important to be able to have a range of balance ratios to control where the fluid will vaporize across the faces. It is for this reason that a pusher type seal is desirable in light S.G. services. Note that some applications can have a S.G. of less than 0.7 and contain solids. In these applications, it is still recommended to use a pusher-type seal, however provisions need to be made to ensure the seal will not hang up in operation. Take a look at Figure 8.3.3.

- Single coil spring – reduces potential for springs to hang up since one large diameter spring is used
- Filtration – can be high cost and needs to be maintained for high reliability
- External flush – if an external flush plan can be used, it will provide optimal seal life in this application

Fig 8.3.3 • Considerations for pusher type seal in a low S.G. dirty service

An advantage for using the bellows seal, apart from being less likely to hang up, is that they typically utilize ‘grafoil’ packing rings as their secondary seals. Grafoil packing rings can withstand temperatures of approx. 425°C (800°F), allowing metal bellows seals to be used in refinery bottoms applications with great success.

Dual un-pressurized vs. dual pressurized seals

Today, due to environmental restrictions, dual seals are being selected for more and more applications. There are two arrangements in which dual seals can be used; namely dual un-pressurized (previously called tandem) and dual pressurized (previously called double). Refer to the diagram in Figure 8.3.4, which shows the dual un-pressurized arrangement.

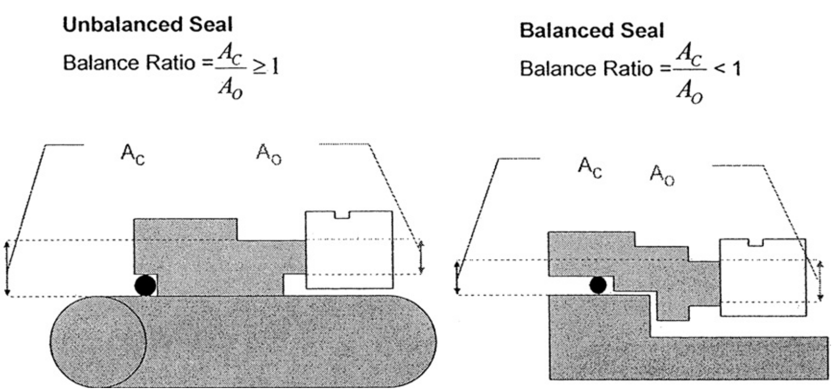


Fig 8.3.2 • Balance ratio = closing area/opening area

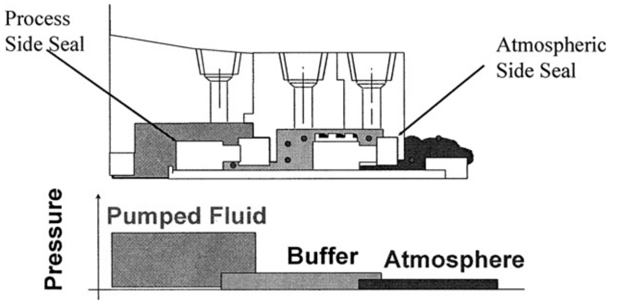


Fig 8.3.4 • Dual un-pressurized

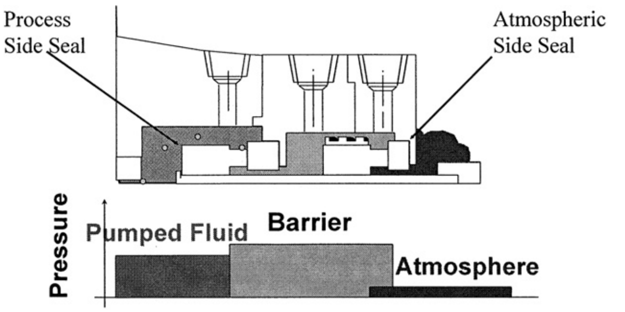


Fig 8.3.5 • Dual pressurized

A dual un-pressurized seal uses a buffer fluid at, or near, atmospheric pressure to lubricate the atmospheric side seal. The buffer fluid pressure is significantly less than the seal chamber pressure, so any leakage occurring across the process side seal will leak into the buffer fluid. This arrangement is very common in applications containing VOCs, as it can potentially reduce the leakage to the environment.

Refer to Figure 8.3.5, showing a dual pressurized seal arrangement. These seals use a barrier fluid (same fluid as dual un-pressurized) at a pressure that is 175 kPa (25 psi) above seal chamber pressure to lubricate the seals. Since the barrier fluid pressure is higher than the seal chamber pressure, any leakage occurring at the process side seal will enter the pump. This arrangement (when working properly) ensures no leakage of the pumped fluid to atmosphere. Note that this arrangement requires the barrier fluid to be compatible with the pumped fluid, since the barrier fluid leaks into the pump. Refer to Figure 8.3.6.

It has been our experience that if an inert gas (or external fluid) is available at the required pressure, and the pumped fluid can accept the barrier fluid (compatible), a dual pressurized seal is potentially more reliable in VOC service.

Material selection of faces and secondary components

It is very important that all parts be resistant to corrosion by the sealing fluid, and allow for optimal sealing at the operating conditions.

Dual Un-pressurized	Dual Pressurized
■ Primary leakage is process fluid, therefore the secondary seal is essentially a backup seal ■ Does not require the use of nitrogen or other inert gases	■ Primary leakage is barrier fluid into the process, therefore the process must be able to accept a small amount of barrier fluid ■ Requires use of nitrogen (or other compatible inert gas) to pressure seal reservoir approx .175 kPa (25 psig) above seal chamber pressure in a Plan 53 ■ A Plan 54 uses an external fluid (synthetic skid or another pump) to lubricate the seals at a pressure of 175 kPa (25 psi) above seal chamber pressure

Fig 8.3.6 • Dual un-pressurized vs. dual pressurized

Metal parts (adaptive hardware)

Adaptive hardware will typically be made of 316 SS, however certain applications can dictate different materials be selected. Refer to Figure 8.3.7.

- Normally 316 SS for large components (sleeve, gland, retainer, etc.)
- Normally hastelloy C or other corrosion resistant alloy for springs
- Acid or chloride services may require hardware be constructed of hastelloy C, chrome alloys, monel or other corrosion resistant material

Fig 8.3.7 • Adaptive hardware materials

Secondary sealing elements ('O' rings)

'O' ring materials vary in their applicability, so caution must be used in their selection. Refer to Figure 8.3.8 for 'O' ring materials and guidelines for use.

- Fluorocarbon (Viton) – most common (relatively cheap) and highly recommended for HC services under 175°C (350°F).
- Perfluoro-elastomer (Kalrez or AFLAS)– used in higher temperature services (175 to 260°C (350 to 500°F)) than Viton and generally highly chemically resistant.
- EPDM – common in hot water (BFW) applications, as it is more resistant to thermal attack in hot water than the two listed above.
- Buna-N – not recommended over the above three materials for most (if not all) applications.

Fig 8.3.8 • 'O' ring material and usage guidelines

Teflon (PTFE), PEEK and Grafoil packing have the highest temperature resistance, but are stiff, and poor secondary sealing elements compared to 'O' rings ('O' rings should be used where the application allows).

Seal faces

Seal face materials also need to be selected to be compatible with the sealing fluid, however there is another reason. If you recall our discussion on face generated heat, you will remember the term 'f' (coefficient of friction). This term describes the amount of friction between the two face (primary ring and mating ring) materials. Refer to Figure 8.3.9 on material selection for faces.

Two dissimilar face materials are typically used (one softer than the other)
- Carbon vs. silicon carbide – f = 0.1
- Carbon vs. tungsten carbide – f = 0.12

Abrasive services may require the use of two hard faces (tungsten carbide vs. silicon carbide – f = 0.15 or more)
- Note that if fluid has a potential toflash, two hard faces should never be used.

Fig 8.3.9 • Seal face materials

Note the last point on Figure 8.3.9. If the sealing fluid is close to its vapor margin (potential to flash), the last thing you want is to create more heat by having more face friction. Hard face combinations (silicon carbide vs. tungsten carbide) should never be used in a sealing fluid with a high vapor pressure. Carbon vs. silicon carbide has the lowest coefficient of friction value and is recommended for these applications.

Optimal seal plans for various applications

We know the design aspects and how to monitor different flush systems, but when would certain flush systems be preferable over others? Figure 8.3.10 lists the parameters required for a reliable flush source.

Mechanical seal flush must possess the following qualities for optimal seal life:
- Cool
- Clean
- Approximately 345 kPa (50 psi) above vapor pressure (psia)
- Most importantly it must be cost effective!

Fig 8.3.10 • Optimal flush plans for various applications – overview

Therefore, if you can say that the flush system for an application can provide all the qualities (most important is cost effectiveness) listed in Figure 8.3.10, then you have selected the optimal flush system.

The following points outline different seal flushing scenarios, listing considerations specific to each. Note that

HC service with no known solid particles and temperature under 150° C (300°F)
- Assuming satisfactory vapor margin (approximately 345 kPa [50 psi] above vapor pressure) in seal chamber a Plan 11 would be the optimal choice.
- A vertical pump application would require venting of the seal chamber back to suction if possible (Plan 13 or 14).
- A Plan 52, 53 or 54 (53 or 54 preferred) would be recommended in low S.G. (< 0.7) services.

Fig 8.3.11 • Optimal flush plans for various applications: option 1

Clean HC service (e.g. #2 FO) between 150 and 230°C (300 and 450°F)
- Plan 23 is most efficient in cooling, however proper installation and venting is required.
- Plan 21 will be sufficient in most cases and will be easier to operate... note that the orifice sizing is critical here as this determines the velocity of the fluid through the heat exchanger.

Fig 8.3.12 • Optimal flush plans for various applications: option 2

Hot oil service above 230°C (450°F [typically tower bottoms])
- A single seal with the use of a Plan 32 is the most reliable/cost effective if a reliable source is available nearby... note that if the source is a product that has to be reprocessed it may not be cost effective.
- A dual pressurized seal would be the best option (Plan 53 or 54) if a Plan 32 is deemed not feasible.

Fig 8.3.13 • Optimal flush plans for various applications: option 3

Acid service (e.g. H_2SO_4)
- Reliable Plan 32 can be used, however it is very critical for it to be operating at times when pump is installed in field (standby and startup situations).
- A dual pressurized seal, whether contacting (Plan 53/54) or non-contacting (Plan 74) will give the optimal seal life with ease of operation... note that process side seal components will need to be constructed of materials that are corrosion resistant to the particular acid.

Fig 8.3.14 • Optimal flush plans for various applications: option 4

Dirty service (containing suspended solid particles)
- A cyclone separator (Plan 31) can be effective, however it must be sized correctly and particle size must not fluctuate greatly.
- A clean external flush source (Plan 32 for single, Plan 54 for dual) will isolate the seal faces from solids that can cause premature failure due to abrasion.

Fig 8.3.15 • Optimal flush plans for various applications: option 5

these are general, and should not be taken as being definitive for every application; however they can aid in flush plan selection. Ultimately, the seal vendor should be consulted when selecting the seal and flush plan; the more input they have, the higher seal reliability your plant should see as a result.

Best Practice 8.4

All cartridge seals should be statically pressure tested prior to installation to ensure that the seal has not been damaged in transit.

To ensure maximum cartridge seal MTBF, the plant should have a static seal test facility to ensure that the seals are in 'as manufactured condition' at the time of installation.

This best practice is analogous to the low speed balance of a rotor prior to installation to ensure proper operation.

Lessons Learned

Cartridge mechanical seals can be damaged in transit and/or at customs! Static testing of all cartridge seals prior to installation will more than justify the cost of a static seal test facility in your plant.

Benchmarks

This best practice has been recommended since 2000 to ensure maximum cartridge seal MTBFs for new projects, as well as in existing facilities that have been increasing their cartridge seal population.

B.P. 8.4. Supporting Material

See B.P: 8.3 for supporting material.

Best Practice 8.5

Use pressurized dual seals whenever possible in applications that can potentially be hazardous to personnel safety and/or the environment.

The use of pressurized dual seals, which require a safe, non-toxic barrier fluid at a higher pressure than the process fluid at the seal face, will positively ensure that the process fluid will be contained, since a positive barrier fluid to process fluid differential pressure will be maintained by the flush system.

Confirmation of compatibility of the barrier and process fluids is required.

The barrier fluid can be pressurized by any of the following means:

- Plant nitrogen header pressure – if the lowest value of header pressure is sufficient
- Nitrogen bottles and a regulator
- A dedicated barrier fluid seal system

It is recommended that a differential pressure gauge be installed to record the differential pressure between the reference fluid pressure (pressure that is being sealed) and the pressure of the barrier fluid. This will ensure that the proper barrier fluid to seal fluid pressure is maintained, and that the pressure of the barrier fluid is not high enough to force the inner seal open.

Lessons Learned

The use of unpressurized seals in hydrocarbon and/or toxic applications has resulted in flammable, toxic and/or sour fluid releases to the plant environment, which has exposed the plant and personnel to hazards.

Benchmarks

This best practice has been used since the mid-1990s when multiple, unpressurized (tandem) seal failures and process fluid releases were experienced, due to improper operational and monitoring procedures. Since that time this best practice has resulted in optimum plant safety and seal MTBFs (greater than 100 months in some cases).

B.P. 8.5. Supporting Material

Dual unpressurized vs. dual pressurized seals

Today (2010), dual seals are being selected for more and more applications, due to environmental restrictions. There are two arrangements in which dual seals can be used, namely dual unpressurized (previously called tandem) and dual pressurized (previously called double).

Refer to the diagram in Figure 8.5.1, which shows the dual unpressurized arrangement. This type of seal uses a buffer fluid at or near atmospheric pressure to lubricate the atmospheric side seal. The buffer fluid pressure is significantly less than the seal chamber pressure, so any leakage

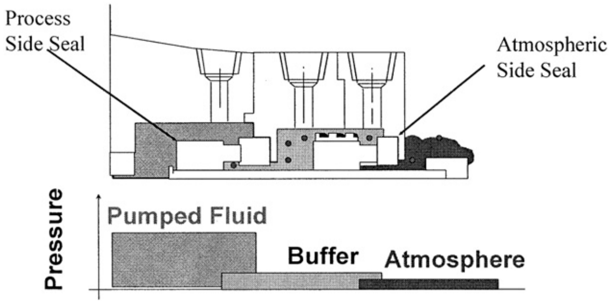


Fig 8.5.1 • Dual un-pressurized

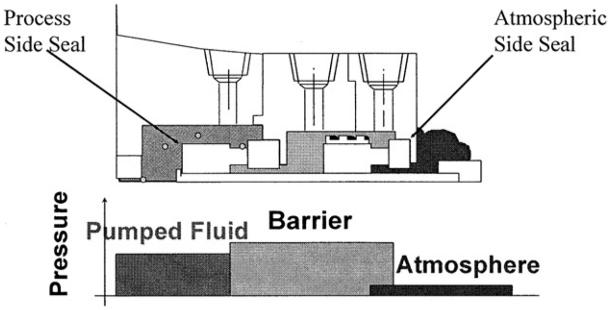


Fig 8.5.2 • Dual pressurized

occurring across the process side seal will leak into the buffer fluid.

This arrangement is very common in applications containing VOCs as it can potentially reduce the leakage to the environment.

Refer to Figure 8.5.2, showing a dual pressurized seal arrangement. These seals use a barrier fluid (same fluid as dual un-pressurized) at a pressure that is 175 kPa (25 psi) above seal chamber pressure to lubricate the seals. Since the barrier fluid pressure is higher than the seal chamber pressure, any leakage occurring at the process side seal will enter the pump. This arrangement (when working properly) ensures no leakage of the pumped fluid to atmosphere. Note that this arrangement requires the barrier fluid to be compatible with the pumped fluid, since the barrier fluid leaks into the pump. Refer to Figure 8.5.3.

Dual Un-pressurized	Dual Pressurized
■ Primary leakage is process fluid, therefore the secondary seal is essentially a backup seal ■ Does not require the use of nitrogen or other inert gases	■ Primary leakage is barrier fluid into the process, therefore the process must be able to accept a small amount of barrier fluid ■ Requires use of nitrogen (or other compatible inert gas) to pressure seal reservoir approx .175 kPa (25 psig) above seal chamber pressure in a Plan 53 ■ A Plan 54 uses an external fluid (synthetic skid or another pump) to lubricate the seals at a pressure of 175 kPa (25 psi) above seal chamber pressure

Fig 8.5.3 • Dual un-pressurized vs. dual pressurized

It has been our experience that if an inert gas (or external fluid) is available at the required pressure and the pumped fluid can accept the barrier fluid (compatible), a dual pressurized seal is potentially more reliable in VOC service.

Best Practice 8.6

Require that a pressure gauge is installed in the seal chamber or the flush line immediately before the seal chamber to ensure optimum mechanical seal MTBF.

Maintaining the proper seal chamber pressure above the vapor pressure is a key factor in obtaining maximum mechanical seal MTBFs.

Most seal applications do not use a pressure gauge to continuously monitor seal chamber pressure.

It is recommended that this best practice be specified for all hydrocarbon seal applications and in other applications where 'bad actor seals' (more than one seal failure per year) have been experienced.

Lessons Learned

Failure to identify seal chamber pressure caused by the following issues has resulted in mechanical seal MTBFs far below expected values (lower than 12 months):

- Blocked flush line orifice
- Eroded or missing flush line orifice
- Plugged cyclone separator
- Plugged flush line strainer
- Worn throat bushing

Benchmarks

The installation of seal chamber pressure gauges has been recommended for all projects since 1990, especially for all 'bad actor' seals (more than one seal failure per year).

B.P. 8.6. Supporting Material

Pressure monitoring

Unless the plant assigned machinery specialist is 'world class' there are usually no pressure gauges in the flush line or seal chamber. Our recommendations concerning seal chamber pressure monitoring are presented in Figure 8.6.1. The measured seal chamber pressure must be compared to the seal vendor's assumed value, which should be on the seal layout drawing. If this value is not present, the seal vendor should be consulted. Note that the seal vendor assumed seal chamber pressure is the value that was used in the calculation of the seal PV. If the measured value does not agree with the data sheet, consult with operations first to see if process changes can be made. If process changes cannot be made, discuss this fact with the seal vendor representative.

- Install a pressure gauge in the seal chamber
- Modify the seal flush line for installation of a pressure gauge

Fig 8.6.1 • Seal chamber pressure monitoring guidelines for 'bad actor' seals

Best Practice 8.7

Perform checks of all associated seal systems every time that the mechanical seal is replaced, to optimize safety and MTBF.

Like bearings, the reliability of mechanical seals is a direct function of the reliability of all of its support system components.

To ensure that the root cause of a mechanical seal failure was not caused by a flush system component, all of the components in the flush system should be checked each time a mechanical seal is replaced.

As a minimum, the following items should be checked:

- The flush line orifice for proper dimensions and cleanliness
- The flush line does not contain any restrictions (especially where flush line tubing is used and may be crimped)
- All flush line components (cooler, strainer, cyclone separator, etc.)
- The gland plate to ensure all ports are fully open and do not contain any debris
- Throat bushing clearances

Lessons Learned

Failure to completely check the flush line, gland plate ports and all flush line components during a mechanical seal replacement has resulted in low seal MTBFs and has exposed plant personnel and assets to damage.

The root cause of many repeated mechanical seal failures has often been eventually found in the flush system.

Benchmarks

This best practice has been used since 1990. Since that time, this recommended maintenance procedure has resulted in optimum mechanical seal safety and reliability for all projects and field mechanical seal reliability optimization audits.

B.P. 8.7. Supporting Material

The importance of seal face lubrication

In order to understand the functions and design parameters of mechanical seal flush systems one must first understand the need for these flush systems. Refer to Figure 8.7.1, outlining the importance for seal face lubrication.

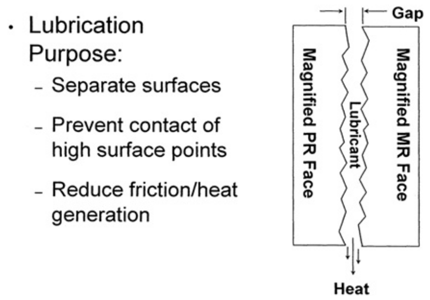


Fig 8.7.1 • Importance of seal face lubrication

As is the case with hydrodynamic bearings, a mechanical seal consists of a rotating part (shaft or thrust disc) and a stationary part (shoes or brg housing). Contact between these two, just like rubbing your hands together, creates heat generated by friction. Therefore, some type of lubricant (whether it be the pumped fluid or an external fluid) is required to keep a gap between the two surfaces to keep them from contacting. This gap (created by lubrication) results in a reduction of friction (or heat) generated between the two faces.

The fluid used to lubricate and take the heat away from the seal faces must have certain characteristics to ensure optimal seal face life. Refer to Figure 8.7.2.

- Vapor pressure – the fluid should be at least 345 kPa (50 psi) above its vapor pressure
- Viscosity – typical limitations are no more than 100 Cst at minimum pump temperature and no less than 1 Cst at highest pump temperature
- Cleanliness – fluid shall contain little to no suspended solids

Fig 8.7.2 • Sealing fluid characteristics

The most critical of these characteristics is the fluid's vapor pressure. The sealing fluid needs to be 345 kPa (50 psi) above its vapor pressure (at the pump's operating temperature) in the seal chamber to ensure it does not vaporize too early across the seal faces. Seal vendors refer to this value as the 'vapor margin'. It is a great concern for fluids that have a specific gravity of less than 0.7 at the pump's operating temperature, as they will have a higher vapor pressure.

If the pressure and/or temperature within the seal chamber changes during operation (this could be the result of operating in a region of the centrifugal pump curve where vaporization can occur), the vapor margin will become smaller. Therefore it is essential to know the conditions within the seal chamber during process changes. Figure 8.7.3 represents estimated seal chamber pressures for different pump configurations.

- Single stage overhung with balance holes – $P_{SC} = P1 + (0.15 \times (P2 - P1))$
- Single stage overhung without balance holes – $P_{SC} = .8 \times P2$
- Double suction – $P_{SC} = P1$
- Multistage with balance drum and new bushing – $P_{SC1} = P1$ $P_{SC2} = P1 + 525$ kPa (75 PSI)
- Multistage with balance drum and old bushing – $P_{SC1} = P1$ $P_{SC2} = P2$
- Multistage opposed impellers with new bushing – $P_{SC1} = P1$ $P_{SC2} = P1 + 525$ kPa (75 PSI)
- Multistage opposed impellers with old bushing – $P_{SC1} = P1$ $P_{SC2} = P1 + (0.5 \times (P2 - P1))$
- Multistage vertical with bleed off – $P_{SC} = P1 + 525$ kPa (75 PSI)
- Multistage vertical no bleed off – $P_{SC} = P2$

Fig 8.7.3 • Seal chamber pressure estimations

In Figure 8.7.3 P_{SC} is seal chamber pressure, P1 is pump suction pressure, and P2 is pump discharge pressure.

The calculations shown are estimates, and should be verified accurately before consulting the vendor about bad actor seals. The best way to confirm the seal chamber pressure, as will be discussed later, is simply by installing a pressure gauge in the seal chamber. If ports are not available to do so, the pump OEM should be contacted as they can give an accurate estimate based on the pump being in good condition.

In addition to the fluid characteristics listed in Figure 8.7.2, the specific heat (C_P) of the sealing fluid also has an effect on seal life. The specific heat describes how much heat is needed to increase the temperature of a fluid by one degree. Therefore, a fluid with a higher specific heat would be affected less (temperature won't increase as much) than a fluid with a lower specific heat in the same conditions.

All of the fluid characteristics, along with heat generation, influence the amount of seal fluid flow to the seal faces that is necessary. Take a look at Figure 8.7.4, which shows the equation used to determine the minimum required flow to the seal faces.

The amount of flow required (M) to the seal faces is directly proportional to the heat load (Q) and inversely proportional to the fluid's specific heat, specific gravity, and desired temperature rise in the seal chamber. Note that the heat load used in determining the flush flow required is the total heat load. This

$M = Q/(500 \times C_P \times SG \times \Delta T)$

- Where M (LPM or GPM) is the seal fluid massflow required
- Where Q (BTU/hr) is the total heat load in the seal chamber
- Where S.G. is the specific gravity of the seal fluid
- Where ΔT is the desired temperature rise of the sealingfluid (this varies depending on the fluid characteristics)
- 500 is a conversion factor to convert the flow to GPM (125 used for LPM)

Fig 8.7.4 • Equation for seal flow

not only accounts for the seal face generated heat but also adds any heat generated via heat, or conduction through the pump casing. An example where heat soak needs to be taken into account is a BFW service. In this service the heat from the pump fluid is transmitted through the shaft to the seal. In addition, heat will also be transferred through the casing to the seal fluid. The heat soak value is an estimate, and varies between seal vendors. It typically is negligible in the calculation in Figure 8.7.4 except in applications above 177°C (350°F).

Considerations for process flush systems

Now that we understand why it is so important to lubricate mechanical seal faces and what fluid characteristics need to be considered for optimal seal life, we will discuss the design considerations for all the major mechanical seal flush plans. First, we will take a look at the flush plans categorized as process flush plans, which utilize the pumped fluid to lubricate the seal faces.

API Plan 11

The most commonly used flush plan, an API Plan 11 flush, utilizes the pumped fluid to lubricate the seal faces. The pumped fluid is taken from discharge and sent to the seal chamber through an orifice. Refer to Figure 8.7.5 for a flush plan schematic.

The orifice is used to control the flow of the pumped fluid above the minimum required flow rate. Seal vendors require an orifice to ensure the flow is not too great either, as high flow rates can cause erosion of the seal faces. Equally important, is the fact that an orifice limits the amount of recirculation through the seal chamber back to the pump (the pump pumps money). A 3 mm (1/8") orifice is the most commonly used size, as it is the smallest practical size and Plan 11 flushes are normally used in services that are easy to seal (good lubricating qualities). Refer to Figure 8.7.6, outlining general guidelines for orifice sizes.

Remember that Figure 8.7.6 just shows guidelines for orifice sizes that are not always followed, but if followed they should not harm the seal if the pump is operating in a region of the centrifugal pump curve where vaporization can occur. Note that orifice sizing does not give an exact flow, due to the system friction (piping, coolers, etc.), therefore the more information about the system the seal vendor has, the more accurately they can size the orifice.

A vendor may require a close clearance throat bushing be installed in certain instances to increase the pressure in the seal chamber above the vapor pressure of the pumped fluid. As the

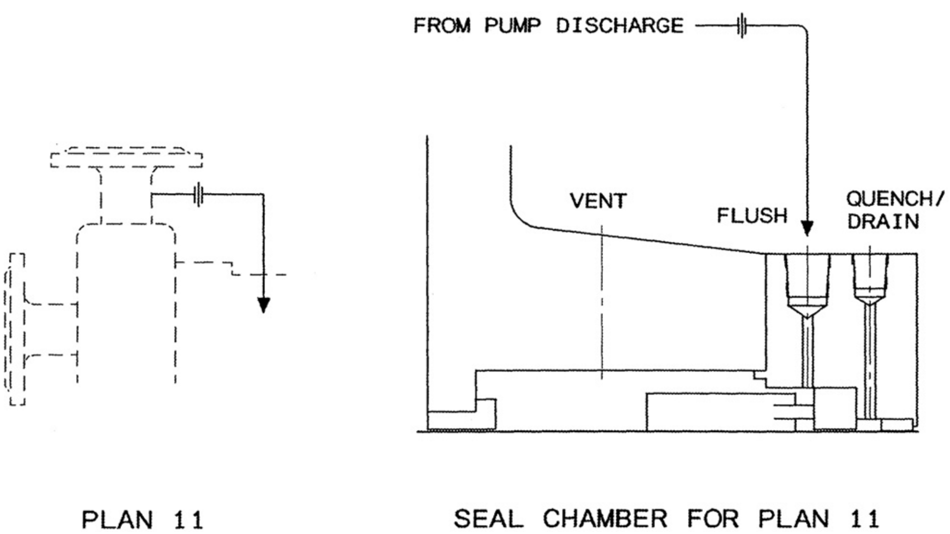


Fig 8.7.5 • Plan 11 – recirculation from discharge through orifice and to mechanical seal

- Single stage pumps – 1/8"
- Multistage pumps – gang orifices (series of orifices back to back, sized by seal vendor)
- Low S.G. fluids may require higher flow rates to prevent flashing (early vaporization) where a 3/16" orifice may be necessary

Fig 8.7.6 • Orifice size guidelines – where used

throat bushing wears over time, the seal chamber pressure will drop.

In a Plan 11 flush system, the main thing to monitor is the temperature across the orifice. If this drops by more than 10%, the orifice is most likely plugging up. If this is the first occurrence and the fluid does not normally contain solids, the best option would be to clean the orifice out quickly and continue to check the temperature drop after the pump is restarted. If the orifice has plugged up more than once, another flush plan option should be considered. Refer to Figure 8.7.7.

- Monitor temperature across the orifice
- If temperature decreases by more than 10% it is beginning to plug
- If first occurrence, clean and continue to monitor
- If problem reoccurs, consider using a different flush plan

Fig 8.7.7 • Orifice monitoring

Look at Figure 8.7.8, which shows a typical Plan 11. This is a between bearing double suction pump with an orifice to each seal.

API Plan 13

An API Plan 13 is widely used in vertical pump applications, or when seal chamber pressure is at or near the discharge pressure of the pump. This flush plan basically vents the seal chamber at

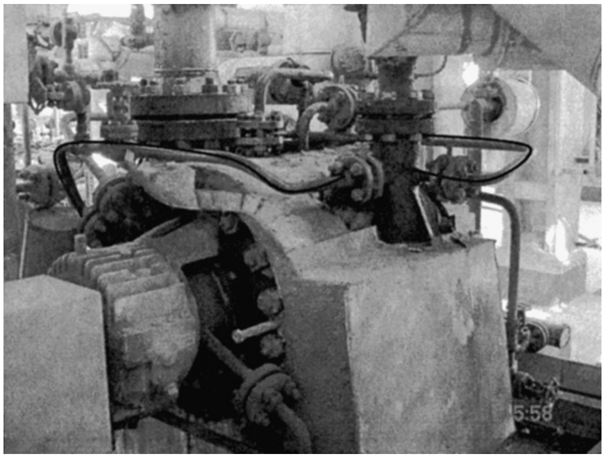


Fig 8.7.8 • Typical API Plan 11

a high point back to the suction of the pump (ideally a high point in the suction piping). Refer to Figure 8.7.9 for a schematic of a Plan 13 flush.

As with a Plan 11, this flush plan also utilizes an orifice, however it is more to create a back pressure on the seal chamber than to control flow. The orifice in a Plan 13 flush is typically 6 mm (¼"), which is usually large enough to vent vapors accumulated in a vertical pump, while keeping the vapor margin at 345 kPa (50 psi).

Since a Plan 13 uses a larger orifice, it is not typically monitored with the frequency of a Plan 11. From time to time, however, it is a good idea during rounds to check the temperature across the orifice as in a Plan 11, to ensure flow out of the seal chamber and no plugging of the orifice.

Very commonly, a Plan 13 will be used in conjunction with a Plan 11; this is defined by API as a Plan 14 flush. The same monitoring rules apply to a Plan 14 as to flush Plans 11 and 13.

API Plan 21

A flush Plan 21 is a Plan 11 with the addition of a cooler to lower the temperature of the pumped fluid. This plan is generally used

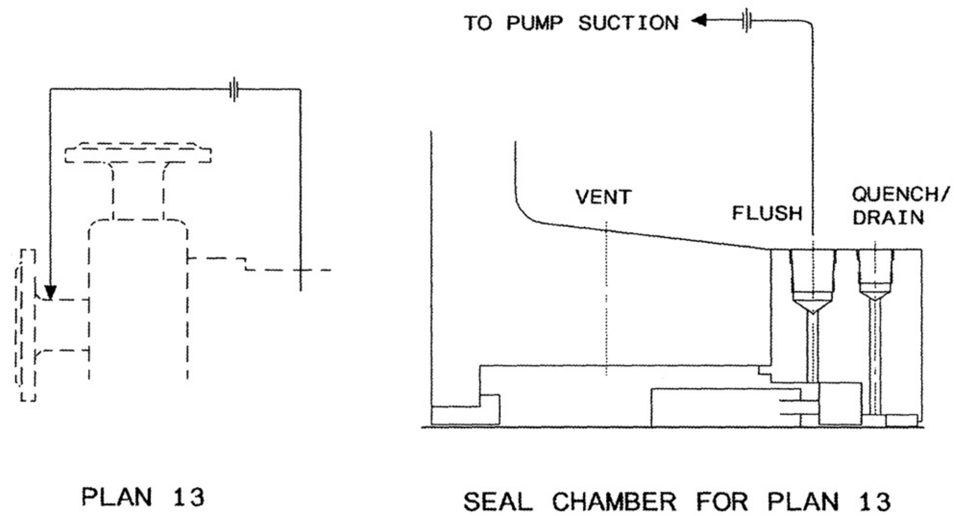


Fig 8.7.9 • Plan 13 – recirculation from seal chamber through orifice and back suction

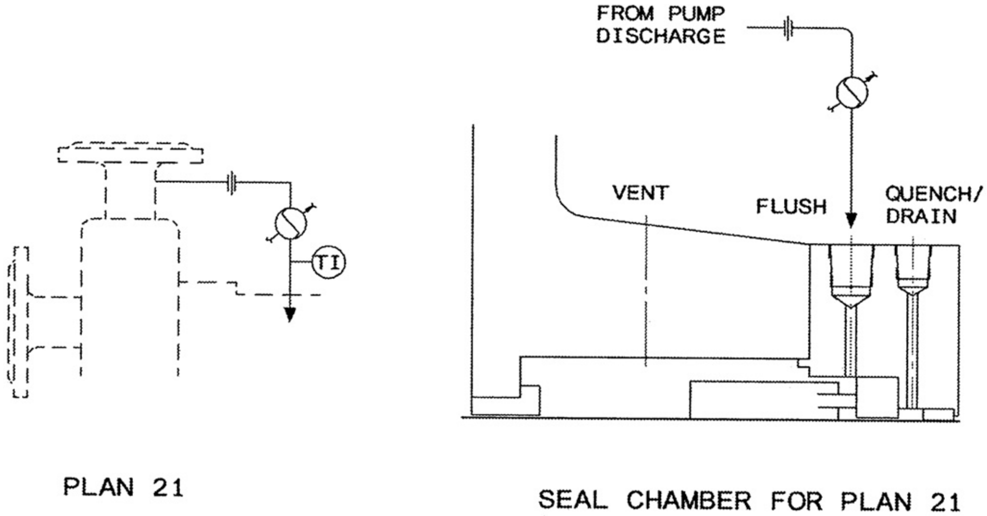


Fig 8.7.10 • Plan 21 – recirculation from pump discharge through an orifice and heat exchanger to the mechanical seal

when the pumped fluid is naturally a good lubricant, but the vapor margin is low at the current pump operating temperature, and it therefore requires the addition of a cooler to increase the vapor margin. Refer to the schematic of a Plan 21 in Figure 8.7.10.

Just like a Plan 11, a Plan 21 utilizes an orifice to control the flow of the pumped fluid. In this plan, orifice sizing is more critical, since it is preferred to have a flow that is close to the minimum required for maximum cooling of the pumped fluid. It has also been found to be beneficial to place this orifice downstream of the cooler (especially in low S.G. fluids) to prevent vaporization before the cooler.

With the addition of the cooler, it is essential to check the temperature differential across it. This should be done at initial installation (or after cleaning of the cooler) and trended on a time basis. Typical seal flush coolers should provide a temperature drop greater than 38°C (100°F), however this depends on the cooling media temperature, showing why trending is important. If a water-cooled heat exchanger is being used, cooling water (CW) temperatures should also be checked regularly, as a decrease in cooling efficiency could be the result of CW temperature or flow changes. It is ideal to have a thermometer installed after the cooler to allow easy check of the seal fluid temperature. A thermometer is recommended by API as an option, and if possible, FAI recommends it be installed on a Plan 21. The pump operating conditions need to be considered as they could alter the cooler inlet temperature.

Refer to Figure 8.7.11. This pump is utilizing a process flush with a cooler, before making its way to the seal. Note that this installation also includes a component called a cyclone separator. This is used in services that contain suspended solids, and it works by sending the heavier solids back to the suction of the pump while the 'clean' liquid goes to the seal chamber. Our experience with cyclone separators has been one of frustration at times. It is essential that it be sized correctly, and that the solids are significantly heavier (and don't change size) than the pumped fluid. If not, the solids will carry over to the seal and may even potentially plug the cyclone

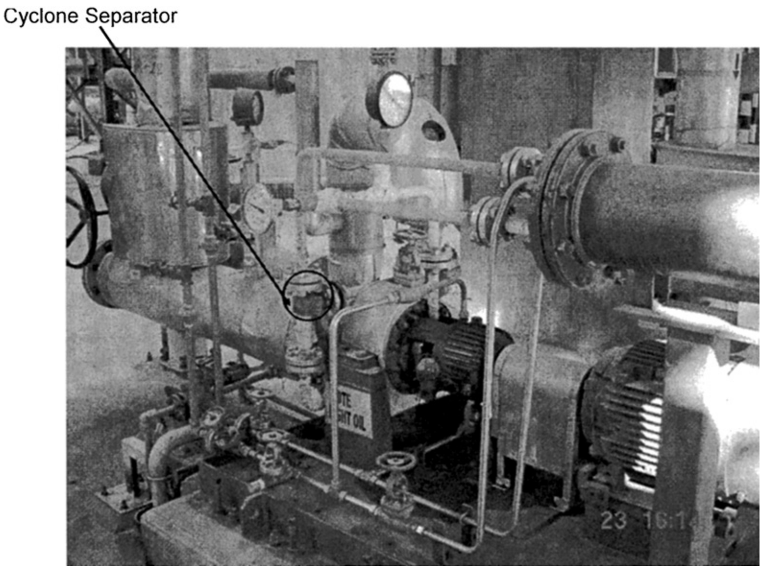


Fig 8.7.11 • (Cyclone Separator) API Plan 41

separator, preventing seal face lubrication. In addition, the solids that cause heavy seal face wear are smaller than 2 microns (typical gap between seal faces), which a cyclone separator cannot separate from the pumped fluid. With that said, when a cyclone separator is properly sized it can reduce the potential of seal hang-up due to solids building up on the springs or dynamic 'O' ring.

The picture shown above is an example of an excellent flush plan installation. As you can see, a pressure gauge and thermometer are installed in the piping just before entering the seal. These instruments give you an idea of the flush system performance (and aid in troubleshooting failures) by allowing the seal chamber pressure and temperature to be monitored easily (vapor margin!).

Considerations for external flush plans

When the pumped fluid contains significant amounts of solids, is very corrosive, or is at a very high or low temperature, an external flush should be considered. An external flush plan uses a fluid from another pump, or it may be a separate console with a process compatible liquid to lubricate the seal faces. We will now discuss two types of external flush plans; a Plan 32 and a Plan 54.

API Flush Plan 32

A flush Plan 32 utilizes a fluid from another pump with good lubricating qualities (cool, clean and at an acceptable vapor margin) at a higher pressure than the seal chamber (by at least 66 kPa (10 psi), to keep the pumped fluid out of the seal chamber. Refer to Figure 8.7.12.

Since the Plan 32 is at a higher pressure than the seal chamber, the Plan 32 fluid will leak into the pump, therefore it must be compatible with the pumped fluid and at an acceptable vapor margin. The leakage into the pump is controlled by a bushing located either in the seal itself or the pump casing known as the throat bushing.

For optimal seal life with a Plan 32 flush (very common to exceed 3 years), the flow is very critical. A typical value used by seal vendors is 3 LPM (0.75 GPM) per inch of seal size, however it is essential to ensure the minimum required flow is met, while minimizing the flow into the pump. Using the equation from Figure 8.7.4 and the Plan 32 fluid qualities, the seal vendor will calculate the minimum required flow for the seal. This flow can be controlled using a variety of methods (usually manual). Refer to Figure 8.7.13, which describes the methods for controlling the flow to the seal in a Plan 32.

Although they can be somewhat expensive, a flow meter installed in conjunction with a throttle valve is a very accurate way of controlling the flow to the seal. No matter which method is used, operator training on the method is important in assuring a reliable system.

A flush Plan 32 can provide very long seal life, however it always needs to be justified. A compatible fluid (with the process and with vapor margin) is required and it must be provided at all times from start-up through to shutdown of the pump. Also, if the selected fluid is a product of the plant, long term costs (reprocessing a product because the product is not compatible with the pumped fluid) may need to be considered.

API Plan 54

A flush Plan 54 is an external flush used on dual pressurized (double) seals. This flush plan is typically used in applications with very corrosive fluids, or when a Plan 32 is not feasible. Refer to Figure 8.7.14 for a schematic of a Plan 54 flush.

A Plan 54 can use a fluid from another pump in the plant (process Plan 54), like a Plan 32, or it can be a separate console with process compatible liquid with sufficient vapor margin (synthetic Plan 54). The advantage of a separate console is that a potential product of the plant is not used, therefore long-term costs for reprocessing need not be considered. The synthetic Plan 54 can potentially produce the longest seal life, but the up-front cost (capital expenditure) of this flush plan needs to be justified.

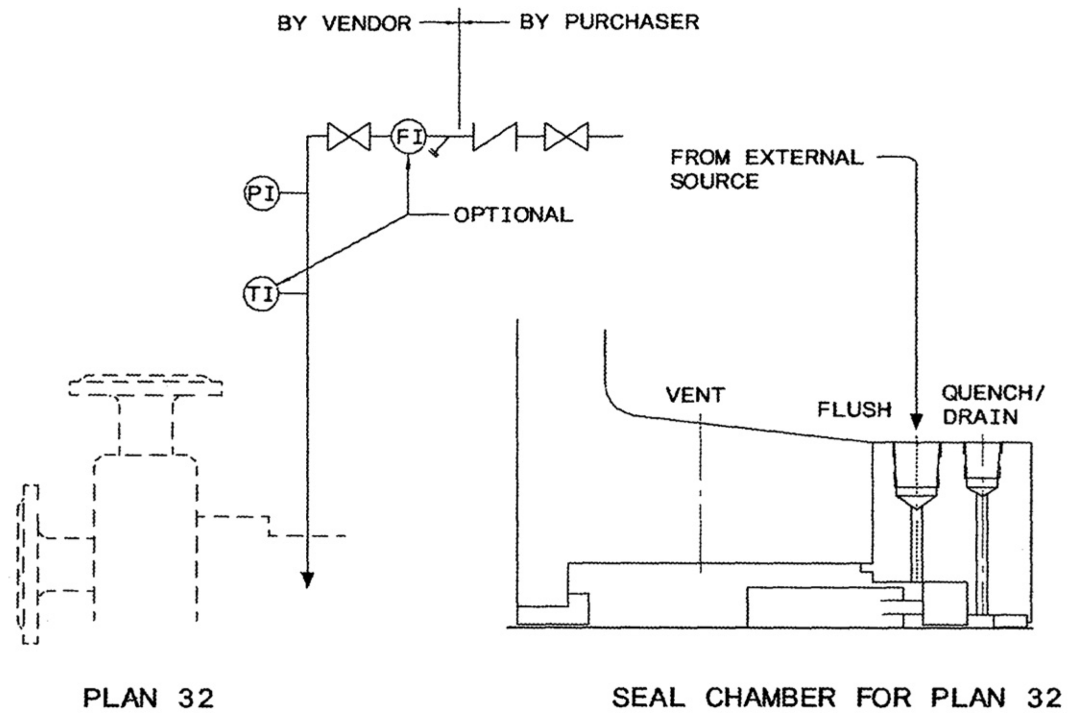


Fig 8.7.12 • Plan 32 – clean seal flush from external source

- Orifice – not very accurate, since all system losses need to be considered
- Throttle valve and pressure gauge – can be effective if seal chamber pressure does not fluctuate much
- Throttle valve and flow meter – most effective manual flow control

Fig 8.7.13 • Manual flow control options for API Plan 32

As with a Plan 32 flush, the seal vendor will calculate the minimum flow required for the seal (using the heat generated from two sets of faces). This flow is used in sizing the seal pump (value can be from 24 to 40 LPM [6 to 10 GPM]). Since the flow is not going directly to the seal chamber (through the bushing into the pump) but is a 'through' flow back to the seal reservoir, the pressure is controlled at 175 kPa (25 psi) above the seal chamber pressure. This ensures the Plan 54 fluid is pushed across the faces, providing adequate lubrication. A process Plan 54 will most likely use a throttle valve and pressure gauge to set the pressure, while

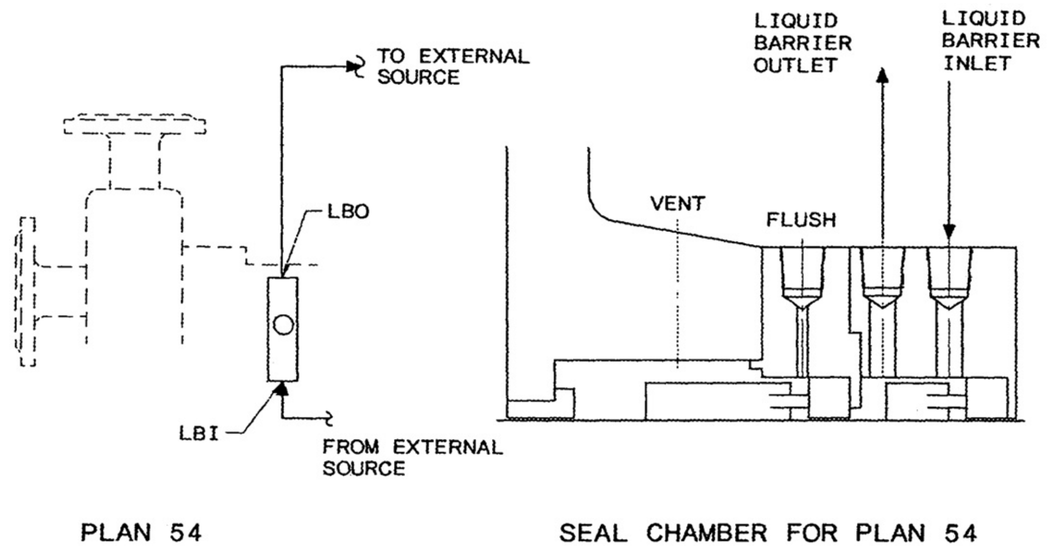


Fig 8.7.14 • Plan 54 – dual pressurized seal using external source to lubricate both seals (source can be a process fluid, or synthetic oil, which is preferred)

a synthetic Plan 54 is usually provided with a pressure control valve to automatically control the pressure. Note that pressure control is adequate in most applications, however if the seal chamber pressure fluctuates during operation, a differential pressure control should be considered.

Another consideration when making a decision on a synthetic Plan 54 vs. a Process 54 or a Plan 32, is the potential addition of extra components. A synthetic Plan 54 will contain pump(s), coolers, filters, and instrumentation. As everybody knows, the most reliable equipment has a balance of quality components and is as simple as possible (fewest number of components).

Considerations for closed loop system

API Flush Plan 23

A flush Plan 23 is used on single seals, and consists of piping (usually tubing) connected to the seal and a cooler. The fluid (seal chamber is filled with the pumped fluid) is circulated to the cooler and back to the seal (see Figure 8.7.15).

This flush plan offers more efficient cooling than a flush Plan 21, since it does not have to continually cool down the pumped fluid. This can be related to your car air conditioning; most cars today have a button for recirculation of air. This closes the valve to the atmosphere, so that the A/C just has to cool the air in the car, not the air pulled in from outside. A Plan 23 works in the same fashion. This, however, creates some concerns, because it relies heavily on the pumping ring to circulate the fluid to the cooler. High system friction or vapor pockets (not vented properly) will result in limited to no circulation, due to the limited pumping capability of the pumping ring.

Therefore, the cooler needs to be placed close to the seal (3 meters of total tubing and 7–10 cm above the seal [10 feet of total tubing and 18–24" above the seal]) to ensure the pumping ring will be sufficient. The cooler needs to be above the seal to allow for venting and flow back to the seal (through gravity and thermosyphon). It is also essential in a Plan 23 to have a high point vent and block valve installed, to allow for proper venting before pump start-up. Operators should be trained to understand the necessity of venting these systems.

In monitoring this system, the temperature drop needs to be checked across the cooler. Note that this temperature drop will be less than in a Plan 21, due to the cooler not needing to provide the same amount of cooling. If the temperature drop across the cooler is high (like a Plan 21 should be), it indicates that the throat bushing clearance has increased and mixing of the pumped fluid and seal fluid is occurring. Typical temperature drop in a Plan 23 can be anywhere from 7°C to 10°C (20°F to 50°F).

This flush plan is most common in BFW applications, since water has poor lubricating qualities and needs to be as cool as possible. Refer to Figure 8.7.16, which shows a flush Plan 23.

The installation shown in Figure 8.7.16 is once again excellent as the location of the cooler is very close to the seal, giving

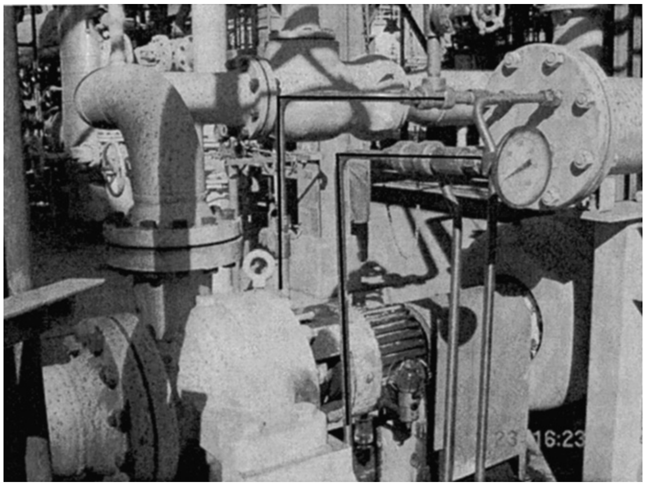


Fig 8.7.16 • Plan 23 flush

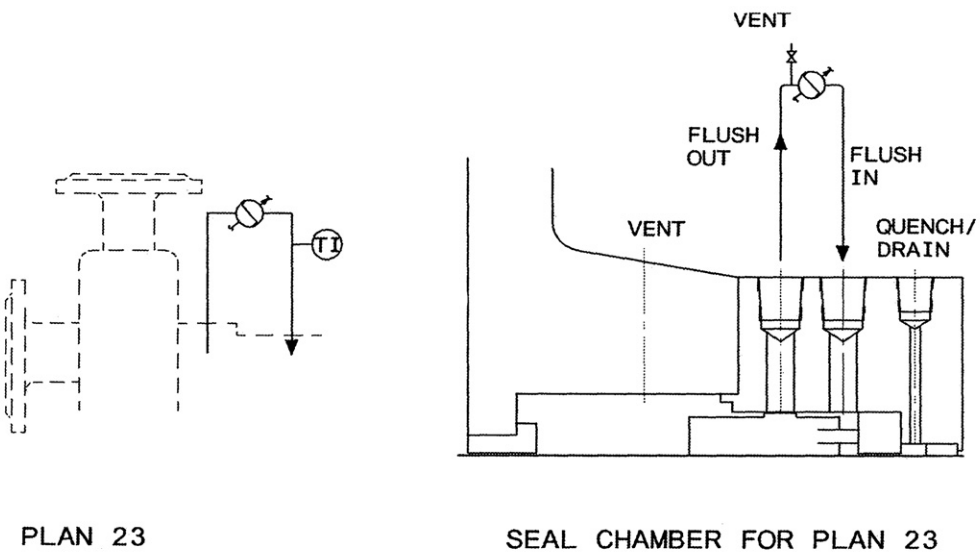


Fig 8.7.15 • Plan 23 – closed loop circulation of process fluid through a heat exchanger via a pumping ring

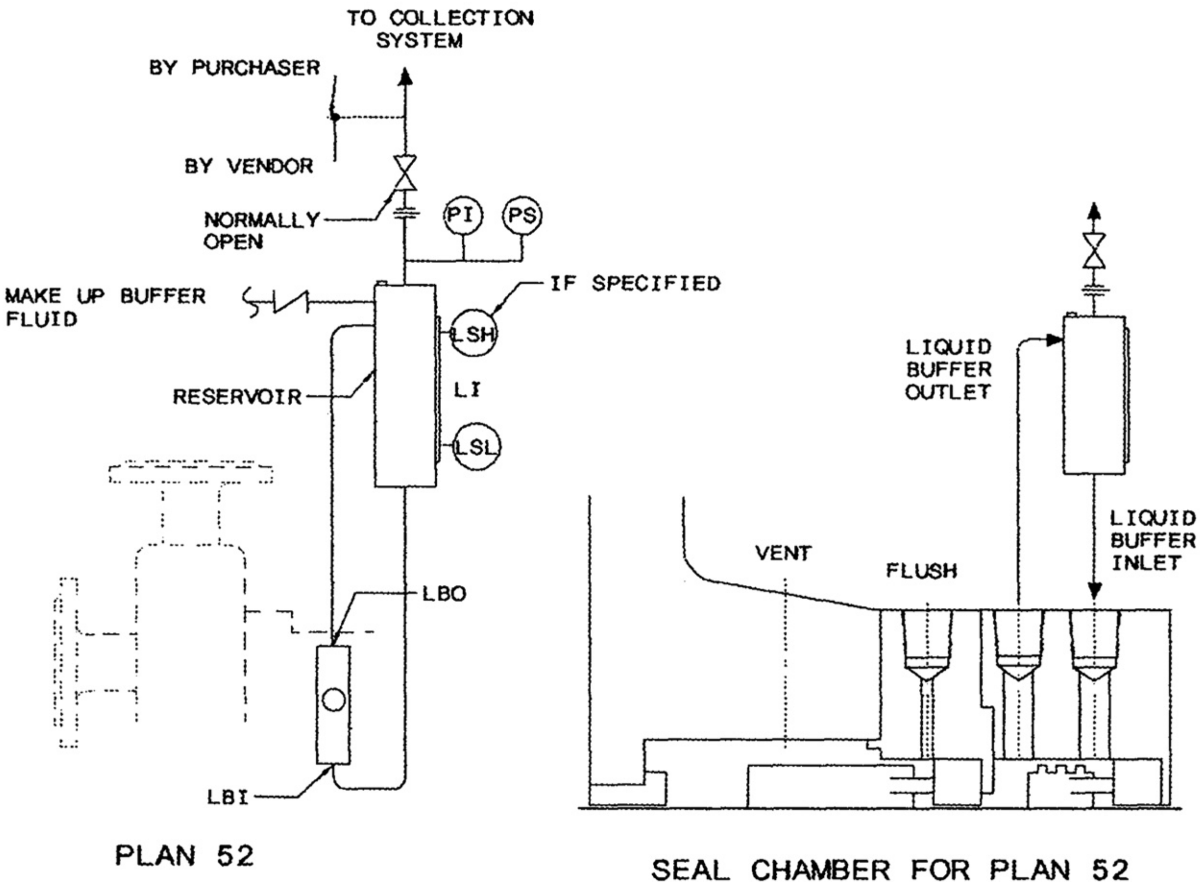


Fig 8.7.17 • Plan 52 – dual un-pressurized seal using synthetic buffer fluid to lubricate the atmospheric side seal. A pumping ring in the seal circulates the buffer fluid (pressure less than seal chamber) to the reservoir.

the pumping ring as little system resistance as possible. Also, the vent is in its proper location and a thermometer is installed in the return line to the seal for easy cooler and throat bushing condition monitoring.

API Flush Plan 52

Dual un-pressurized seals (tandem) rely on a buffer fluid at or near atmospheric pressure to lubricate the atmospheric side seal. This buffer fluid is circulated via a pumping ring from the seal to the seal reservoir and back to the seal (in a closed loop). Take a look at Figure 8.7.17 for a schematic.

This flush plan is very common in applications with VOCs (Volatile Organic Compounds), however if not set up similarly to the schematic in Figure 8.7.16 it may not be effective.

The reservoir is at atmospheric pressure (less than seal chamber pressure), so the leakage across the process side seal faces migrates into the seal reservoir, and will either increase pressure, level, or both in the reservoir. Since every seal does leak a certain amount, it is essential to have the reservoir vented to a flare or vapor collection system. If the reservoir is allowed to reach the seal chamber pressure, the atmospheric side seal will most likely fail (if it hasn't already) as it is not typically designed to handle seal chamber pressure. If this is a concern in the plant, you may want to consider requesting the seal vendor to redesign the atmospheric seal to handle maximum seal chamber pressure. In addition, as the process side seal leaks in this flush plan, the atmospheric side seal will essentially be sealing the pumped fluid, exposing the plant to the release of flammable and/or toxic vapors.

Monitoring of seal leaks can be done by checking the level and pressure of the reservoir, as one or both may increase in the event of excessive leakage. The seal vendor (or support system vendor) may supply high level and/or pressure switches which would alert the operators to a seal leak. It is highly recommended to specify this instrumentation in new projects, as it will cut down on the already high workload of operators (if the alarm sounds then the level or pressure can be personally verified in the field).

In addition to checking for excessive leakage (pressure or level increase), temperature in and out of the seal can help verify proper circulation. In a reservoir that is not cooled, there should be a temperature drop of approx. 1 to 2°C (5°F to 10°F) from the seal outlet to the seal inlet. If there is no temperature drop (or if the temperature increases), this indicates zero or possibly reverse circulation.

It has been our experience that Plan 52 flushes may not be vented properly (blocked in) and level may be low or at zero in the reservoir. For these reasons, it is very important for the operators to be trained to understand the necessity of monitoring this system.

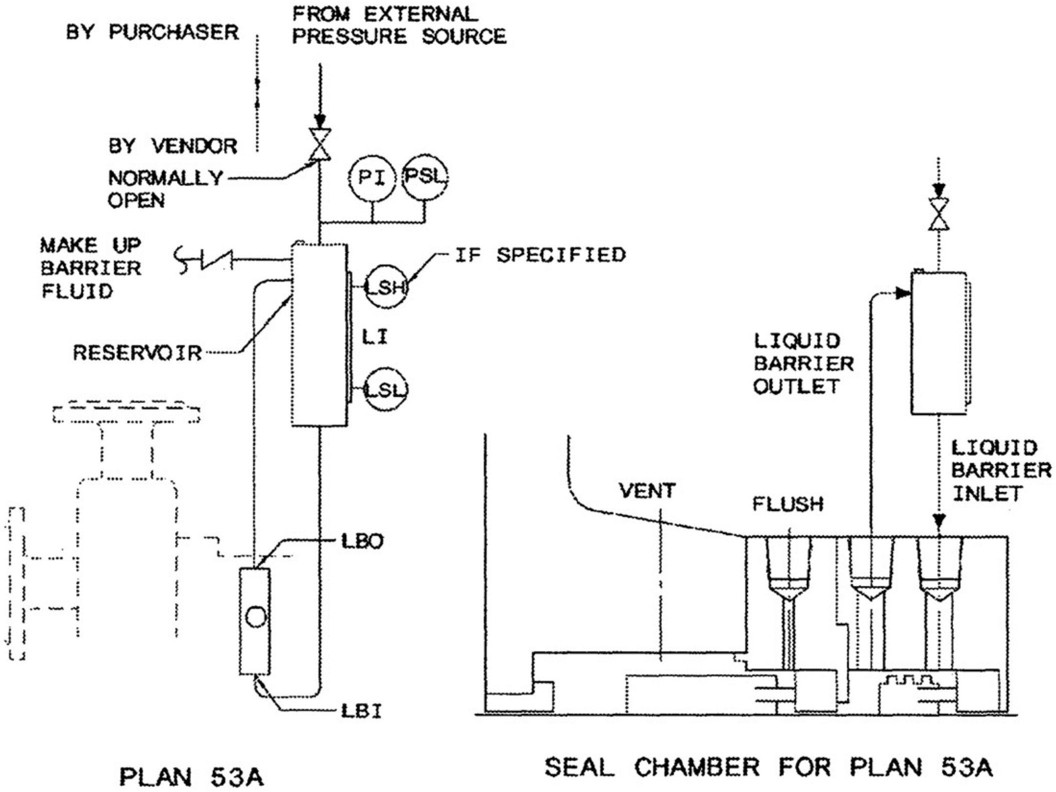


Fig 8.7.18 • Plan 53 – dual pressurized seal using a pressurized barrier fluid (usually pressurized by a nitrogen blanket) to lubricate the seals

API Flush Plan 53

A Plan 53 is basically a combination of a 52 and 54. It uses the same reservoir as a Plan 52, however it is pressurized at 175 kPa (25 psi) above the seal chamber pressure, just like a Plan 54. Refer to Figure 8.7.18.

The reservoir is pressurized, however the system holds a constant pressure of 175 kPa (25 psi) above seal chamber pressure, hence requiring a means of circulation (pumping ring). The pumping ring circulation can be monitored the same way as in a Plan 52.

Also, as in a Plan 52, pressure and level switches are recommended to set off an alarm on excessive leakage. Since this system is at a higher pressure than the seal chamber, leakage will migrate into the pump. A low pressure or level in the reservoir will indicate this excessive leakage.

Considerations for quench

A quench (known as an auxiliary flush plan) is a flush plan that uses a medium (steam, nitrogen, or water) on the atmospheric

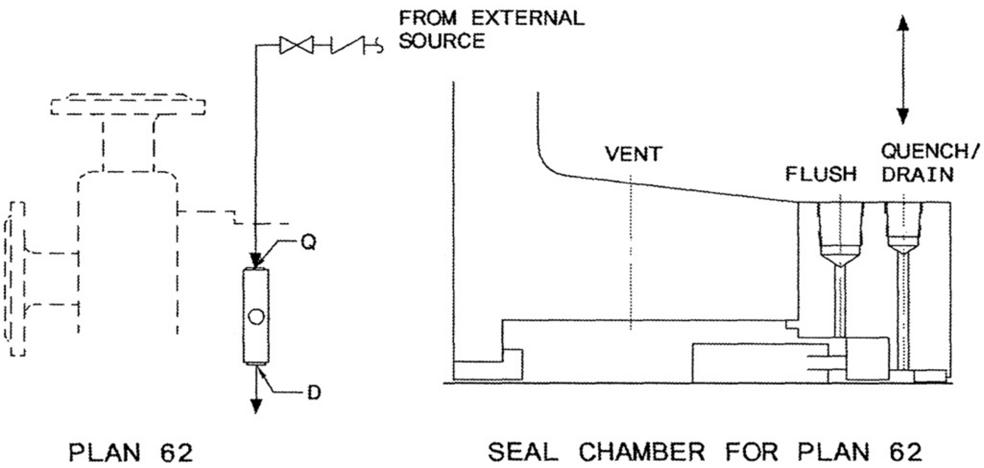


Fig 8.7.19 • Plan 62 – a medium (usually steam or water) is introduced on the atmospheric side of the seal to prevent crystallization or buildup of solids

side of the seal to wash away any solid buildup from the faces. The buildup is drained to a collection system. Refer to Figure 8.7.19 for a schematic.

A quench is most commonly used in single seals with steam as the medium, if the seal fluid is hot and can form coke particles and/or if the seal fluid is flammable or toxic. Note: many countries today require dual (tandem or double) seals if the seal fluid is flammable or toxic, in order to meet environmental requirements. The steam should be regulated to a pressure of approx. 20 to 33 kPa (3 to 5 psi), which is just enough to wash the accumulated solids off the atmospheric side of the faces. It is essential for the steam to be superheated (dry), to prevent flashing of water at the faces, causing premature failures and to ensure that moisture does not enter the bearing chamber. We have experienced plant fires that resulted from water contamination in the bearing housing, leading to a hot bearing, which served as an ignition source for a single seal leaking a flammable vapor. We recommend that an 'oil condition bottle', to monitor water in the oil, always be installed in the bearing housings when a steam or water quench is used. Finally, refer to Figure 8.7.20, highlighting the importance of operator training on flush systems.

- Using information in this section (or other source), discuss parameters to monitor on different flush plans
- Walk to the equipment and monitor these parameters
- Discuss action plans to be executed to increase reliability of these systems
- Remember, operators see this equipment every day; the more they understand about flush system, the more reliable the seals will be

Fig 8.7.20 • Flush system training for operators

Best Practice 8.8

Use medium pressure steam in seal jackets for hot pump (above 300°C) services to cool the seal fluid during operation and keep standby pump seal fluid warm.

Most hot pump services (bottoms and gas oil refinery services) use bellows seals (to eliminate the dynamic secondary seal) and dead ended (no flush) configurations to minimize ingress of fluid particles into the seal chamber.

As a result, seal chamber jacket cooling is required during operation.

Medium pressure steam in the seal chamber jacket has proven to be the best solution in this case, since it will provide adequate cooling during operation and keep the seal fluid viscosity in the standby pump (frequently on auto-start in these applications) at an acceptable level to prevent excessive seal wear during start-up and operation.

Lessons Learned

Low seal MTBFs have been experienced in hot services (bottoms and gas oil) caused by incompatible external flushes (too high a vapor pressure) or failure of the standby pump seal during start-up.

This best practice solves both issues by eliminating the need for an external flush as well as ensuring the seal fluid in the standby pump chamber will be at an acceptable viscosity under start-up conditions.

Benchmarks

This best practice has been used since the mid-1970s to optimize refinery bottoms and gas oil pump mechanical seal MTBFs (greater than 36 months).

B.P. 8.8. Supporting Material

The part of the pump that is exposed to the atmosphere and that the rotating shaft or reciprocating rod passes through is called the stuffing box. A properly sealed stuffing box prevents the escape of pumped liquid. Mechanical seals are commonly specified for centrifugal pump applications (refer to Figure 8.8.1).

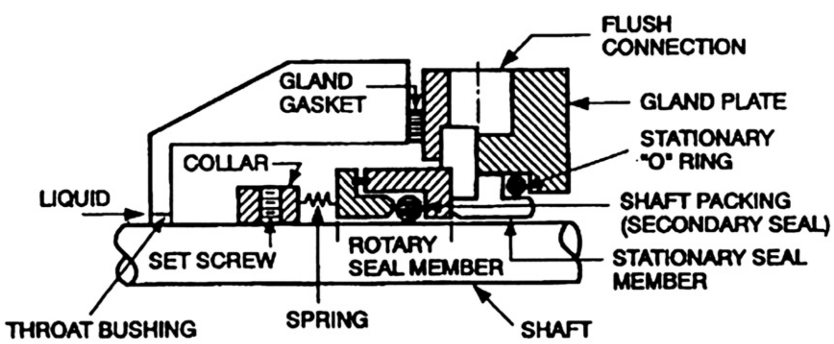


Fig 8.8.1 • Typical single mechanical seal

Function of mechanical seals

The mechanical seal is comprised of two basic components (refer to Figure 8.8.2).

- Stationary member fastened to the casing
- Rotating member fastened to shaft, either direct or with shaft sleeve

Fig 8.8.2 • Basic seal components

The mating faces of each member perform the sealing. The mating surface of each component is highly polished, and they are held in contact with a spring or bellows which results in a net face loading closure force (refer to Figure 8.8.1).

In order to prevent fluid escaping to the atmosphere, additional seals are required. These seals are either 'O' rings, gaskets or packing (refer to Figure 8.8.1). For high temperature applications (above 200°C [400°F]) the secondary seal is usually 'Graphoil' or 'Kalrez' material in a 'U' or chevron configuration. An attractive alternative is to eliminate the secondary seal entirely by using a bellows seal, since the bellows replaces the springs and forms a leak-tight element thus eliminating the requirement for a secondary seal (refer to Figure 8.8.3).

To achieve satisfactory seal performance over extended periods of time, proper lubrication and cooling is required. The lubricant, usually the pumped product, is injected into the seal chamber and a small amount passes through the interface of the mating surfaces. Therefore, it can be stated that all seals leak, and the amount of leakage depends on the pressure drop across the faces. This performance can be considered to be flow through an equivalent orifice (refer to Figure 8.8.4).

The amount of heat generated at the seal face is a function of the face loading and the friction coefficient, which is related to the materials and lubrication of the faces. Figure 8.8.5 shows the equation for calculating the amount of heat that needs to be removed by the flush liquid.

As the lubricant flows across the interface, it is prone to vaporization. The initiation point of this vaporization is dependent upon the flush liquid pressure, and its relationship to the margin of liquid vapor pressure at the liquid temperature. The closer the liquid flush pressure is to the vapor pressure of the liquid at the temperature of the liquid, the sooner vaporization will occur (refer to Figure 8.8.6).

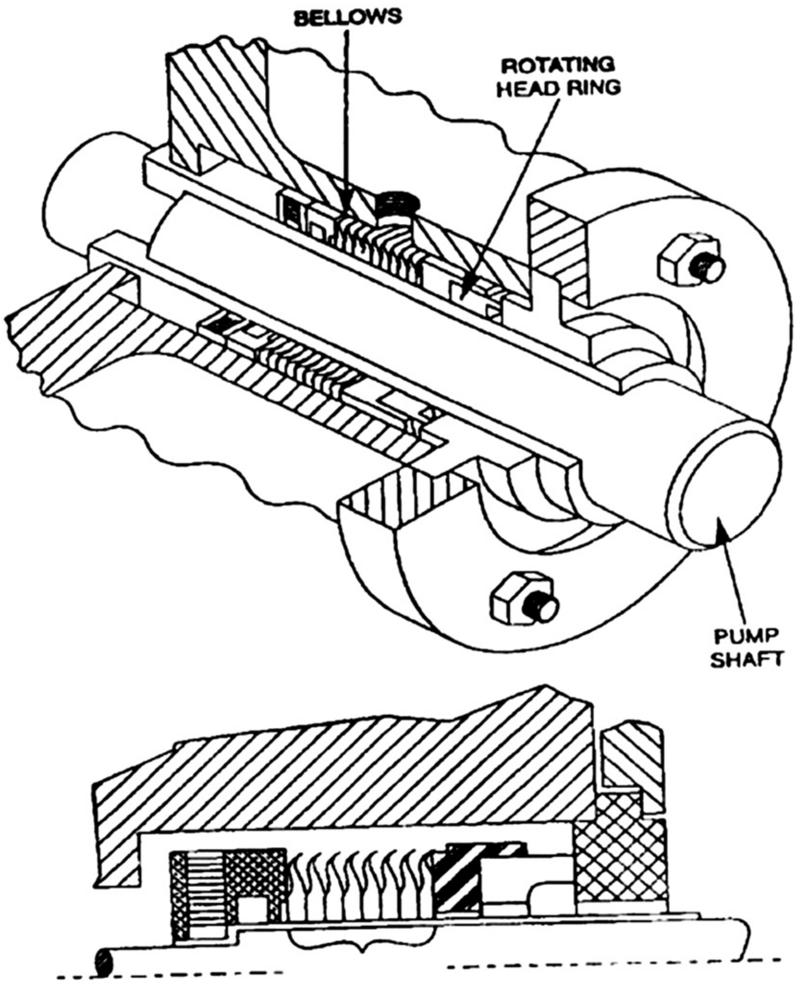


Fig 8.8.3 • Metal bellows seal

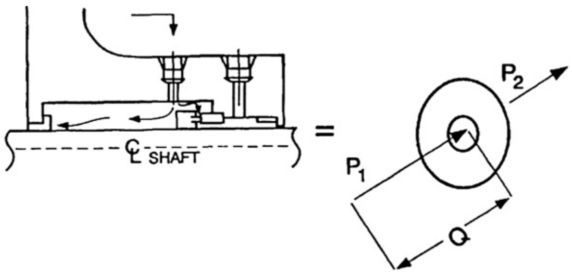
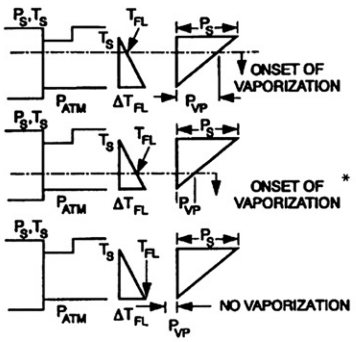


Fig 8.8.4 • Equivalent orifice flow across seal faces

$Q = 500 \bullet S.G. \bullet Q_{INJ} \bullet Cp \bullet \Delta T$
where: Q = heat load (BTU/HR)
S.G. = specific gravity of injection liquid
Cp = specific heat of injection liquid $\left(\frac{BTU}{lb-°F}\right)$
ΔT = temperature rise of injection liquid (°F)
Q_{INJ} = injection liquid flow rate (G.P.M.). If flow is in LPM, constant = 125.

Fig 8.8.5 • Heat generated by a mechanical seal



*This represents the design case, vaporization approximately 3/4 down the faces

Fig 8.8.6 • Typical seal face pressure temperature relationship to vaporization

The seal system

To ensure reliable, trouble-free operation for extended periods of time, the seal must operate in a properly controlled environment. This requires that the seal be installed correctly, so that the seal faces maintain perfect contact and alignment, and that proper lubrication and cooling be provided. A typical seal system for a simple, single, mechanical seal is comprised of the seal, stuffing box throat bushing, liquid flush system, auxiliary seal and auxiliary flush or barrier fluid (when required) (refer to Figure 8.8.7).

The purpose of the seal is to prevent leakage of pumped product from escaping to the atmosphere. The liquid flush (normally pumped product from the discharge) is injected into the seal chamber to provide lubrication and cooling. An auxiliary seal is sometimes fitted to the gland plate on the atmospheric side of the seal chamber. Its purpose is to create a secondary containment chamber, when handling flammable or toxic fluids that would be considered a safety hazard to personnel if they were to leak to atmosphere. A liquid (non-toxic) flush or barrier fluid, complete with a liquid reservoir and appropriate alarm devices can be used to ensure toxic fluid does not escape to the atmosphere.

Controlling flush flow to the seal

The simple seal system shown in Figure 8.8.8 incorporates an orifice in the flush line from the pump discharge to the mechanical seal. Its purpose is to limit the injection flow rate to the seal and to control pressure in the seal chamber. A minimum bore diameter or 3 mm (1/8") is normally specified (to minimize potential of blockage) and the orifice can either be installed between flanges or in an orifice nipple.

Examining some causes of seal failures

An indication of some causes of seal failures can be obtained while the seal is operating. When you consider the seal as an equivalent orifice, an examination of ‘tell-tale’ symptoms can causes of indicate potential failure for which corrective action can be implemented or at least can provide direction of

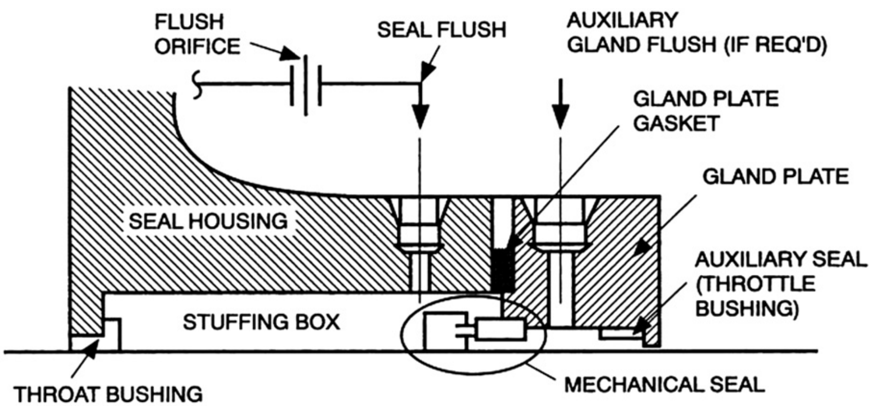


Fig 8.8.7 • Simple seal system

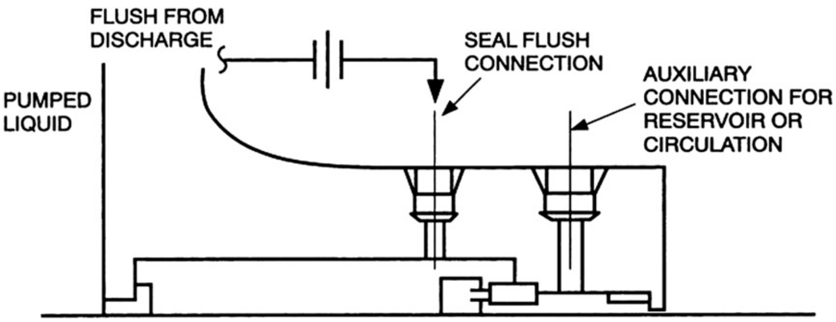


Fig 8.8.8 • Seal flow control orifice

subsequent failure analysis (refer to Figure 8.8.9). It should be noted that improper application, installation, and/or manufacturing errors can also result in mechanical seal failures.

Comments	Possible causes	Comments/ recommendations
■ Seal squeal during operation	Insufficient amount of liquid to lubricate seal faces	Flush line may need to be enlarged and/or orifice size may need to be increased
■ Carbon dust accumulating on outside of seal area	Insufficient amount of liquid to lubricate seal faces	See above
	Liquid film vaporizing/ flashing between seal faces	Pressure in seal chamber may be too low for seal type
■ Seal spits and sputters in operation (popping)	Product vaporizing/ flashing across seal faces	Corrective action is to provide proper liquid environment of the product at all times 1. Increase seal chamber pressure if it can be achieved within operating parameters (maintain at a minimum of 175 kPa (25 psig) above suction pressure) 2. Check for proper seal balance with manufacturer 3. Change seal design to one not requiring as much product temperature margin (ΔT) 4. Seal flush line and/or orifice may have to be enlarged 5. Increase cooling of seal faces Note: A review of seal balance requires accurate measurement of seal chamber pressure, temperature and product sample for vapor pressure determination

Fig 8.8.9 • Possible causes of seal failure

Seal configurations

Mechanical seals are the predominant type of seals currently in use in centrifugal pumps. They are available in a variety of configurations, depending upon the application service conditions and/or the user's preference (refer to Figures 8.8.10 to 8.8.13 for the most common arrangements used in refinery and petrochemical applications).

Single mechanical seal applications

Single mechanical seals (refer to Figure 8.8.10) are the most widely used seal configuration, and should be used in any application where the liquid is non-toxic and non-flammable.

As mentioned earlier in this section, many single mechanical seals are used with flammable and even toxic liquids, and rely solely on the auxiliary seal throttle bushing to prevent leakage to atmosphere. Since the throttle bushing does not positively contain leakage, state and federal environmental regulations now require the use of a tandem or double seal for these applications. In some plants, a dynamic type throttle bushing ('Impro' or equal) is used to virtually eliminate leakage of the pumped fluid to atmosphere in the event of a mechanical seal failure.

Tandem mechanical seal applications

Tandem mechanical seals (refer to Figure 8.8.11) are used in applications where the pumped fluid is toxic and/or flammable.

They consist of two (2) mechanical seals (primary and back-up). The primary seal is flushed by any selected seal flush plan. The back-up seal is provided with a flush system incorporating a safe, low flash point liquid. There is a pressure alarm which actuates on an increase in stuffing box pressure between the primary and back-up seal thus indicating a primary seal failure. Since the pumped product now occupies the volume between the seals, failure of the back-up seal will result in leakage of the pumped fluid to atmosphere. In essence any time a tandem seal in alarm, it is actually a single seal and should be shut down immediately to ensure that the toxic and/or flammable liquid does not leak to atmosphere.

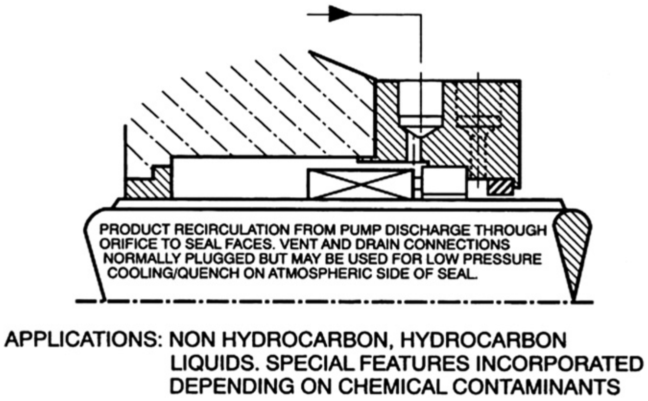


Fig 8.8.10 • Single mechanical seal

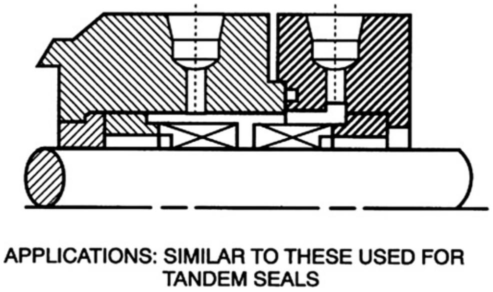


Fig 8.8.12 • Double mechanical seal

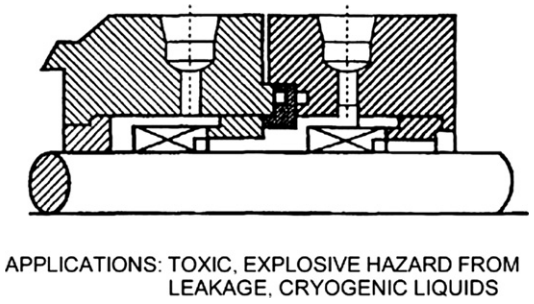


Fig 8.8.11 • Tandem seal arrangement

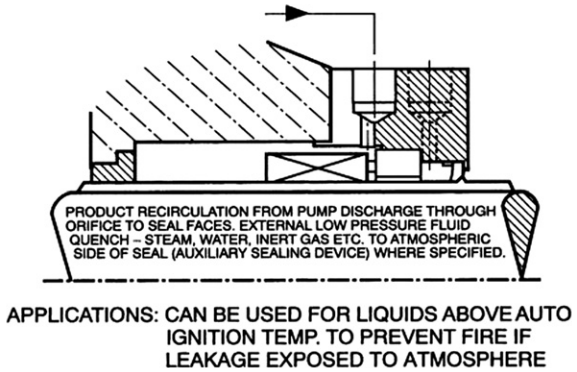


Fig 8.8.13 • Liquid/gas tandem seal combination

Double mechanical seal applications

Double mechanical seals (refer to Figure 8.8.12) are used in applications where the pumped fluid is flammable or toxic, and leakage to atmosphere cannot be tolerated under any circumstances. Typical process applications for double seals are H_2S service, hydrofluoric acid alkylation services or sulfuric acid services.

Leakage of the pumped fluid to the atmosphere is positively prevented by providing a seal system, whose liquid is compatible with the pumped liquid, which continuously provides a safe barrier liquid at a pressure higher than the pumped fluid. The seals are usually identical in design with the exception that one seal incorporates a pumping ring to provide a continuous flow of liquid to cool the seals. Typical double seal system components are: reservoir, cooler, pressure switch and control valve.

Liquid/gas tandem mechanical seal applications

In this configuration (refer to Figure 8.8.13) a conventional single liquid mechanical seal is used as the primary and a gas seal (non-contacting faces) that can temporarily act as a liquid seal in the event of primary seal failure serves as the back-up seal. This seal configuration is used in low specific gravity applications where the pumped fluid is easily vaporized. Using a gas seal as the back-up has the advantage of eliminating the vessel, cooler and pumping ring necessary for conventional tandem liquid seals.

This application is well proven, and has been used successfully for natural gas liquids, propane, ethylene, ethane and butane pump applications.

Double gas seal applications

Before leaving this subject, a relatively new application utilizes two (2) gas seals in a double seal configuration, and uses N_2 or air as a buffer, maintained at a higher pressure than the pumped fluid to positively prevent the leakage of pumped fluid to atmosphere. This configuration, like the tandem liquid/gas seal mentioned above, eliminates the seal system required in a conventional liquid double seal arrangement. Note however, that the pumped product must be compatible with the small amount of gas introduced into the pumped fluid. This configuration cannot be used in recycle (closed loop) services.

An excellent resource for additional information covering design, selection and testing criteria is API Standard 682.

Best Practice 8.9

Avoid the use of flush line strainers and cyclone separators in dirty services, and use an external flush or a dual pressurized seal to optimize mechanical seal MTBFs:

Mechanical seal reliability is significantly affected by the operational characteristics of the flush system and its components.

In services where the pumped fluid can contain solids, API 610 and API 682 both offer the option of using flush line strainers or cyclone separators.

Flush line strainers can become blocked resulting in immediate seal failure.

Cyclone separator effectiveness is dependent on the relative density difference between the fluid and solid particles and the flush line piping (adequate slope for the debris drain line back to the pump suction).

Using an external clean flush, if available, or a dual pressurized seal arrangement will eliminate the need for flush line strainers and cyclone separators and ensure optimum mechanical seal MTBFs.

Lessons Learned

Flush line strainers and cyclone separators have been the cause of low seal MTBFs (less than 12 months) in many applications where the seal fluid contains solid particles. Eventual modification to a clean external flush or a dual pressurized seal has significantly increased seal MTBFs (greater than 48 months).

Benchmarks

This best practice has been used since the 1990s to thoroughly investigate, with the process engineers, the possibility of using a clean external flush source in services where solid particles were contained in the seal fluid. A contingency recommendation where a clean external flush was not available was to use a dual pressurized seal.

In many cases, the additional cost of an external flush or dual pressurized seal was justified on the basis of past plant mechanical seal maintenance history and the loss of revenue when the standby pump was under maintenance and the operating pump failed.

B.P. 8.9. Supporting Material

See B.P: 8.2 for supporting material.

Best Practice 8.10

Properly design and monitor quench systems for optimum seal reliability and to ensure that moisture does not enter the bearing housing.

The proper function of seal quench systems is to remove solid particles from the lower seal face to prevent premature seal wear.

The term ‘quench’ originally came from the use of steam to buffer the outer seal chamber, between the seal and throttle bushing, to dilute the hydrocarbon fluid leaking from the seal to a safe non-flammable level. Today (2010) this practice is not acceptable and dual seals are required to be used for all hydrocarbon services.

When a quench is used today (2010) either steam (where solid hydrocarbon particles can form) or water (where water soluble particles can form) are used. It is solely for the purpose of removing solid particles from the lower seal face. Both of these alternatives expose the bearing housing to entrance of water vapor which will impact bearing reliability and MTBFs.

Proper system design to control the amount and condition of the quench fluid and the mandated use of a bearing bracket oil condition monitoring bottle is essential to the reliability of mechanical seals employing quench systems.

Lessons Learned

Failure to regulate quench fluid conditions and to monitor the operation of seal quench systems has resulted in low mechanical seal MTBFs (lower than 12 months). In one case, it resulted in a refinery fire when the steam quench became saturated and displaced all of the oil in a pump bearing bracket. An excessive hydrocarbon seal leak was ignited by the ‘red hot’ bearing bracket.

Benchmarks

This best practice has been used since the mid-1980s when I was faced with contamination of pump bearing brackets with saturated steam from malfunctioning steam quench systems in refinery service. Since that time, this best practice has optimized mechanical seal quench system safety and reliability.

B.P. 8.10. Supporting Material

Considerations for quench

A quench (known as an auxiliary flush plan) is a flush plan that uses a medium (steam, nitrogen, or water) on the atmospheric side of the seal to wash away any solids buildup from the faces. The buildup is drained to a collection system. Refer to Figure 8.10.1 for a schematic.

A quench is most commonly used in single seals with steam as the medium, if the seal fluid is hot and can form coke particles, and/or if the seal fluid is flammable or toxic. Note: many

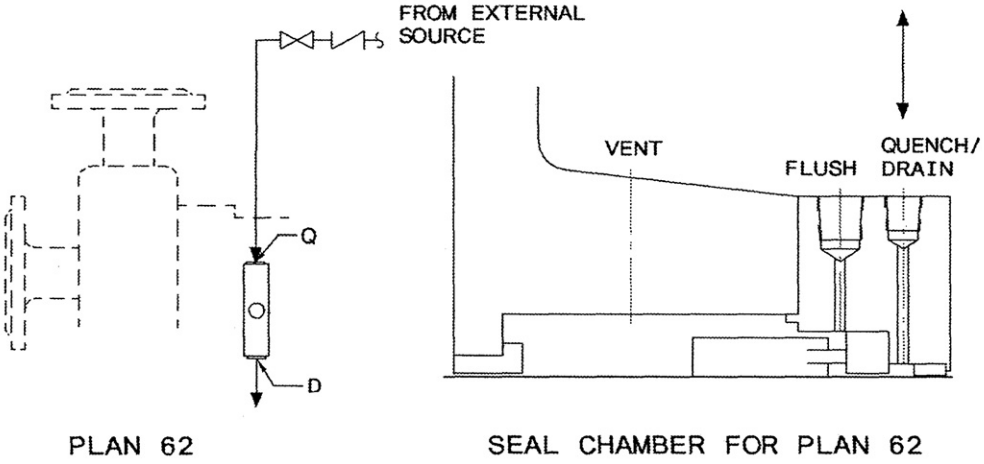


Fig 8.10.1 • Plan 62 – a medium (usually steam or water) is introduced on the atmospheric side of the seal to prevent crystallization or buildup of solids

countries today require dual (tandem or double) seals to meet environmental requirements if the seal fluid is flammable or toxic. The steam should be regulated to a pressure of approx. 20 to 33 kPa (3 to 5 psi), which is just enough to wash the solids accumulation off the atmospheric side of the faces. It is essential for the steam to be superheated (dry), to prevent flashing of water at the faces, causing premature failures and to ensure that moisture does not enter the bearing chamber. We have experienced plant fires as a result of water contamination in the bearing housing resulting in a hot bearing which served as an ignition source for a single seal leaking a flammable vapor. We recommend that an 'oil condition bottle', to monitor water in the oil, always be installed in the bearing housings when a steam or water quench is used.

Best Practice 8.11

Execute the following mechanical seal monitoring best practices for optimum mechanical seal MTBFs:

- Confirm that the pump is operating in its EROE (see B.P 2.7 for EROE details)
- Check for a plugged flush line orifice by taking temperature reading across it
- Confirm seal chamber pressure is at least 345 kPa (50 psi) above the fluid's vapor pressure
- If the flush system contains a cyclone separator or strainer, check for plugged components by taking temperature readings
- If the flush fluid is cooled, confirm the proper function of the cooler by checking the temperature drop across the cooler (should be approx. 10 to 38°C [50 to 100°F] normally) and the temperature rise of the cooling medium
- Check the temperature difference on seal reservoirs (pots) between buffer/barrier in and out lines to ensure circulation to and from the outboard seal. If there is no temperature difference, the circulation through the buffer/barrier circuit has halted
- Always vent seal pot systems at the highest point to ensure pumping ring circulation
- For dual pressurized seal applications install a permanent differential pressure gauge to ensure that the seal pot filling pressure is not excessive, which will force open the inner seal

We recommend that the above predictive maintenance (PDM) guidelines be followed for all centrifugal pumps in your facility at the following times:

- At pump commissioning
- After the pump operating conditions are normalized
- Whenever the plant condition monitoring results indicate a change (in vibration, temperature, seal leakage, noise, etc.)
- Immediately after seal replacement

Lessons Learned

Failure to monitor centrifugal pump performance and the effect of the process conditions on pump flow rate are the major contributors to centrifugal pump mechanical component failure (seals, bearings, wear rings and impeller).

Most plant condition monitoring programs do not integrate centrifugal pump performance (operating point and produced head) with mechanical condition (vibration and temperature). Neglecting pump performance, in FAI experience, neglects consideration of approximately 80% of the potential root causes for mechanical seal failure.

Benchmarks

This best practice has been used since the mid-1990s to recommend plant PDM practices for pumps with mechanical seals to optimize plant centrifugal pump safety and mechanical seal reliability (MTBFs above 48 and as high as 80 months).

B.P. 8.11. Supporting Material

Seal vendors design the seal balance ratio and select face materials based on a parameter known as the PV to be able to change the fluid to a vapor approximately $^3/_4$ of the way down the seal faces (see Figure 8.11.1).

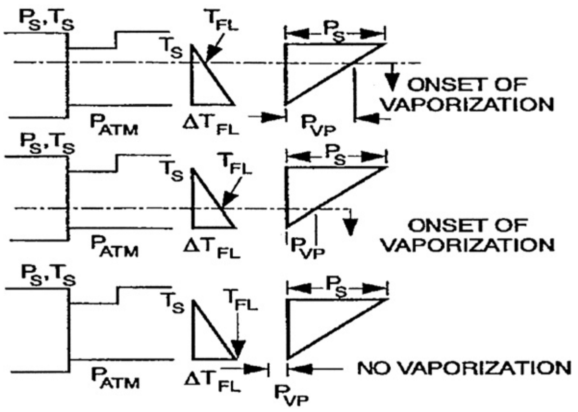


Fig 8.11.1 • Mechanical seal primary face vaporization point

As can be seen in Figure 8.11.2, head required is a function of the pressure differential across the pump flanges and inversely proportional to the specific gravity of the pumped fluid. Therefore, process changes of P2, P1 and/or S.G. will change the pump flow in any centrifugal pump if a process control system is not present or operational (failed or in manual mode). These facts are presented in Figure 8.11.3.

In Figure 8.11.1, three distinct operating modes are shown for the primary ring (on the left) and the mating ring (on the right).

- The top mode shows a condition of early vaporization which can occur on light fluids, or hot fluids or where seal chamber pressure is lower than designed (note: seal chamber pressure is designed to be at least 345 kpa (50 psi) or above the seal fluid vapor pressure).
- The middle figure shows the desired design condition of changing the fluid to a vapor approximately $^3/_4$ of the way down the faces. This is the design basis since it is known that fluid characteristics, temperature and/or pressure will change during operation a certain amount. Excessive changes in any or all of these parameters will lead to seal failure.
- The last mode shows the case of no vaporization and apparent seal failure. In reality, this is not a failure but only a seal fluid condition change that does not allow the seal fluid to reach the fluid vapor pressure between the seal faces before it exits the seal.

As can be seen from Figure 8.11.1, the ability to achieve the objective of vaporization $^3/_4$ or 75% down the seal faces (see center case in Figure 8.11.1) depends on the following:

- Seal fluid characteristics – cleanliness, specific heat, vapor pressure and viscosity
- Pressure of the seal fluid in the seal chamber
- Temperature of the seal fluid entering the seal chamber

What determines the condition of items listed above? The process. Therefore, effective mechanical seal condition monitoring requires that all seal process conditions, as noted above are considered.

Decreased Pump Flow:
- Increased P2
- Decreased P1
- Decreased S.G.

Increased Pump Flow:
- Decreased P2
- Increased P1
- Increased S.G.

Fig 8.11.3 • Process effects on centrifugal pump flow

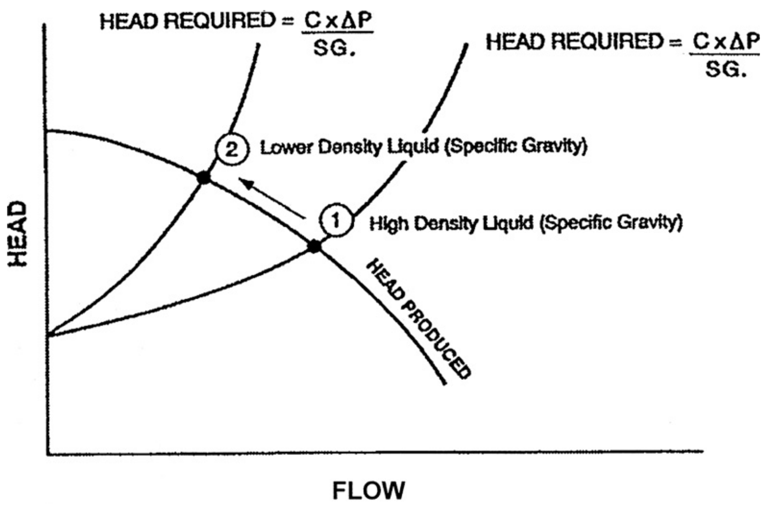


Fig 8.11.2 • Head required by lower density fluid

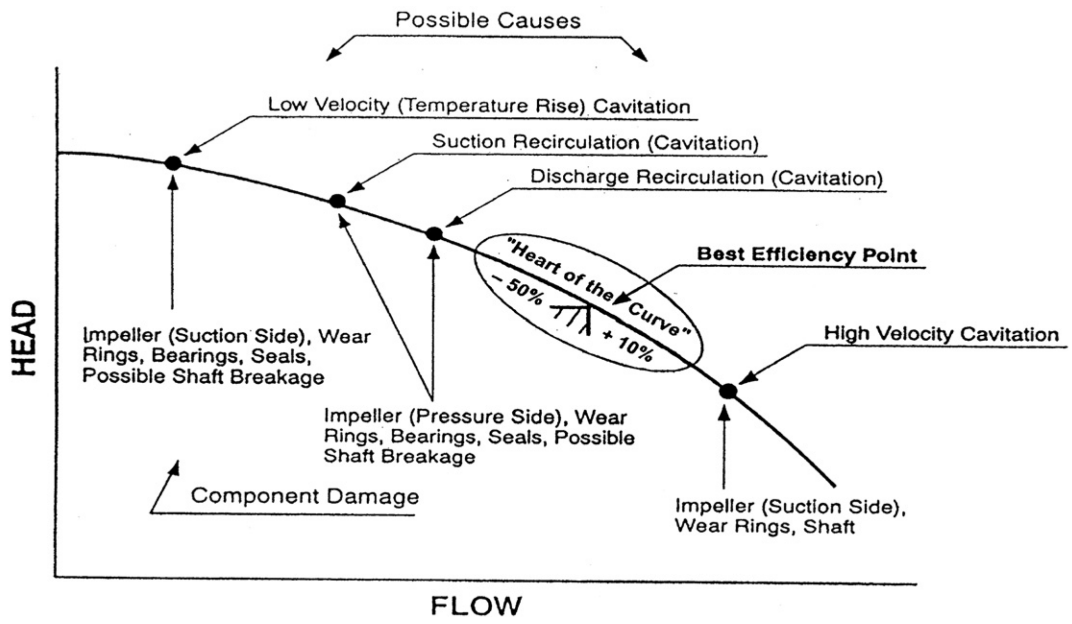


Fig 8.11.4 • Typical centrifugal pump head vs. flow curve

EROE (equipment reliability operating envelope) determination

Figure 8.11.2 shows the effect of the head required by the process on centrifugal pump flow rate. As noted in Figure 8.11.3, process changes will vary the flow of any centrifugal pump.

If the centrifugal pump flow is too high or too low, hydraulic disturbances will be present that can change the pumped fluid pressure and/or temperature. Since the majority of mechanical seal applications use the pumped fluid in the seal chamber, the seal chamber pressure and/or temperature will be affected. As can be seen in Figure 8.11.1, these changes will directly impact mechanical seal life and reliability.

Figure 8.11.4 shows a typical centrifugal pump head vs. flow curve with the following items noted:

- The 'desirable region' of operation – heart of the curve or EROE
- Regions of hydraulic disturbances – on the upper portion of the curve
- The pump components affected – on the lower portion of the curve

The 'heart of the curve' is the flow region for any centrifugal pump that will be free of hydraulic disturbances and where the seal fluid should be free of vapor if the seal fluid conditions stated on the pump and seal data sheets are present during pump field operation.

This flow region is also called the EROE – the equipment reliability operating envelope

Figure 8.11.5 presents facts concerning the EROE.

In many pump installations, neither a flow meter nor a suction pressure gauge is installed. A calibrated suction pressure gauge can be installed in the suction pipe drain connection (always present). Be sure to obtain a MOC (management of change) and work permit, and any other plant required permission prior to installing a suction pressure gauge as the pumped fluid could be sour (H_2S), flammable and/or carcinogenic.

If a flow meter is not installed, Figure 8.11.6 defines the options available to determine the pump flow so the EROE can be obtained.

The flow values in Figure 8.11.6 can be determined by hand calculations using the equations available in any pump text (power equation and pump temperature rise equation). It can be seen that the EROE will provide a reasonable guide that usually will eliminate the hydraulic disturbances that could cause seal

1. The EROE flow range is + 10% and – 50% of the pump best efficiency point (BEP) flow
2. All 'bad actor' pumps (more than one component failure per year) should be checked for EROE
3. To determine that the pump is operating in EROE:
   - Calculate the pump head required
   - Measure the flow
   - Plot the intersection of head and flow on the pump shop test curve

Fig 8.11.5 • EROE facts

1. Measure motor amps and calculate power
2. Record control valve position, valve differential pressure, fluid S.G. and calculate valve flow (pump flow)
3. Measure pump pipe differential temperature and calculate pump efficiency
4. Obtain an ultrasonic flowmeter to measure flow
5. For items 1 and 3, locate the calculated value (power or efficiency) on the pump test curve to determine pump flow

Fig 8.11.6 • Available pump flow determination options

chamber pressures and temperatures to change and lead to premature seal wear and/or failure. Note that the stated EROE low flow range can be reduced if the pump or fluid have any of the characteristics noted in Figure 8.11.7.

- Pumps with suction specific speeds > 8,000 (customary units)
- Double suction pumps
- Water pumps with low NPSH margin
- Fluids with S.G.< 0.7

Fig 8.11.7 • Factors that can reduce low flow EROE range

Therefore, we always recommend that the first step in seal condition monitoring is determination of pump operation within its EROE. If the 'bad actor' pump is operating outside its EROE, we recommend the action shown in Figure 8.11.8.

- Consult operations to determine if process changes can be made to operate in EROE
- Define target EROE parameters for operations (flow, amps, control valve position, delta T)

Fig 8.11.8 • If a centrifugal pump is outside its EROE:

If seal reliability does not improve when operating within the EROE, further investigation is required concerning the process conditions in the seal chamber and/or flush system as noted in the remaining sections.

Confirm process conditions are as stated in data sheets

If the 'bad actor' pump seal reliability, when operating in the EROE, does not significantly improve, a complete check of seal fluid conditions in the seal chamber is required. Figure 8.11.9 shows the process variables that influence seal reliability.

1. Seal Chamber Temperature
2. Seal Chamber Pressure
3. Seal Fluid Characteristics:
   - Cleanliness
   - Vapor Pressure
   - Viscosity
   - Specific Heat
   - Specific Gravity

Fig 8.11.9 • Seal reliability and pump fluid conditions are based on:

Temperature monitoring

Since seal chamber or seal flush line pressure gauges are not usually installed on new pumps, the first place to start is with seal chamber temperature. Flush plans with coolers (21, 23, 41 etc.) have an option for a thermometer downstream of the cooler which can be used to determine the seal chamber temperature. This value must be compared to the PT (pumping temperature) value listed on the data sheet. If the measured value does not agree with the data sheet, consult with operations first to see if process changes can be made. If they cannot be made, discuss the measured temperature with the seal vendor representative. Alternative options for measuring seal fluid temperature are noted in Figure 8.11.10.

- Temperature gun close to seal gland – be sure to monitor off of a non-reflective surface
- Surface contact thermometer on flush line – add 5°C to measured value if pipe and use recorded value if SS tubing
- Install a thermometer in the flush line
- Use an infrared camera to record temperature and temperature profile

Fig 8.11.10 • Alternative methods to measure seal fluid temperature

Pressure monitoring

Unless the plant assigned machinery specialist is 'world class' there are usually no pressure gauges in the flush line or seal chamber. Our recommendations concerning seal chamber pressure monitoring are presented in Figure 8.11.11. The measured seal chamber pressure must be compared to the seal vendor's assumed value, which should be on the seal layout drawing. If this value is not present, the seal vendor should be consulted. Note that the seal vendor assumed seal chamber pressure is the value that was used in the calculation of the seal PV. If the measured value does not agree with the data sheet, consult with operations first to see if process changes can be made. If process changes cannot be made, discuss this fact with the seal vendor representative.

- Install a pressure gauge in the seal chamber
- Modify the seal flush line for installation of a pressure gauge

Fig 8.11.11 • Seal chamber pressure monitoring guidelines for 'bad actor' seals

Seal fluid characteristics

If all of the above mentioned items are in accordance with the seal design (EROE, seal chamber temperature and seal chamber pressure), a check of the seal fluid characteristics is required. Figure 8.11.12 presents guidelines for checking the seal fluid characteristics.

If the measured fluid sample parameters are not as stated on the data sheet, consult with operations first to see if process changes can be made. If process changes cannot be made, discuss this fact with the seal vendor representative.

- Take a sample to determine if debris is present
- If seal fluid is not water or product fluid (which must meet product specifications) have sample analyzed
- Send sample results to seal vendor

Fig 8.11.12 • Seal fluid check guidelines

- Orifice condition
- Seal 4
- Throat bushing clearance
- Cooler condition
- Strainer condition
- Cyclone separator condition
- Buffer fluid condition (tandem seal)
- Buffer fluid level (tandem seal)
- Pumping ring condition
- Buffer fluid pressure (tandem seal)

Fig 8.11.13 • Seal reliability as a function of flush system component condition is based upon:

Variance in seal chamber pressures and temperatures, if the pump is operating in the EROE, are most likely to be caused by malfunction of components in the flush system. Figure 8.11.13 defines the flush system components which can affect seal chamber pressures and temperatures.

Based on the seal flush plan used, the flush system components should be checked in the logical order – starting with the beginning of the flush system and ending with the throat bushing. (The exception is flush Plan 13 which begins with the throat bushing and ends at the pump suction pipe.) FAI 'seal maintenance best practice' requires that the flush system be completely checked (including the throat bushing clearance) each time a seal is changed. Note that this recommendation applies to all seal configurations: single, tandem (dual un-pressurized) or double (dual pressurized).

Coordination with the seal vendor representative

After all mechanical seal and flush system condition parameters have been measured, coordination with the seal vendor representative and/or in-house seal vendor engineer is necessary. A 'team' approach is very effective and is being used by all seal vendors today in large gas plants, refineries and chemical plants. This approach provides immediate seal condition monitoring and troubleshooting assistance as well as on site manufacturing capability. Suggested seal vendor field information is presented in Figure 8.11.14.

The following field seal information is suggested to be exchanged with the seal vendor for seal reliability issues:

- If the pump is operating in the EROE (provide curve)
- Seal chamber temperature
- Seal chamber pressure
- Seal fluid sample results
- Seal flush system component operation confirmation
- Throat bushing clearance confirmation
- Seal jacket temperature change (if applicable)
- Seal quench fluid pressures, temperatures and condition (if applicable)

Fig 8.11.14 • Seal vendor field information

Best Practice 8.12

All flush Plan 52s (un-pressurized tandem seals) must be continuously vented to flare or safe point to ensure that the seal pot is not contaminated with the inner seal process fluid which will expose the plant to a process fluid release through the outer seal. A laminated warning sign is recommended at each application of this kind.

Un-pressurized dual seals (formerly called 'tandem') require an open path to flare or high point safe location to remove all seal fluid vapors (usually flammable and/or toxic) from the seal pot and direct them to a safe location.

A valve closure and/or restriction in the seal pot vent line will prevent removal of the process fluid, contaminate the seal pot safe barrier fluid and expose the plant and its personnel to a flammable or toxic fluid release in the event of an outer (atmospheric side) seal failure.

It is recommended that a local, laminated, warning sign be positioned at the pump.

In addition, a short 'mini information' presentation by site rotating machinery specialists has been effective in making operators aware of this potential safety hazard.

Lessons Learned

Most plants have experienced a release of process fluid from dual un-pressurized (tandem) seal systems when the vent line had been improperly closed off.

Lack of awareness of dual un-pressurized (tandem) seal system operational principles has resulted in flammable and toxic fluid releases in many plants.

Benchmarks

This best practice has recommended since the mid-1980s. Since that time, this best practice, when implemented by plant operations, has resulted in optimum dual un-pressurized (tandem) seal safety and reliability (MTBFs > 48 and as high as 80 months).

B.P. 8.12. Supporting Material

API Flush Plan 52

Dual un-pressurized seals (tandem) rely on a buffer fluid at or near atmospheric pressure to lubricate the atmospheric side seal. This buffer fluid is circulated via a pumping ring from the seal to the seal reservoir and back to the seal (in a closed loop). Take a look at Figure 8.12.1 for a schematic.

Plan 52 – dual unpressurized seal using synthetic buffer fluid to lubricate the atmospheric side seal. A pumping ring in the seal circulates the buffer fluid (pressure less than seal chamber) to the reservoir.

The reservoir is at atmospheric pressure (less than seal chamber pressure), so the leakage across the process side seal faces migrates into the seal reservoir, and will either increase pressure, level, or both in the reservoir. Since every seal does leak a certain amount, it is essential to have the reservoir vented to a flare or vapor collection system. If the reservoir is allowed to reach the seal chamber pressure, the atmospheric side seal will most likely fail (if it hasn't already) as it is not typically designed to handle seal chamber pressure. If this is a concern in the plant, you may want to consider requesting the seal vendor to redesign the atmospheric seal to handle maximum seal chamber pressure. In addition, as the process side seal leaks in this flush plan, the atmospheric side seal will essentially be sealing the pumped fluid, exposing the plant to the release of flammable and/or toxic vapors.

Monitoring of seal leaks can be done by checking the level and pressure of the reservoir, as one or both may increase in the event of excessive leakage. The seal vendor (or support system vendor) may supply high level and/or pressure switches which would alert the operators to a seal leak. It is highly recommended to specify this instrumentation in new projects, as it will cut down on the already high workload of operators (if the alarm sounds then the level or pressure can be personally verified in the field).

In addition to checking for excessive leakage (pressure or level increase), temperature in and out of the seal can help verify proper circulation. In a reservoir that is not cooled, there should be a temperature drop of approx. 1 to 2°C (5°F to 10°F) from the seal outlet to the seal inlet. If there is no temperature drop (or if the temperature increases), this indicates zero or possibly reverse circulation.

It has been our experience that Plan 52 flushes may not be vented properly (blocked in) and level may be low or at zero in the reservoir. For these reasons, it is very important for the operators to be trained to understand the necessity of monitoring this system.

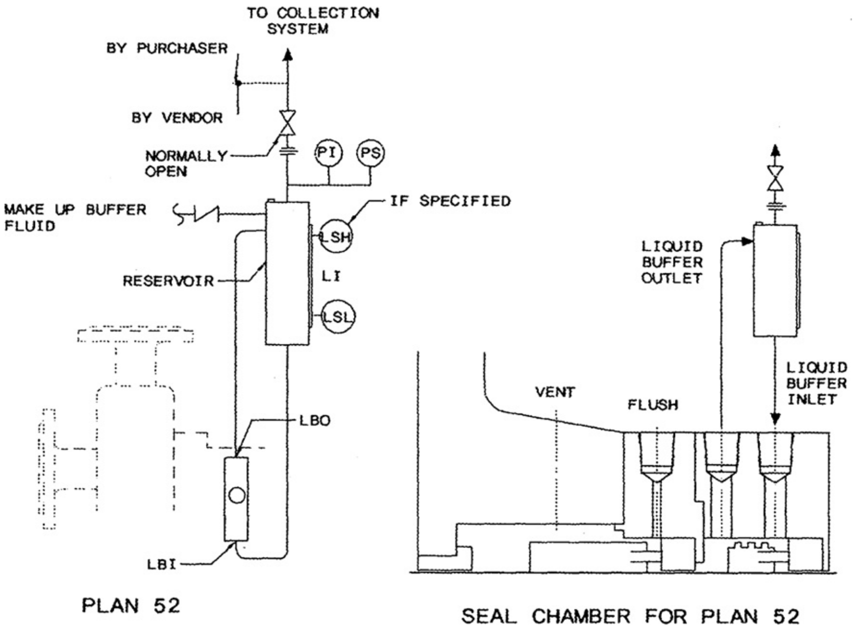


Fig 8.12.1 • Plan 52

Best Practice 8.13

Always execute the following seal installation best practice guidelines for component and cartridge mechanical seals to achieve maximum pump mechanical seal MTBFs:

- Component seal faces must be cleaned with clean cloth and rubbing alcohol (or alcohol wipe) just before installation. Proper care must be taken to not touch the faces after cleaning.
- Limited grease should be used on seal gland 'O' ring as it can migrate to the seal faces and cause damage. Proper practice is to sparingly apply grease to 'O' ring groove then install it in the gland. This will ensure 'O' ring will stay in the groove when installing the seal.
- Use liquid soap sparingly on sleeve 'O' ring to assist in installation.

It is recommended that this best practice be incorporated into the plant maintenance procedure manual for mechanical seal installation.

Lessons Learned

Failure to have and implement a required mechanical seal installation procedure has been a principle cause of low mechanical seal MTBFs (lower than 12 months).

Benchmarks

FAI has used this best practice since the 1990s to ensure that plant mechanical seal installation procedures are in complete accordance with mechanical seal vendors' recommendations and to ensure optimum installed seal safety and reliability.

B.P. 8.13. Supporting Material

Component and cartridge mechanical seal installation guidelines

Component mechanical seal installation

- Refer to the vendor instruction book and obtain all required mechanical seal components.
- Have a clean area available for the components and ensure that there are no finger prints, oil or grease on the primary seal faces. Wipe with alcohol using a lint free cloth if necessary.
- Primary seal face flatness will have been confirmed by the seal manufacturer and should be within 1–2 helium light bands. For this reason, the primary seal faces must be free of fingerprints, oil and/or grease.
- Inspect the rotating head assembly as follows:
- Secondary 'O' ring in pusher seal – inspect for discontinuities, confirm proper material and durometer. Note: do not lubricate 'O' ring.
- Ensure that all springs (if a pusher seal) are sitting upright and have not been dislodged from their counter-bores.
- Confirm that shaft or shaft sleeve surface finish, where secondary 'O' ring or Teflon wedge will ride, is maximum of 32 rms, using a comparator.
- Clean and de-burr the shaft area and seal chamber area as required using a solvent that will be compatible with the process fluid. Note: assure all piping/tubing is isolated from flush/quench ports. Plastic plugs need to be installed in all flush/quench after isolation of piping/tubing up to the point of the piping/tubing reinstallation.
- Install the seal head carefully being sure not to damage the secondary 'O' ring, or the ID of the primary ring on the shaft/sleeve.
- Proper spring compression check ('working height') – obtain dimension from seal drawing which shows the distance between the back of the seal retainer and the seal chamber face. Then fit the bearing bracket and shaft up against the casing and mark (with bluing pen) the point on shaft or sleeve directly below the seal chamber face. Remove shaft and bearing bracket from casing and mark the dimension where the back of the retainer sits when assembled.
- Attach seal head to shaft/sleeve and ensure that set screws are installed on clean and undamaged shaft areas. If this is not possible, carefully de-burr with a file and smooth out with emery cloth.
- Note: If sleeve 'O' ring is installed, confirm proper material, size and durometer and apply a small amount of 'Krytox' or equal lubrication and install. Assure shaft surface is clean and free of any burrs and carefully install sleeve being sure 'O' ring is not damaged.
- Install the stationary seal face into gland evenly to ensure face is not cocked. Assure that any finger prints are removed with alcohol and lint free towel.
- Stationary 'O' ring checks – confirm proper material, durometer and inspect for discontinuities.
- Install gland to stuffing box using opposite and even tightening technique. Prior to doing so, a sweep of the seal chamber face is required to ensure proper sealing between gasket/'O' ring on seal gland and seal chamber. A dial indicator should be attached to the shaft and the shaft will be rotated to cover a complete sweep of the seal chamber face. TIR should be no more than 0.005'.
- After seal is completely installed, conduct a static pressure check. Consult the seal vendor for the proper pressure if static limit is not indicated on seal drawing. If any leaks are observed, consult seal vendor immediately.
- Typical installation errors:
- Contaminated faces – Can affect face flatness, resulting in excessive leakage. In addition, certain oils can set-up like adhesives and pull material out of the carbon primary ring faces after start-up, resulting in premature seal failures.
- Applying lubricant to secondary seals which can enter primary seal face areas – the same results apply as above.
- Not setting proper spring compression: excessive compression – more seal wear. Low compression – will not provide sufficient face contact during start-up and stand by conditions allowing excessive leakage.

- Improper installation of mating ring into the gland resulting in excessive leakage due to improper face contact. Note that if back of mating ring is visibly 'not flat', the same result can apply.
- Stuffing box face TIR should be checked with dial indicator on shaft. Seal chamber face flatness should be no more than 0.005' TIR.
- Installation of gland to stuffing box not using equal and opposite tightening technique. Failure to tighten the nuts/capscrews properly can cock the mating ring, which will not allow for proper contact between the faces and result in excessive leakage between the seal faces.
- Sleeve 'O' ring cut during installation, resulting in excessive leakage between shaft and sleeve.

Cartridge mechanical seal installation

- Refer to pump vendor manual and maintenance records to ensure proper cartridge seal will be used.
- Assure the seal chamber is cleaned and shaft is de-burred to remove any sharp material that could damage seal sleeve 'O' ring.
- Assure seal chamber is isolated from all flush and quench connections systems. Plastic plugs need to be installed in all flush/quench after isolation of piping/tubing up to the point of the piping/tubing reinstallation.
- Assure that the proper durometer, material and size 'O' ring is supplied and lubricate seal sleeve 'O' ring with 'Krytox' or equivalent lubricant.
- Slide sleeve over shaft carefully.
- Install gland to seal chamber using equal and opposite tightening technique. Prior to doing so, a sweep of the seal chamber face is required to ensure proper sealing between gasket/'O' ring on seal gland and seal chamber. A dial indicator should be attached to the shaft and the shaft will be rotated to cover a complete sweep of the seal chamber face. TIR should be no more than 0.005'.
- Tighten set screws to engage shaft/sleeve by equal and opposite technique. Assure that a clean shaft surface is available for all set screw locations. If not, use a file and emery cloth to remove any damaged spots. Be sure to completely clean shaft to remove all debris.
- Completely remove setting clips. If setting clips are slid up on the gland, they can potentially fall back down to the sleeve or drive collar resulting in an ignition source (the writer has experienced fires that have started due to this issue).

9

Dry Gas Seal Best Practices

Introduction

Dry gas seals were introduced in the late 1960s but did not gain wide acceptance until 20 years later as the ultimate solution to compressor seal oil system problems. The dry gas seal was marketed as a 'simple' solution to seal oil system reliability issues, but in reality, when one considers all the possible process and operational related issues, its complexity rivals that of the oil systems. There is no doubt, however, that a properly specified, selected and designed dry gas seal system will positively prevent the ingress of oil into the process system and deliver high degrees of safety and reliability.

This chapter will therefore present the best practices used since the mid-1980s for developing and selecting DGS systems of the highest degree of safety and reliability.

Best Practice 9.1

To ensure optimum safety and reliability of dry gas seal systems, end users must be proactive in the project phase or during seal system modifications to specify operating parameters.

The main parameters fall into the following categories:

- All possible operating, start-up and upset conditions on the seal data sheet
- Required system design details by incorporating all site, company and industry lessons learned into the project or revamp specification
- A detailed (P&ID) and data sheet to quoting machinery vendors that will completely specify system and component design

Allowing the EP&C (contractor) and/or machinery vendor to design the dry gas seal system will expose the plant to safety and reliability issues that cannot be known by other parties.

Following the guidelines completely in this best practice and requiring compliance with all specified details will ensure a safe and trouble-free system of the highest reliability.

Lessons Learned

Failure to consider specific plant operating conditions and seal system lessons learned has resulted in dry gas seal systems of low MTBF (less than 12 months) and large revenue losses.

The following examples highlight omitted details in dry gas seal specifications that have resulted in seal MTBFs of less than 12 months:

- Failure to identify the actual gas properties (sour gas, gas composition)
- Failure to identify saturated seal gas conditions at start-up, upset or operating conditions
- Failure to properly specify maximum flare header pressure
- Failure to define the actual dew point of supplied nitrogen for intermediate and separation gas
- Failure to prohibit the use of orifices in the secondary vent resulting in seal pressure reversals
- Failure to specify oil sampling devices in the secondary seal vent port (sight glasses, valves or automatic drainers) leading to secondary seal oil contamination and eventual failure.

Benchmarks

This best practice has been used since the late 1990s to specify dry gas seal system requirements during projects and for field modifications. This approach has resulted in dry gas seal systems of the highest safety levels and reliability (seal MTBFs greater than 90 months).

B.P. 9.1. Supporting Material

Dry gas seal (DGS) systems have been used for the past two decades, and are specified by many end users as the seal of choice for most compressor applications. One would therefore think that seal and system designs are well-known and proven. However, experience shows that failures are still quite common. For instance, in 2007, FAI dealt with nearly 50 DGS failures.

These failures raise several questions. Are they all caused by 'foreign material' contamination or ingestion? Are they connected with improper seal selection or unreliable system hardware? Who is responsible: seal vendors, compressor vendors, or end users?

In reviewing DGS failures experienced in 2006 and previous years, the conclusion was drawn that in the majority of cases, the root cause is that the seal and system configuration were not designed to handle all the actual site operating conditions, including start-up, shut-down and upsets that should and could have been anticipated.

The end user has the most complete knowledge of the process and plant operating procedures. Therefore, he or she needs to be proactive in terms of project DGS requirements, and specify the type of seal and system most suited to the plant and application, based on his or her knowledge and experience. Seal and compressor vendor input and experience are obviously required, but neglecting to evaluate the proposed system in detail against all operating modes subjects the user to the risk of unacceptable downtime and revenue losses, particularly in the 'mega plants' being built today.

Figure 9.1.1 shows a recommended 'best practice' P&ID for a tandem dry gas seal system in a critical (un-spared) application

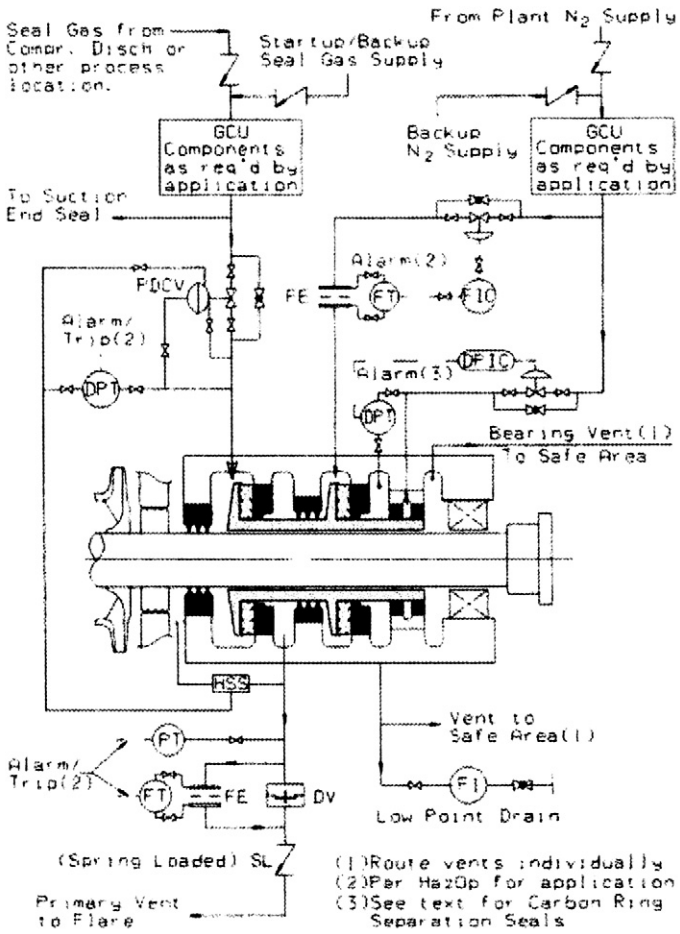


Fig 9.1.1 • Best Practice tandem seal P&ID

used in a large plant of high daily revenue (greater than $1 MM/day).

The reliability of critical equipment is dependent on the reliability of each component in every auxiliary system connected with the critical equipment unit. How do we maximize critical equipment reliability? The easiest way is to eliminate the auxiliary systems. Imagine the opportunity to eliminate all of the components; pumps, filters, reservoirs, etc. and thereby increase reliability and hopefully, the safety of the equipment. The gas seal as used in compressor applications affords the opportunity to achieve these objectives. However, the gas seal is still part of a system and the entire gas seal system must be properly specified, designed, maintained and operated to achieve the objectives of optimum safety and reliability of the critical equipment.

In this section, the principles of gas seal design will be discussed and applied to various gas seal system types. In addition, best practices will be discussed for saturated gas systems as well as shutdown philosophies.

System function

The function of a gas seal system is naturally the same as a liquid seal system. The function of a fluid seal system, remembering that a fluid can be a liquid or a gas, is to continuously supply clean fluid to each specified seal interface point at the required differential pressure, temperature, and flow rate. Therefore, one would expect the design of a gas seal and a liquid seal to be very similar, which, in fact, they are. Then why are their systems so different?

Comparison of a liquid and gas sealing system

Figure 9.1.2 shows a liquid sealing system. Compare this system to Figure 9.1.3 which shows a gas seal system, if the same compressor were retrofitted for a gas seal. WOW! What a difference. Why are there such a small number of components for the gas seal system? As an aid, refer to Figure 9.1.4, which shows a typical pump liquid flush system as specified by the American Petroleum Institute. This system incorporates a liquid mechanical seal and utilizes pump discharge liquid as a flush for the seal. Refer now to Figure 9.1.3 and observe the similarities. It should be evident that a gas seal system is simplified in compressor applications over a liquid seal system, merely because the gas seal utilizes the process fluid. This is exactly the same case for a pump. By using the process fluid, and not a liquid, one can eliminate the need to separate liquid from a gas, thereby totally eliminating the need for a liquid supply system and the need for a contaminated liquid (sour oil) drain system.

Referring back to Figure 9.1.2, therefore, we can see that the following major components are eliminated:

1. The seal oil reservoir
2. The pumping units
3. The exchangers
4. The temperature control valves
5. The overhead tank
6. The drain pot
7. The degassing tank
8. All control valves
9. A significant amount of instrumentation

Referring back to the function definition of the gas seal system, all requirements are met. 'Continuously supplying fluid' is achieved by utilizing the discharge pressure of the compressor. The requirements for 'specified differential pressure, temperature and flow rate' are met by the design of the seal itself, which can accommodate high differential pressures, high temperatures, and is sized to maintain a flow rate that will remove

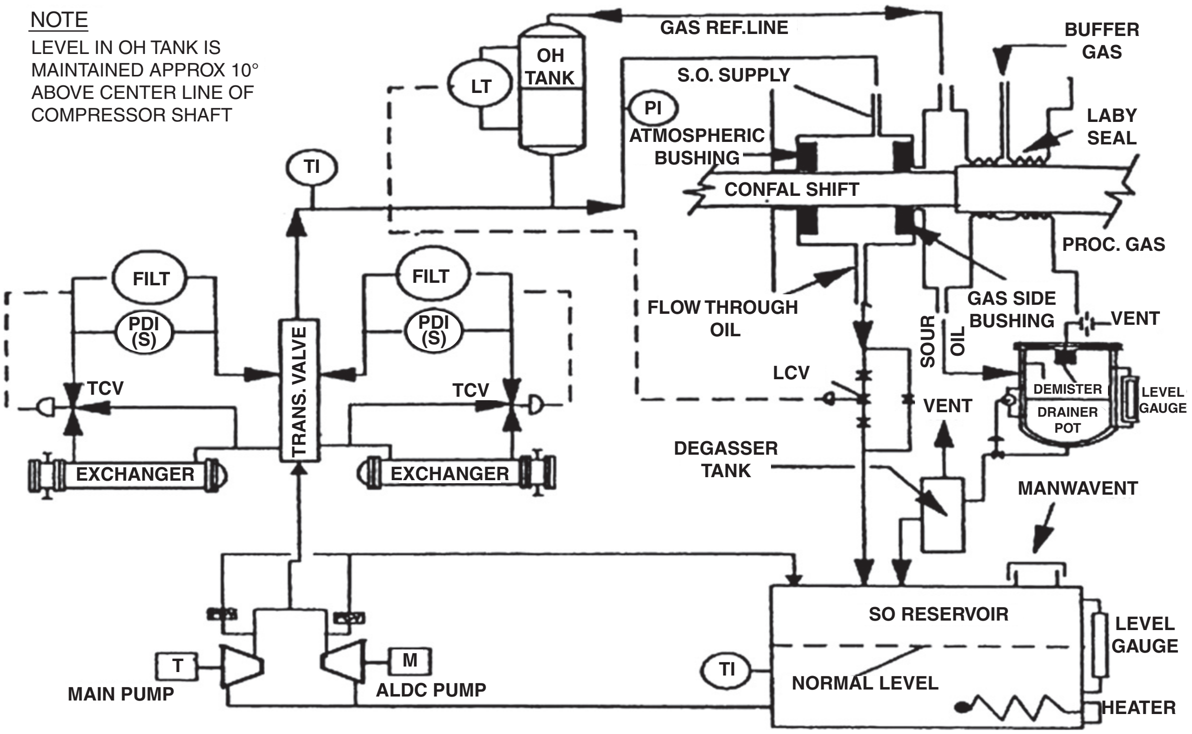


Fig 9.1.2 • Typical seal oil system for clearance bushing seal (Courtesy of M.E. Crane, Consultant)

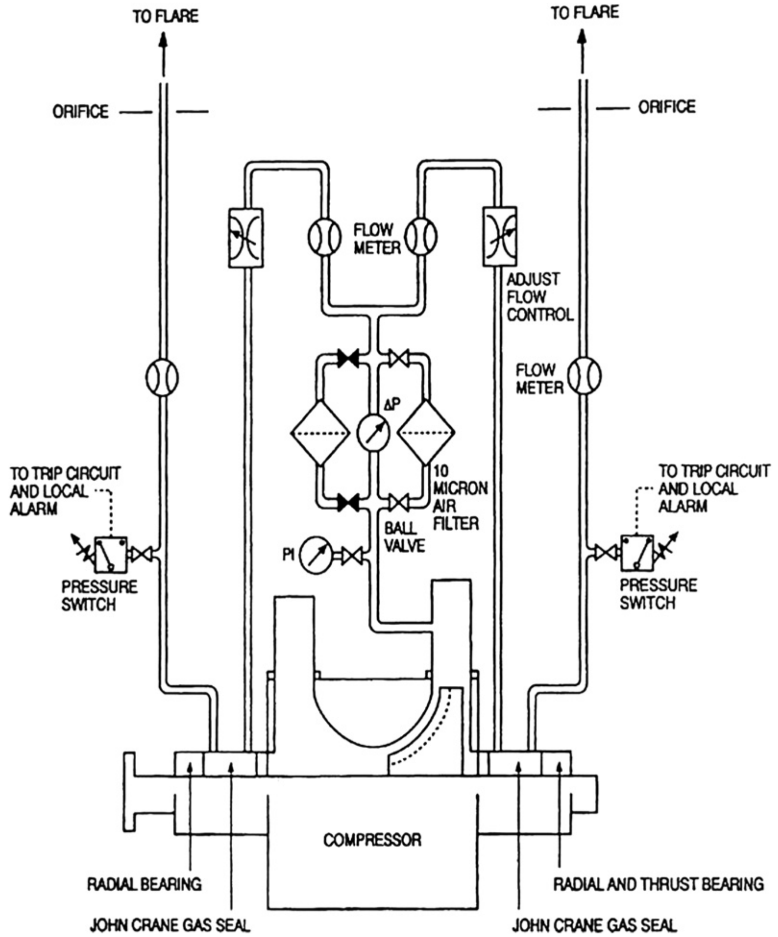


Fig 9.1.3 • Typical gas seal system for dry air or inert gas (Courtesy of John Crane Co.)

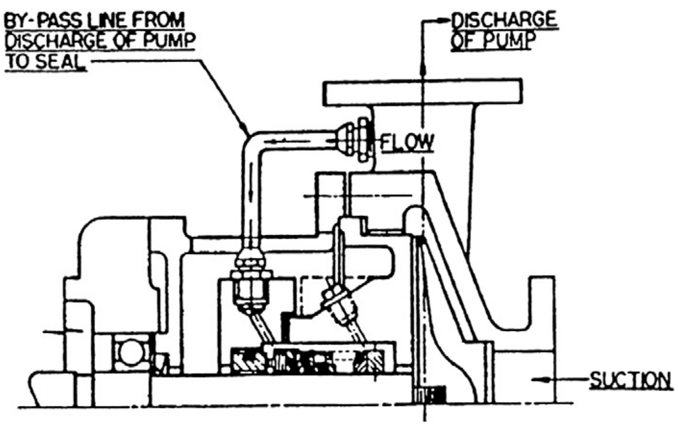


Fig 9.1.4 • Liquid seal flush (Courtesy of John Crane Co.)

frictional heat necessary to maintain seal reliability. The only requirement not met is that of supplying a clean dry fluid, and this can be seen in Figure 9.1.3. This requirement is met by using a dual system coalescing filter.

When one considers all the advantages, the next question to ask is, 'Okay, what are the disadvantages?' Naturally, there are disadvantages. However, proper design of the gas seal system can minimize and eliminate many of these. Do not forget that the requirements for any system mandate proper specification, design, manufacture, operation and maintenance. One can never eliminate these requirements in any critical equipment system.

Considerations for system design

As mentioned above, there are disadvantages to a gas seal system, which are not insurmountable but must be considered in the design of such a system. These considerations are as follows:

Sensitivity to dirt – since clearances between seal faces are usually less than 0.0005 inch and seal design is essential to proper operation, the fluid passing between the faces must be clean (5–10 microns maximum particle size). If it is not, the small grooves (indentations) necessary for seal face separation will become plugged, thus causing face contact and seal failure.

Sensitivity to saturated gas – saturated fluids increase the probability of groove (indentation) blockage.

Lift-off speed – as will be explained below, a minimum speed is required for operation. Care must be taken in variable speed operation to ensure that operation is always above this speed. It is recommended that the seal test be conducted for a period at turning gear speed to confirm proper 'lift off' followed by seal face inspection.

Positive prevention of toxic gas leaks to atmosphere – since all seals leak, the system must be designed to preclude the

possibility of toxic of flammable gas leaks out of the system. This will be discussed in detail below.

Possible oil ingestion from the lube system – a suitable separation seal must be provided to eliminate the possibility of oil ingestion from the bearings. Whenever a gas seal system is utilized, the design of the critical equipment by definition incorporates a separate lube oil and seal system. Consideration must be given during the design or retrofit phases to the separation between the liquid (lube) and gas seal system.

'O' ring (secondary seal components) design and maintenance – most seal vendors state that 'O' ring life is limited, and they should be changed every five years for operating seals as well as spare seals. Experience has shown that dry gas 'O' ring seals can exceed this limit. It is recommended that seal vendors be required to provide references for similar applications prior to making a decision to change out the seals after five years.

If all of the above considerations are incorporated in the design of a gas seal system, its reliability has the potential to exceed that of a liquid seal system and the operating costs can be reduced.

Before moving to the next section, however, one must consider that the relative reliability of gas and liquid seal systems is a function of proper specification, design, etc. as mentioned previously. A properly designed liquid seal system that is operated and maintained can achieve reliabilities comparable to a gas seal system. Also, when one considers the operating costs of the two systems, various factors must be considered. While the loss of costly seal oil is positively eliminated, with a gas seal system (assuming oil ingestion from the lube system does not occur) the loss of process gas, while minimal, can be expensive. It is argued that the loss of process gas from a liquid seal system through drainer and degassing tank vents is also significant. While this may be true in many cases, a properly specified, designed and operated liquid seal system can minimize process gas leakage such that it is equal or even less than that of a gas seal.

There is no question that gas seal systems contain far fewer components and are easier to maintain than liquid seal systems. These systems will be used extensively in the years ahead. The intention of this discussion is to point out that existing liquid seal systems which cannot be justified for retrofit, or cannot be retrofitted easily, can be modified to minimize outward gas leakage and optimize safety and reliability.

Dry gas seal design

Principles of operation

The intention of this sub-section is to present the principles of operation of a dry gas seal in a conceptual form. The reader is directed to any of the good literature available on this subject for a detailed review of gas seal design.

Refer to Figure 9.1.5, which shows a mechanical seal used in pump applications, while Figure 9.1.6 shows a dry gas mechanical seal utilized for compressor applications. The seal designs appear to be almost identical. Close attention to Figure 9.1.6, however, will show reliefs of the rotating face of the seal.

Considering that both seals operate on a fluid may give some hint as to why the designs are so very similar. The objective of seal design is to positively minimize leakage while removing frictional heat, in order to obtain reliable, continuous operation of the seal. In a liquid application, the heat is removed by the fluid which passes between the rotating and stationary faces and the seal flush and changes from a liquid to gaseous state (heat of vaporization). This is precisely why all seals are said to leak and explains the recent movement in the industry to seal-less pumps in toxic or flammable service. If the fluid between the rotating faces now becomes a gas, its capacity to absorb frictional heat is significantly less than that of a liquid. Therefore an 'equivalent orifice' must continuously exist between the faces to reduce friction and allow a sufficient amount of fluid to pass and thus take away the heat. The problem obviously is how to obtain this 'equivalent orifice'. There are many different designs of gas seals. However, regardless of the design, the dynamic action of the rotating face must create a dynamic opening force that will overcome the static closing forces acting on the seal to create an opening and hence 'equivalent orifice'.

Refer to Figure 9.1.7 which shows a typical gas dry seal face. Notice the spiral grooves in this picture; they are typically machined at a depth of 100–400 micro inches.

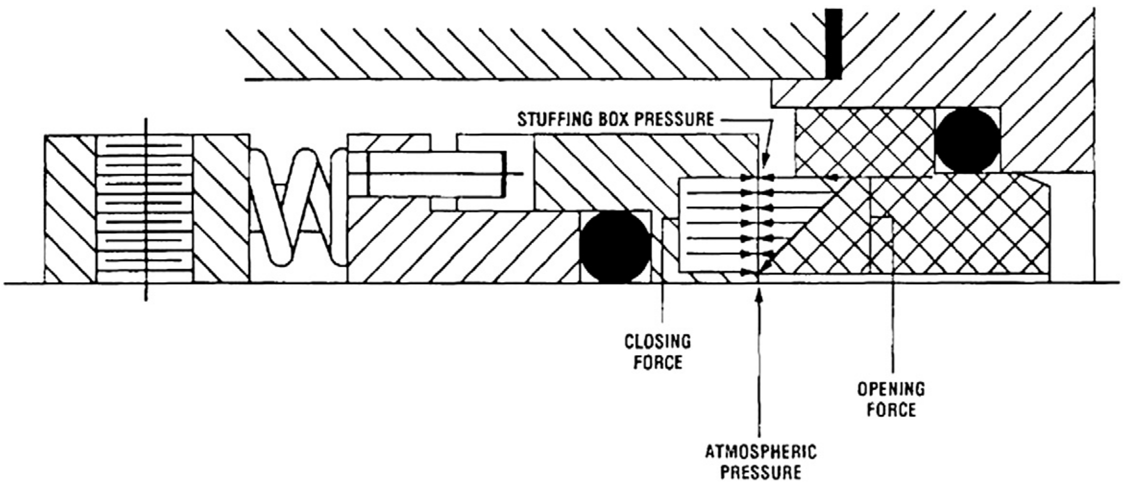


Fig 9.1.5 • Typical pump single mechanical seal

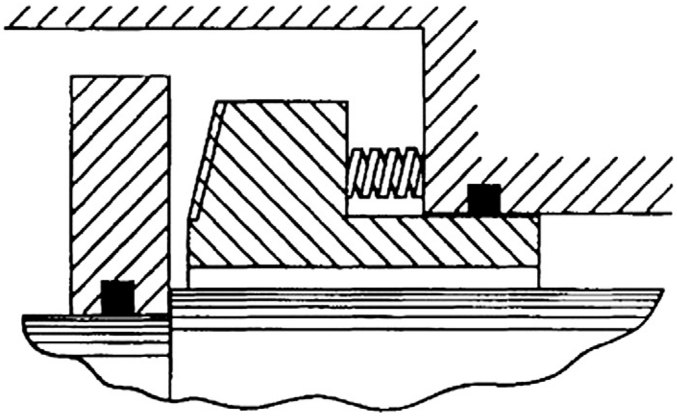


Fig 9.1.6 • Typical design for curved face – spiral groove non-contact seal; curvature may alternately be on rotor (Courtesy of John Crane Co.)

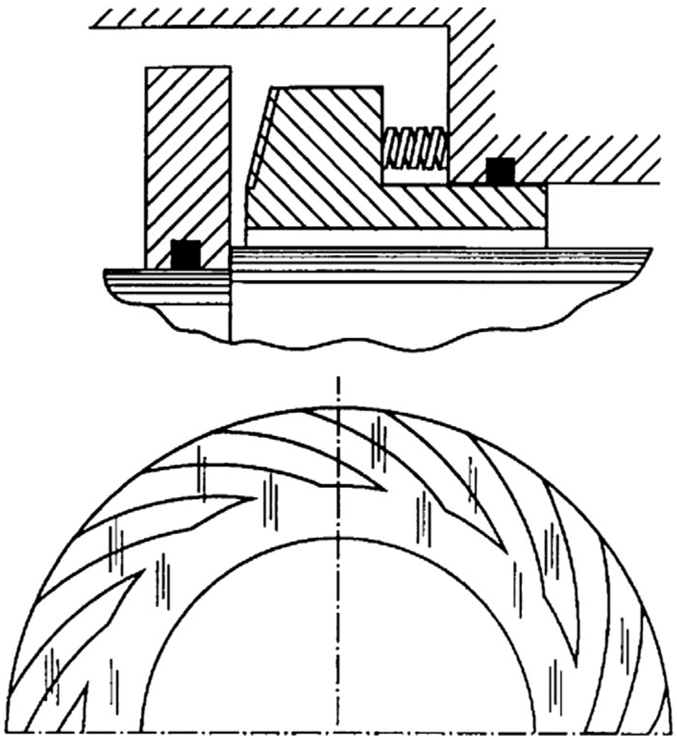


Fig 9.1.7 • Dry gas seal. Top: typical design for curved face – spiral groove non-contact seal; curvature may alternately be on rotor; Bottom: Typical spiral groove pattern on face of seal typical non-contact gas seal (Courtesy of John Crane Co.)

When rotating, these vanes create a high head, low flow, impeller that pumps gas into the area between the stationary and the rotating face, thereby increasing the pressure between the faces. When this pressure is greater than the static pressure holding the faces together, the faces will separate, thus forming an equivalent orifice. In this specific seal design, the annulus below the vanes forms a tight face such that under static (stationary) conditions, zero leakage can be obtained if the seal is properly pressure-balanced. Refer to Figure 9.1.8 for a force diagram that shows how this operation occurs.

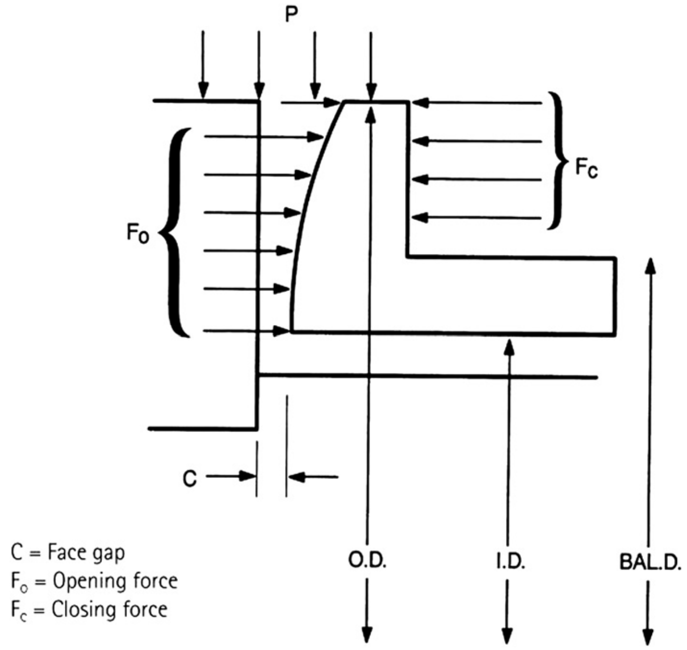


Fig 9.1.8 • Hydrostatic force balance on seal stator ($F_C = F_O$) (Courtesy of John Crane Co.)

In Figure 9.1.7, the rotation of the face must be counter-clockwise, to force the gas into the passages and create an opening (F_o) force. This design is known as a 'uni-directional' design and requires that the faces always operate in this direction. Alternative face designs are available that all rotate in either direction and they are known as 'bi-directional' designs.

Ranges of operation

Essentially, gas seals can be designed to operate at speeds and pressure differentials equal to or greater than those of liquid seals. Present state-of-the-art (2010) limits seal face differentials to approximately 17,250 kPa (2,500 psi) and rubbing speeds to approximately 122 meters/second (400 feet/second). Temperatures of operation can reach 538°C (1,000°F). Where the seal face differential exceeds these values, seals can be used in series (tandem) to meet specifications, provided sufficient axial space is available in the seal housing.

Leakage rates

Since the gas seal when operating forms an equivalent orifice, whose differential is equal to the supply pressure minus the seal reference pressure, there will always be a certain amount of leakage. Refer to Figure 9.1.9 for leakage graphs.

It can be stated in general that, for most compressor applications with suction pressures of the order of 3,450 kPa (500 psi) and below, leakage can be maintained at around one standard cubic foot per minute per seal. For a high pressure application (17,250 kPa or 2,500 psi), differential leakage values can be as high as 8.5 Nm^3/hr (5 scfm) per seal. As in any seal design, the total leakage is equal to the leakage across the seal faces and any leakage across secondary seals ('O' rings, etc.). There have been reported incidences of explosive 'O' ring failure on rapid decompression of systems incorporating gas seals,

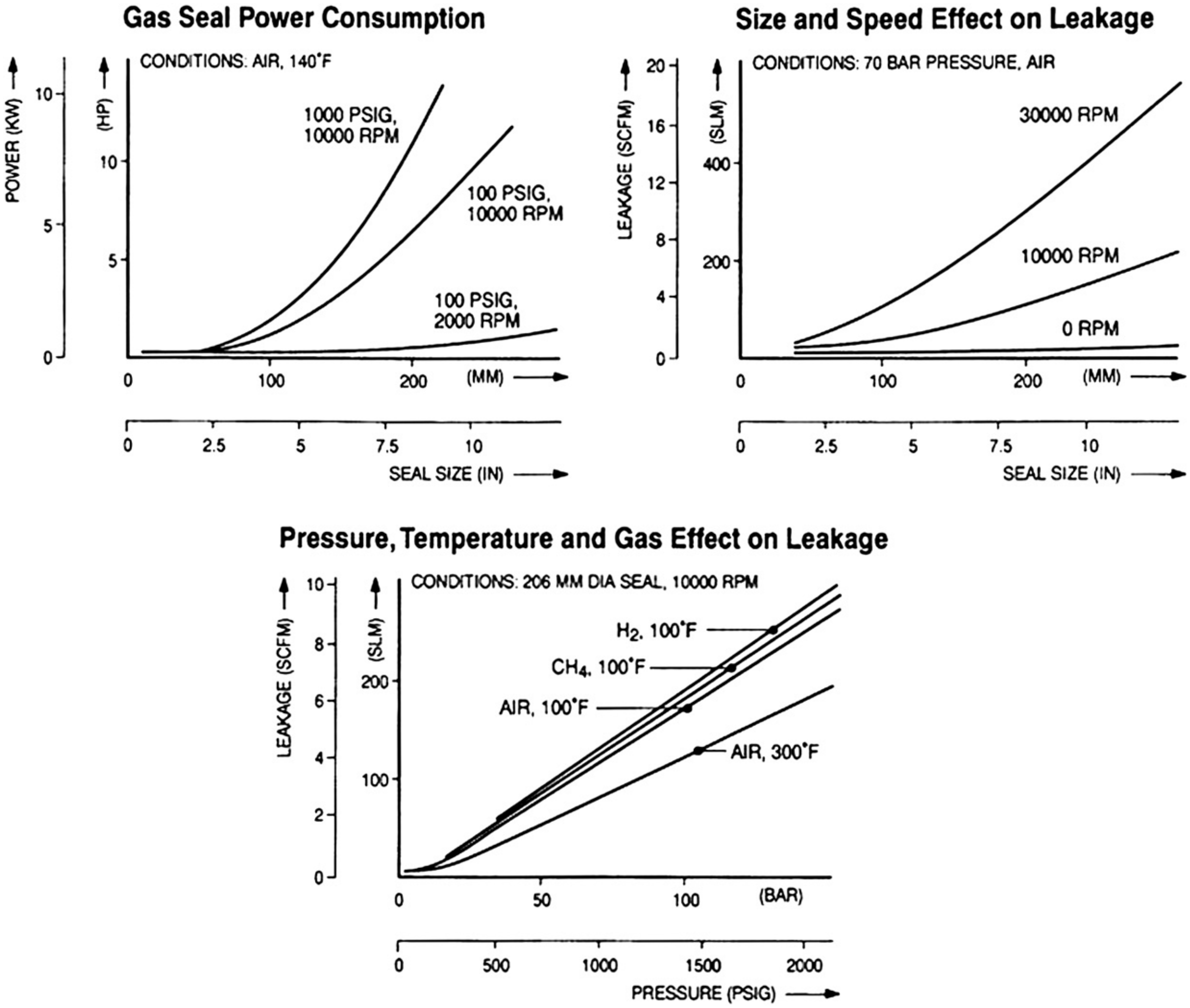


Fig 9.1.9 • Dry gas seal leakage rates (Courtesy of John Crane Co.)

thus resulting in excessive leakage. Consideration must be given to the system under consideration, to tailor system decompression times to meet the requirements of the secondary seals. As previously mentioned, all gas seals will leak, but not until the face 'lifts off'. This speed known, oddly enough, as 'lift-off speed' is usually less than 500 rpm. Caution must be exercised in variable speed applications to ensure the system prevents the operation of the variable speed driver below this minimum lift-off value. One recommendation concerning instrumentation is to provide one or two thermocouples in the stationary face of each seal to measure seal face temperature. This information is very valuable in determining lift-off speed and condition of the grooves in the rotating seal face. Any clogging of these grooves will result in a higher face temperature and will be a good indication of requirement for seal maintenance.

Gas seal system types

As mentioned in this section, in order to ensure the safety and reliability of gas seals, the system must be properly specified and designed. Listed below are typical gas seal system applications in use today.

Low/medium pressure applications – dry air or inert gas

Figure 9.1.3 shows such a system. This system, incorporating a single dry gas seal, is identical to that of a liquid pump flush system incorporating relatively clean fluid that meets the requirements of the seal in terms of temperature and pressure. This system takes the motive fluid from the discharge of the compressor through dual filters (ten microns or less) incorporating a differential pressure gauge and proportions equal flow through flow meters to each seal on the compressor. Compressors are usually pressure balanced such that the pressure on each end is approximately equal to the suction pressure of the compressor. The clean gas then enters the seal chamber and has two main paths:

A. Through the internal labyrinth back to the compressor. Note that the majority of supplied gas takes this path for cooling purposes (99%).

B. Across the seal face and back to either the suction of the compressor or to vent.

Since the gas in this application is inert, it can be vented directly to the atmosphere or can be put back to the compressor

suction. It must be noted, however, that this port is next to the journal bearing. Therefore a means of positively preventing entry of lube oil into this port must be provided in order to prevent the loss of lube oil or prevent the ingestion of lube oil into the compressor if this line is referenced back to the compressor suction. A suitable design must be incorporated for this bushing. Typically called a disaster bushing, it serves the dual purpose of isolating the lube system from the seal system, and providing a means of minimizing the leakage of process fluid into the lube system in the event of a gas seal failure. In this system, a pressure switch upstream of an orifice in a vent line is used as an alarm and a shutdown to monitor flow. This switch uses the concept of an equivalent vessel in that increased seal leakage will increase the rate of supply versus demand flow in the equivalent vessel (pipe) and result in a higher pressure. When a high flow is reached, the orifice and pressure switch setting are thus sized and selected to alarm and shut down the unit if necessary. As in any system, close attention to changes in operating parameters is required. Flow meters must be properly sized and kept clean, so that relative changes in the flows can be detected in order to adequately plan for seal maintenance.

High pressure applications

In this application, for pressures in excess of 6,895 kPa (1,000 psi), a tandem seal arrangement or series seal arrangement is usually used. Since failure of the inner seal would cause significant upset of the seal system, and large amounts of gas to escape to the atmosphere, a backup seal is employed. Refer to Figure 9.1.10, which shows a triple dry gas tandem seal. For present designs up to 17,250 kPa (2500 psi), double tandem seals are proven and used.

The arrangement is essentially the same as low/medium pressure applications except that a backup seal is used in place of the disaster bushing. Most designs still incorporate a disaster bushing between the backup seal and the bearing cavity, known these days as the barrier seal. Attention in this design must be given to control of the inter-stage pressure between the primary and backup seal. Experience has shown that low differentials across the backup seal can significantly decrease its life. As in the case of liquid seals, a minimum pressure in the cavity between the seals of 172–207 kPa (25–30 psi) is usually specified. This is achieved by properly sizing the orifice in the vent or reference line back to the suction to ensure this pressure is maintained. All instrumentation and filtration are identical to that of the previous system.

Dual seal and system options for toxic and/or flammable gas applications

There are many field-proven options which are currently available for use in toxic and/or flammable gas applications. In this section we will discuss the following systems:

- Tandem seals for dry gas applications
- Tandem seals for saturated gas applications
- Tandem seals with interstage labyrinth and nitrogen separation gas
- Double seal system for dry gas or saturated gas applications

Tandem seals for dry gas applications

The tandem seal arrangement for this application is shown in Figure 9.1.11 and a schematic of this seal in the compressor seal housing is shown in Figure 9.1.12. Gas from the compressor discharge enters the port closest to the compressor (labyrinth end) and the majority of the gas enters the compressor through this labyrinth. To ensure that process gas, which is not treated by the

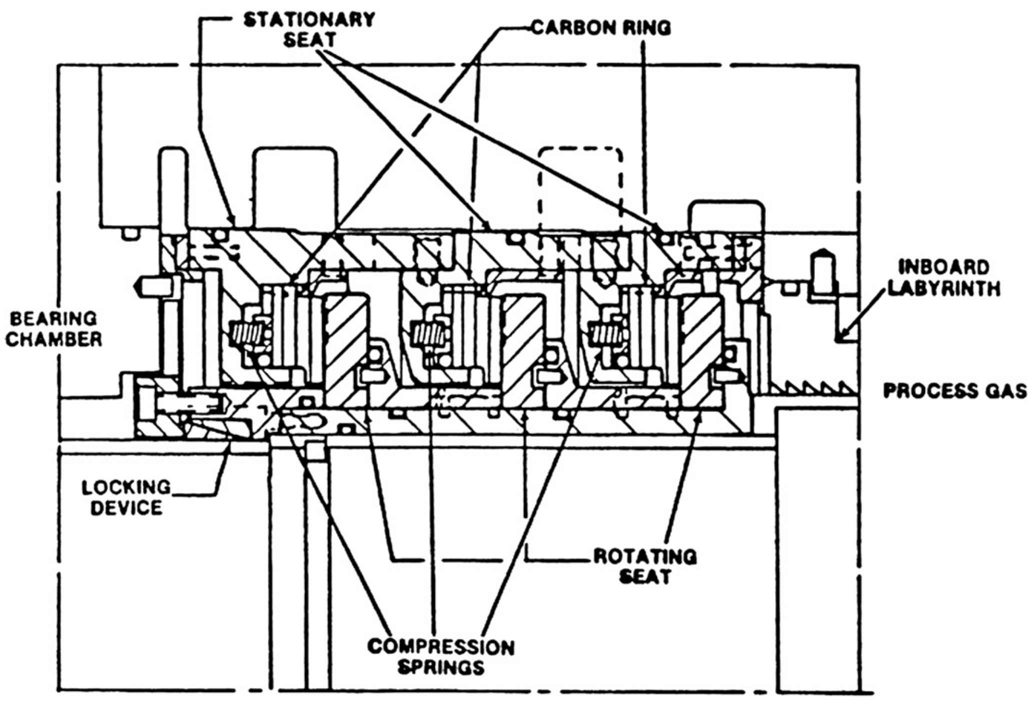


Fig 9.1.10 • Dry gas seal: a triple tandem dry gas seal arrangement (Courtesy of Dresser-Rand Corp.)

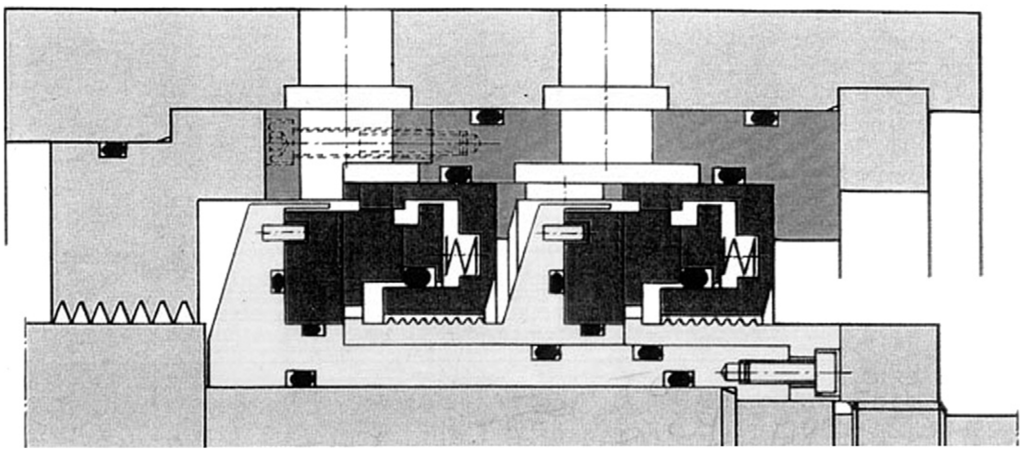


Fig 9.1.11 • Tandem seal (Courtesy of Flowserve Corp.)

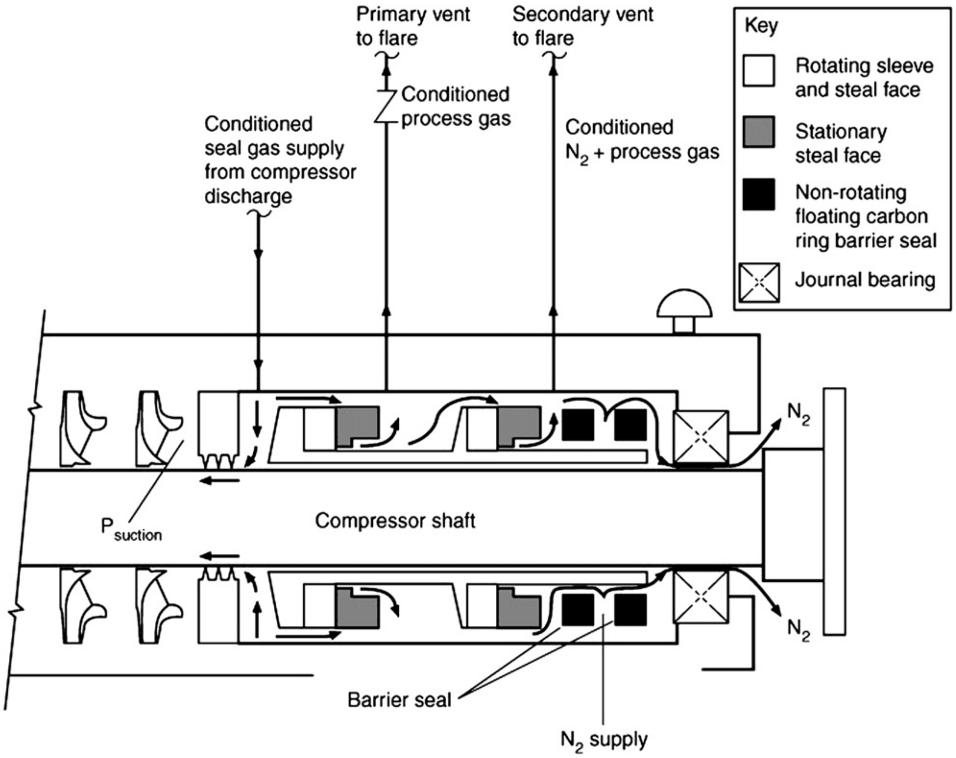


Fig 9.1.12 • Tandem seal and barrier seal typical housing arrangement

dry gas system, does not enter the seal chamber, velocities across the labyrinth should be maintained between 6–15 m/sec (20–50 ft/sec). It is the writer's experience that considering labyrinth wear, the design should be closer to 15 m/sec (50 ft/sec).

Approximately 1.7–3.4 Nm^3/hr (1–2 scfm) flow (standard cubic feet per minute) leak across the first tandem seal faces (primary seal) and exit through the primary vent. Based on the backpressure of the primary vent system, 1.7 Nm^3/hr (1 scfm) or less will pass through the second tandem seal faces (secondary seal) and exit through the secondary vent. To ensure that oil mist from the bearing housing does not enter the dry gas seal chamber and that seal gas does not escape to atmosphere, an additional barrier seal is used and provided with pressurized nitrogen at approximately 35 kPa (5 psi).

A typical seal system for this arrangement is shown in Figure 9.1.13. As previously mentioned, dry gas seal reliability depends on the condition of the gas entering the seal faces. The function of the seal gas supply system for any dry gas seal option is to continuously supply clean, dry gas to the seal faces. During start-up, when the compressor is not operating with sufficient pressure to supply the seals, an alternate source of gas or a gas pressure booster system should be provided. These items are shown in Figure 9.1.13 and are typical for any type of dry gas seal application. Note that the following options exist regarding

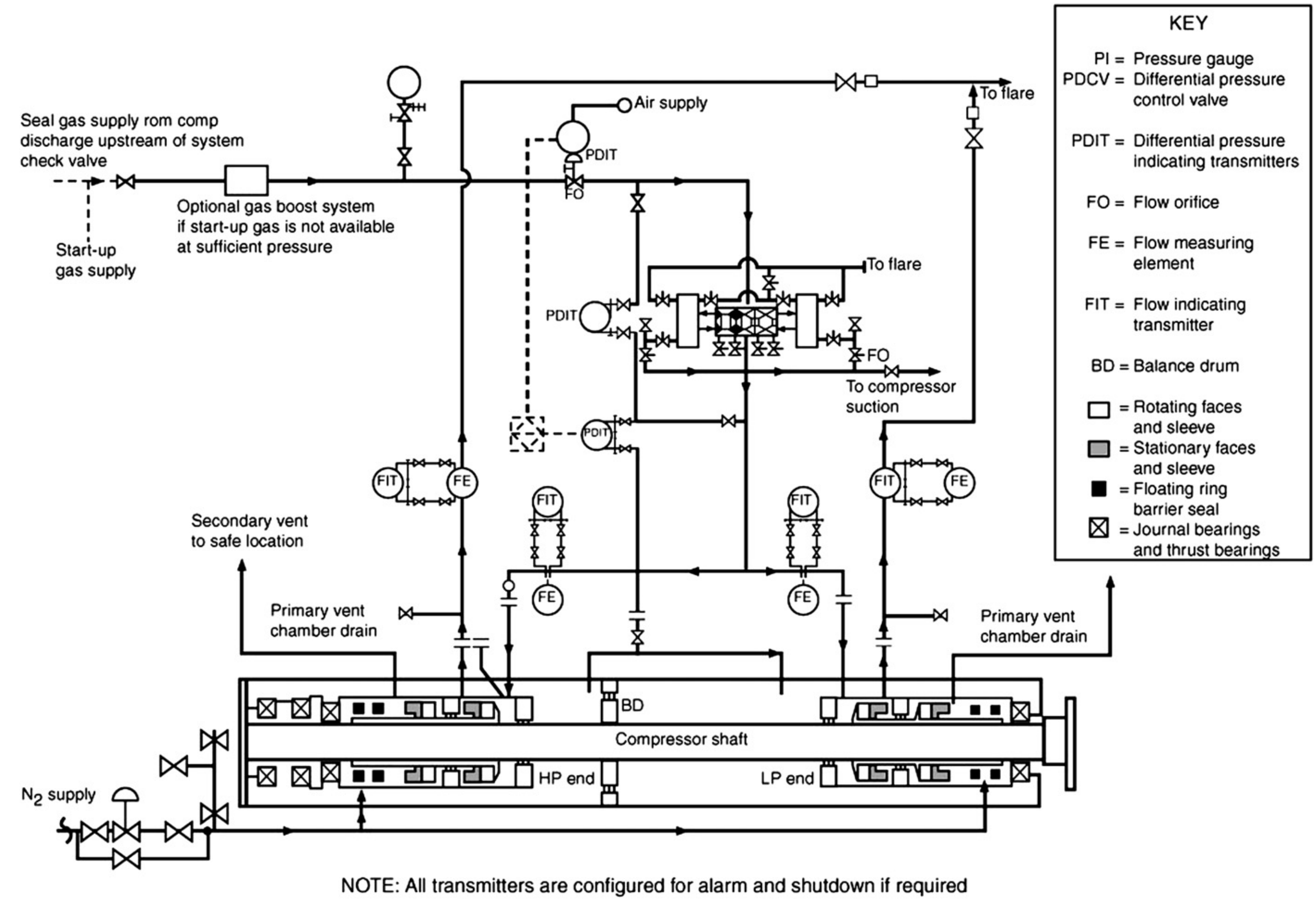


Fig 9.1.13 • Typical tandem seal system for saturated process gas

the primary, secondary vent and barrier seal instrumentation and components:

- Primary seal vent triple redundant (two-out-of-three voting) flow or differential pressure alarm and shutdown.
- Primary seal vent rupture discs in parallel with vent line to rupture at a set pressure and prevent excessive pressure to the secondary seal on primary seal failure.
- Spring loaded exercise valves in the primary vent line to exert a backpressure on the primary seal to close the faces in the event of dynamic 'O' ring hang-up.
- Secondary vent line flow or differential pressure alarms and trips.
- Barrier seal supply pressure alarm and permission not to start the lube oil system until barrier seal minimum pressure is established.

Tandem seals for saturated gas applications

The tandem seal arrangement for this application can be exactly the same as that shown in Figures 9.1.11 and 9.1.12 for the dry gas application. The changes required for a saturated gas are solely in the seal system. A typical system is shown in Figure 9.1.14 and incorporates a cooler, separator and heater in addition to the normal components used for a dry gas application to ensure that saturated gas does not enter the seal chamber. Typical values for the cooler are to reduce the gas temperature to 30°F below the saturation temperature of the gas. The typical dimensions for the separator vessel, complete with a demister, are 460 mm (18 inches) diameter and 1.8 meters (6 feet) high. The typical requirements for the heater are to reheat the gas to 15°C (30°F) above the saturation temperature. Temperature transmitters are provided upstream and downstream of the cooler, and downstream of the heater. As a precaution, in the event of cooler or heater malfunction, a dual filter/coalescer, complete with a drain back to the suction, is provided.

Tandem seals with interstage labyrinth

The present (2010) industry 'best practice' tandem seal arrangement for dry or saturated gas applications is shown in Figure 9.1.15. This arrangement features a labyrinth between the primary and secondary seals. This action ensures that gas vented from the secondary seal will always be nitrogen, since the nitrogen supplied between the primary and secondary seals is differential-pressure controlled to always be at a higher pressure than the primary seal vent, thus ensuring that only nitrogen will be in the chamber between the primary and secondary seals. Figure 9.1.16 shows a typical nitrogen supply system used with this tandem seal configuration.

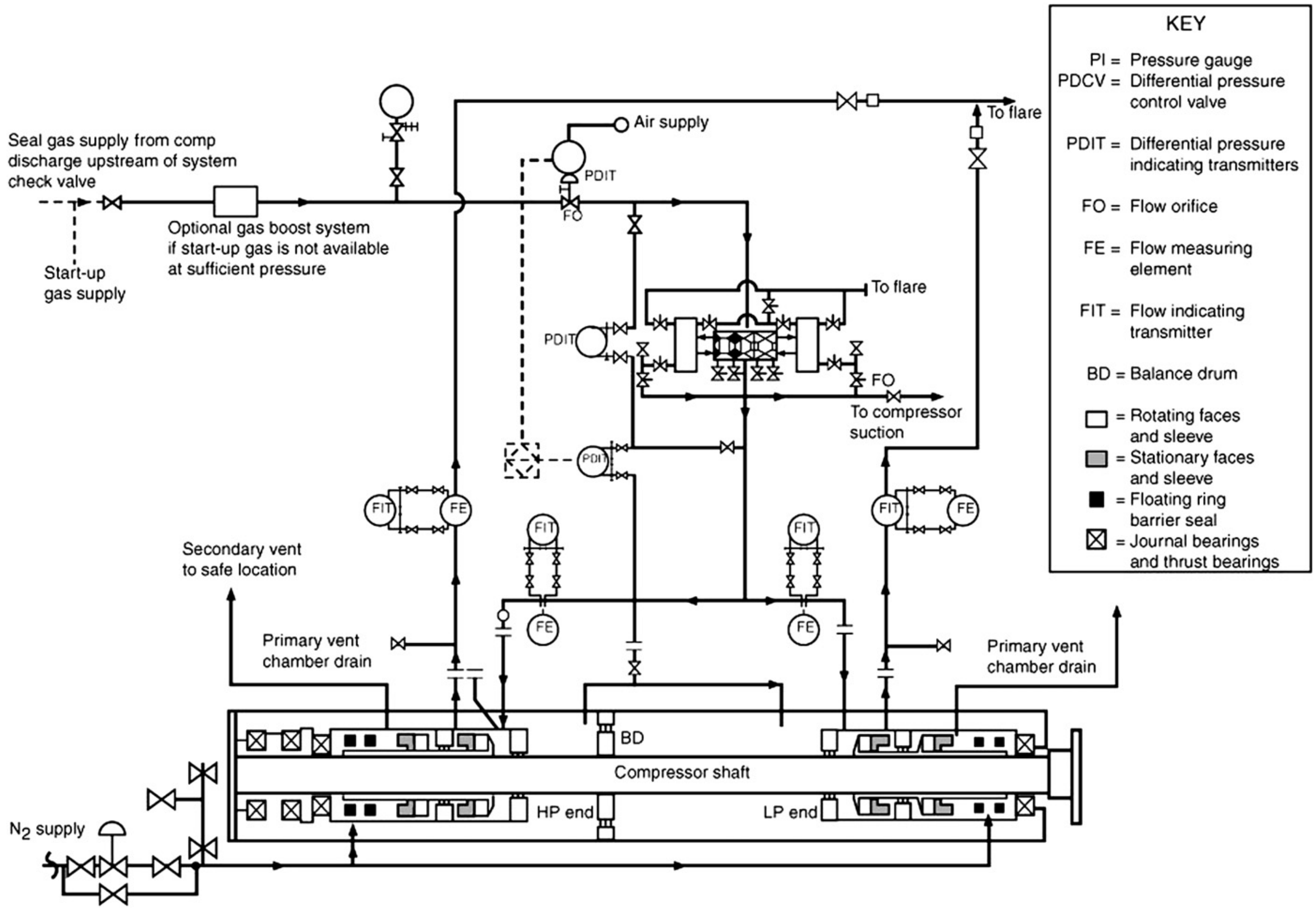


Fig 9.1.14 • Typical tandem seal system for dry process gas

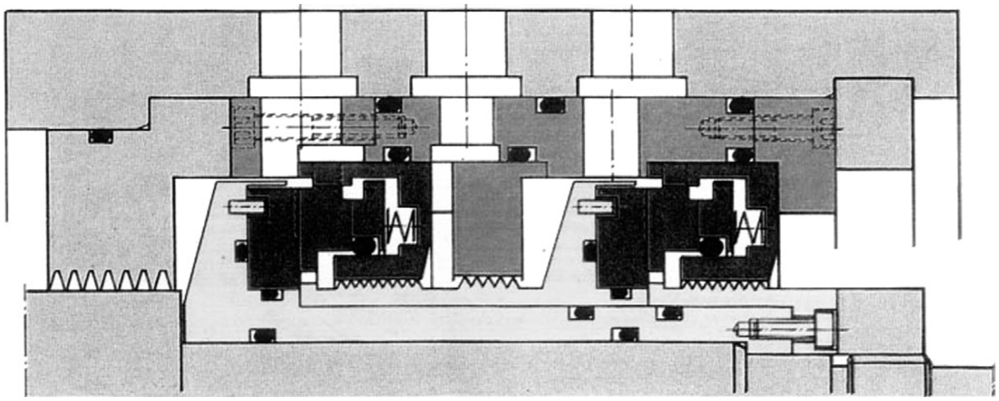


Fig 9.1.15 • Tandem seal with interstage labyrinth (Courtesy of Flowserve Corp.)

Double seal system for dry gas or saturated gas application

Figure 9.1.17 depicts a double seal, used in either dry gas or saturated gas applications where the process gas is not permitted to exit the compressor case. For this application process gas can be used, after it is conditioned, or an external source if it is compatible with the process gas. If the gas used between the seals is toxic or flammable, a suitable barrier seal, provided with nitrogen, as shown in Figure 9.1.12, must be used. The seal systems previously shown will be used for the supply of conditioned gas to the seals as required by the condition of the seal gas (dry or saturated).

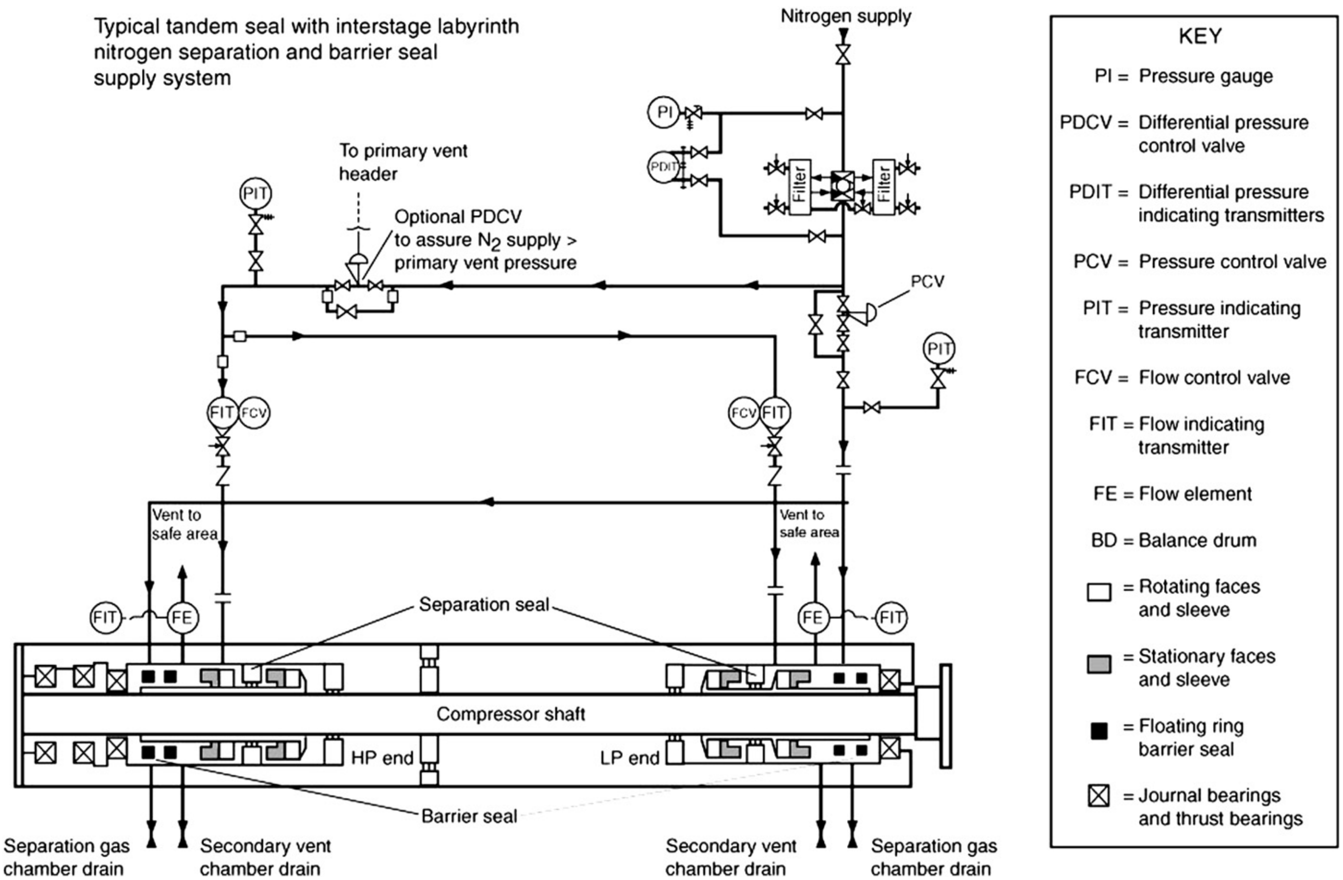


Fig 9.1.16 • Typical tandem seal system with an interstage labyrinth-nitrogen supply

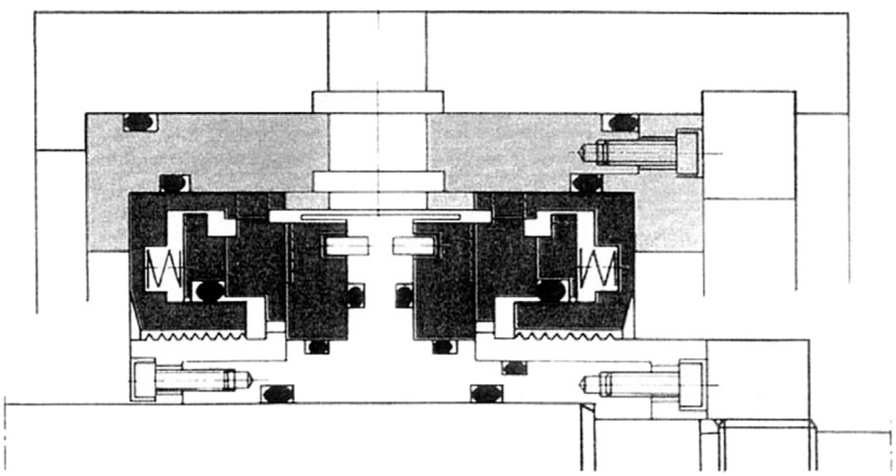


Fig 9.1.17 • Double seal (Courtesy of Flowserve Corp.)

Summary

Since there are significant advantages to the use of dry gas seals, many units are being retrofitted in the field with such systems. In many cases, significant payouts can be realized.

If a unit is to be retrofitted, it is strongly recommended that the design of the gas seal be thoroughly audited to ensure safety and reliability. As mentioned in this section, retrofitting from a liquid to a gas seal system renders the unit a separate system type unit, that is, a separate lube and gas seal system. Naturally, loss of lube oil into the seal system will result in significant costs and could result in seal damage or failure by accumulating debris between the seal rotating and the stationary faces. The design of the separation barriers between the lube and seal face must be thoroughly examined and audited to ensure reliable and safe operation of this system. Many unscheduled field shutdowns and safety problems have resulted from the improper design of the lube system, seal system separation labyrinth. In addition to the above considerations, a critical speed analysis, rotor response and stability analysis (if the operating discharge pressure is above 3,450 kPa [500 psi]) should always be conducted when retrofitting from liquid to dry gas seals.

Best Practice 9.2

Use double seals, with clean gas buffer systems if required, if sufficient N_2 pressure is available and it is compatible with the process. This results in a less complex system of potentially higher reliability than tandem seals.

The reliability of any component is directly related to the complexity of its supporting systems.

Double dry gas seals require the simplest system design (see Figure 9.2.1 in Supporting Material section).

Double seals can be used for all process services that can tolerate N_2 and have a seal reference pressure below 400 kPag (60 psig).

If the process gas can contain debris and/or foul, a source of clean dry buffer gas is required to prevent seal hang up due to contamination of solid particles in the dynamic seal and spring system.

Lessons Learned

Double dry gas seal systems as compared to tandem systems eliminate the following items to reduce complexity and optimize reliability:

- Primary vent hardware and instrumentation
- Concerns with flare header pressures that can cause seal pressure reversals
- The intermediate N_2 gas system

Lack of end user participation in the system specifications has resulted in the use of more complex tandem seal systems where double seal systems could be employed.

Benchmarks

FAI has recommended double seals for new projects since 2000. Double gas seals have been used in low pressure coker and wet gas compressor applications.

B.P. 9.2. Supporting Material

Double seals can help simplify the seal gas control system, minimize the quantity of seal gas, and optimize system reliability. Double seals are normally applied where an inert seal gas (usually N_2), which is compatible with the process, is available at a pressure exceeding the maximum process pressure at the seal interface (to prevent a seal pressure reversal). If N_2 from a regulated system is used, the seal gas control valve can be eliminated (Figure 9.2.1).

If the process gas is sour, a sweet buffer gas must be injected between the process labyrinth and DGS to prevent sour gas contact and potential DGS fouling. Differential pressure control is typically used. Flow control is also an acceptable option, provided the flow is sufficient to maintain a velocity of 15 m/sec (50 ft/sec) through the process labyrinth at twice the maximum design clearance.

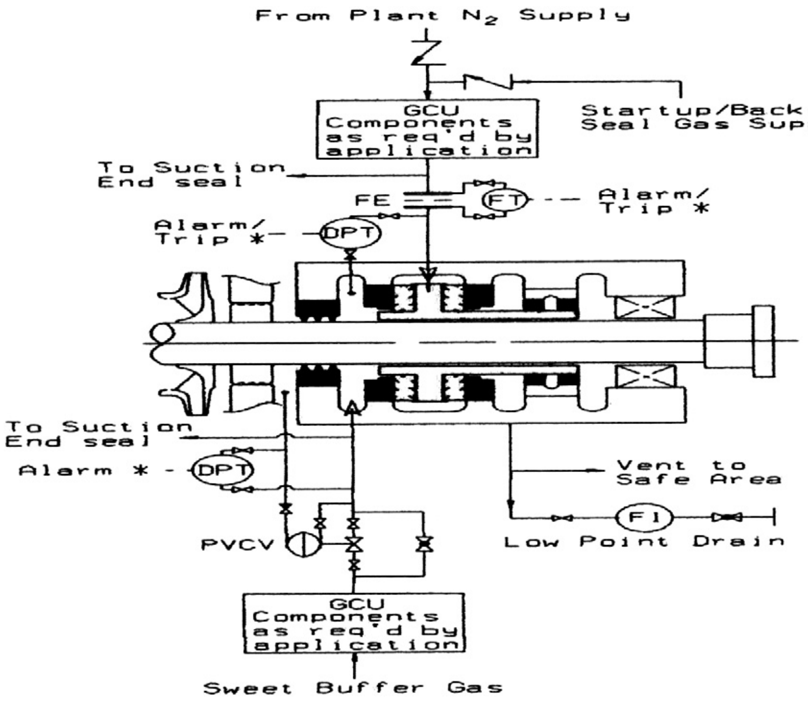


Fig 9.2.1 • Double dry gas seal

Best Practice 9.3

Monitor both primary and secondary tandem seal conditions to ensure plant and personnel safety.

Many tandem dry gas seal systems do not incorporate a means of monitoring the condition of the secondary (atmospheric) seal.

In addition, many applications do not include the injection of intermediate nitrogen between the primary and secondary seals.

Not using intermediate nitrogen to buffer the seal chamber between the primary and secondary seal exposes the seal to a certain process gas release in the event of a secondary (atmospheric) seal failure.

Methods to monitor secondary seal condition include:

- Monitoring high secondary vent pressure
- Monitoring low separation gas to secondary vent differential pressure
- Monitoring low primary vent pressure (where primary vent pressure is kept higher than flare header pressure by a backpressure device)

Lessons Learned

Failure to monitor the secondary tandem seal has resulted in plant safety issues (loss of plant assets and lost time accidents).

In 2007, a number of secondary seal failures forced all compressor vendors to become more aware of the lack of secondary seal monitoring, and to issue directives to their clients to always have a means of secondary seal condition monitoring.

Benchmarks

This best practice has been used since 2005 after investigation of a secondary seal failure after a prolonged period on turning gear (compressor/steam turbine train in a chemical plant).

Since that time, it has been FAI standard practice to monitor secondary seal condition in all tandem seal applications.

B.P. 9.3. Supporting Material

Vent systems between the DGS cartridge and separation seal

Usually referred to as the secondary vent, its purpose is to direct the gas that is present between the DGS (tandem or double arrangement) and the separation seal to a safe location. Most importantly, conditions in this vent can provide information on the health of the outer seal (secondary seal for tandem arrangements and atmospheric seal for double seals) and the separation seal. The majority of installations do not specifically monitor the condition of the secondary or outer seal. Undetected failure of the secondary or the outer seal exposes the plant to a process gas release in the event of primary or inner seal

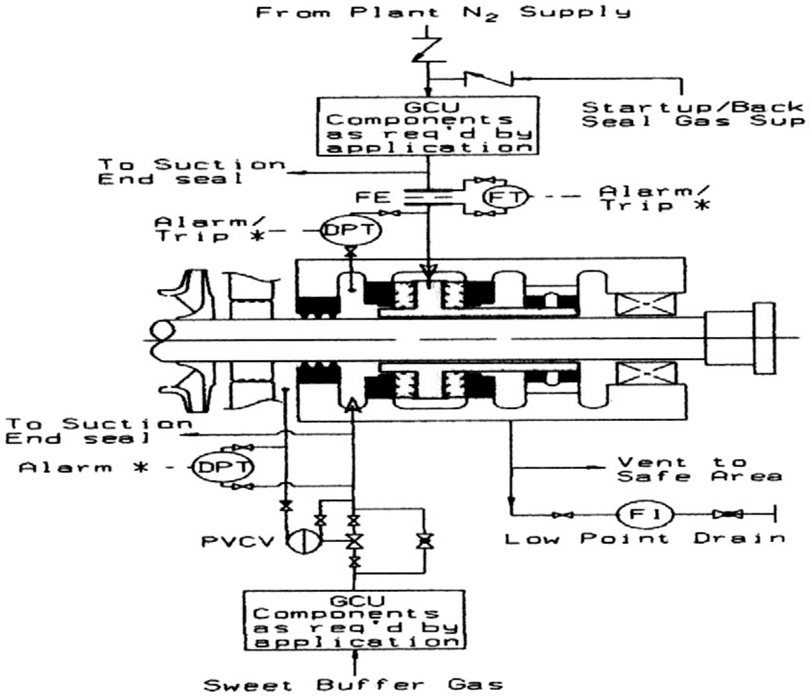


Fig 9.3.1 • Double dry gas seal

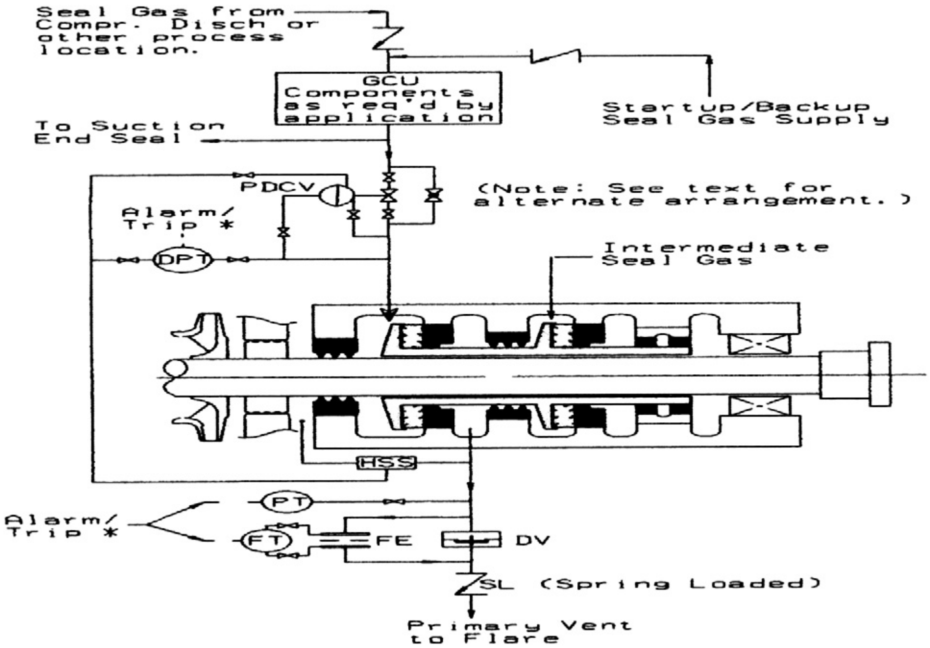


Fig 9.3.2 • Tandem dry gas seal

failure. If process gas can blow through the separation seal and into the bearing housing, a catastrophic equipment failure could occur. Therefore, the best practice is to monitor the condition of the secondary or outer seal by measuring one of the following (Figures 9.3.1, 9.3.2):

- High pressure in the secondary vent – suggested setting: 1–2 kPag (5–10 inches water column).
- Low pressure differential between the separation seal inlet and secondary vent pressure (if the separation gas is controlled at a fixed pressure).
- Low pressure in the primary seal vent – this assumes that a pressure of 30–40 kPag (5 psig) is normally maintained between the primary and secondary seals.

Best Practice 9.4

Use abradable type separation seals for reliability and process advantages.

The main reasons for use of abradable seals are as follows:

- Eliminate the need for a floating carbon seal which has experienced reliability problems (excessive wear, excessive oil migration and low MTBFs)
- Ensure complete oil separation between the bearing housing and seal chamber
- Minimize the usage of N_2 over conventional labyrinth seals
- Avoid a high nitrogen pressure limitation – present in floating carbon ring seals

At the present time (2010) most compressor vendors offer abradable (Fluorisint or equal) seals as an option for separation seals.

Lessons Learned

The nitrogen consumption of conventional labyrinth separation seals is too great to justify their usage, and floating carbon ring seals have not proven to be totally effective in preventing oil migration into the secondary seal vent chamber.

Benchmarks

This best practice has been used since early 2000, when I was involved with a mega ethylene project to optimize the effectiveness of the separation seals and reduce nitrogen consumption.

B.P. 9.4. Supporting Material

Separation systems

Regardless of the type of seal configuration (double or tandem), the function of the separation system is to prevent process gas from entering the bearing housing in the event of a seal failure, and oil from entering the seal cartridge. Entrance of process gas into the bearing housing exposes the plant to catastrophic consequences and extended downtime.

There are several types of separation seals. The choice depends on the availability of the separation gas (usually N_2). The alternatives, arranged in order of highest usage of separation gas, are:

- Labyrinth seals
- Abradable labyrinth seals
- Non-contact carbon seals
- Segmented carbon contact seals

The best practice is to use labyrinth or abradable labyrinth separation seals, if sufficient N_2 is available. This recommendation is based on the reliability of labyrinth-type seals compared to carbon seals, and the fact that the differential pressure across labyrinth seals is not limited, as is the case for most carbon ring seals.

If carbon ring seals are used, the control system must limit the differential pressure to the design maximum. In addition, if carbon contact seals use cryogenic N_2, the best practice is to condition the N_2 to ensure sufficient moisture is present for optimum carbon life (see B.P: 9.5).

Experience shows that, in the case of a catastrophic seal failure, there is a possibility that process gas could enter the bearing housing through the separation seal. For this reason, the best practice is to individually vent each of the bearing housings to a safe location.

The method of separation gas control depends on the type of seal selected. For labyrinth and abradable labyrinth seals, the best practice is to use differential pressure control – seal supply pressure minus secondary vent pressure – to each seal. For carbon ring seals, pressure control could limit the maximum differential pressure across the carbon rings.

The condition of each separation seal can be determined by monitoring and alarming on low differential pressure for labyrinth and abradable labyrinth seals. For carbon ring seals, monitoring and alarming on low pressure is recommended. These parameters should be used as permissive signals to prevent starting the oil system if N_2 gas is not being supplied to the separation seals.

Best Practice 9.5

Ensure that N_2 dew points are above $-30°C$ ($-22°F$) to optimize the life of carbon used in seal faces and separation seals.

This is achieved by:

- Using small air separation units that produce moist N_2 (dew point $> -30°C$ [$-22°F$])
- Using a N_2 bubbling system to condition 'bone dry' N_2 so that the dew point $> -30°C$ ($-22°F$)

Currently, air separation units produce nitrogen with dew points below $-50°C$ ($-58°F$).

The life of carbon seals (radial and face seal) is significantly reduced in dry gas applications where the N_2 dew points are below $-30°C$ ($-22°F$).

Lessons Learned

The use of 'bone dry' N_2 (dew points below $-30°C$) for intermediate and separation sealing duties has resulted in low seal MTBFs (below 12 months).

In some cases, floating carbon seal wear was observed during the factory acceptance tests (FATs).

Benchmarks

This best practice was first used in 2008. Since that time, specifications that require the dew point of supplied N_2 to be above $-30°C$ ($-22°F$) have been produced. It should be noted that small, dedicated N_2 generators can produce N_2 above $-30°C$ ($-22°F$).

B.P. 9.5. Supporting Material

Seal gas conditioning

Cryogenic nitrogen (N_2 that has been liquefied) can damage the carbon stationary faces during slow-speed operation — turning gear ratcheting or slow roll — when the faces are in contact. Cryogenic N_2 is typically very dry, with a dew point as low as $-90°C$ ($-130°F$). But the self-lubricating quality of carbon is based on the ability of its crystalline structure to adsorb and hold certain gases, including water vapor, which significantly reduce rubbing friction.

In the absence of water vapor, carbon has poor lubricating properties, and can wear rapidly. Therefore, dew point conditioning is required whenever carbon stationary elements are used in either face or circumferential seals, when rubbing contact is anticipated for extended periods. For large steam or gas turbine driven compressors that require slow roll for extended periods below the DGS lift-off speed, the best practice is to condition the N_2 upstream of the coalescing filter system, raising its dew point to $-30°C$ ($-22°F$), or higher.

Methods to increase N_2 dew point include mixing saturated nitrogen — from a bubbler chamber — with cryogenic nitrogen in an appropriate ratio, or mixing moist air with cryogenic nitrogen, keeping the oxygen content below 5%. A dew-point monitor and low-dew-point alarm are required for safe operation.

Best Practice 9.6

Always strongly recommend and justify the use of an external source of clean, dry seal gas, if available, to eliminate the issues that will reduce seal system reliability.

The main issues are as follows:

- Contamination of the seal by saturated gas
- Contamination of the seal by process gas debris
- The necessity for a start-up gas
- The necessity of a gas booster compressor system

During the early phases of the project (pre-FEED), discuss and investigate the viability and costs to use an external source of clean, dry seal gas.

If a continuous source of seal gas is available, the majority of issues that affect the safety and reliability of the dry gas seal system will be eliminated!

Lessons Learned

Seal gas conditioning does not ensure trouble free operation in processes that cannot guarantee clean and dry seal gas. A clean, dry, external source of seal gas can be justified on the basis of lost revenue experienced in your plant, or through industry case histories that have not used a clean and dry external source of seal gas.

Benchmarks

This approach has been used in all projects, and for seal gas system modification recommendations since the late 1990s. It has produced dry gas seal systems of the highest degree of safety and reliability (seal MTBFs > 70 months and compressor reliabilities > 99.7%).

B.P. 9.6. Supporting Material

See B.P: 9.1 for supporting material.

Best Practice 9.7

Design the seal gas supply system so that seal gas will always be maintained at a higher than reference gas or flare pressure, to prevent seal face pressure reversals.

Compressor operating pressures and flare header pressures can and will vary from the values specified on the seal data sheets.

Using one of the following methods will ensure that a seal face pressure reversal, which can fail the seal, will not occur:

- A dedicated differential pressure control valve for each seal with a signal select device that will reference the higher value of seal gas reference pressure or primary vent pressure upstream of the primary vent orifice (see Figure 9.7.1 in supporting information below).
- One differential pressure control valve supplying seal gas to each seal in the compressor that senses pressure in the primary vent header downstream of the orifices and set a sufficient high differential pressure (3–4 barg) to ensure that supplied seal gas pressure will always be greater than the maximum flare header pressure. (This alternative is used for large flow compressors where the excessive seal gas flow entering the compressor is still a small percentage of total compressor flow.)

Lessons Learned

The majority of dry gas seal failures occur in low suction pressure services, where the flare header pressure can exceed the seal gas supply pressure. This situation will cause a seal face pressure reversal, which will force the seal faces together and fail the seal.

In 2006, FAI was involved in the root cause analysis of a large number of dry gas seal failures in low pressure services. In the majority of cases, failure of the seal system to respond to a larger than expected primary vent flare header pressure was determined to be the root cause of the seal failures.

Benchmarks

This best practice has been used since 2006 for new projects, and during dry gas seal field audits to ensure that a seal face pressure reversal does not occur. At the present time (2010) all projects and field modifications using this best practice have not encountered a seal failure (MTBFs greater than 48 months and counting!).

B.P. 9.7. Supporting Material

Tandem seals are the most common DGS application, and are required when an inert gas is either not available at sufficient pressure, or not compatible with the process. The traditional control arrangement is to maintain a differential of 35–70 kPa (5–10 psid) over the balance or equalization chamber pressure, using one pressure differential control valve (PDCV). This arrangement is adequate when the sealing pressure is greater than the maximum pressure that can occur in the primary seal vent cavity.

This control scheme can, however, expose the primary seal to possible pressure reversals in low-pressure services, where the maximum cavity pressure of the primary vent can exceed the sealing pressure. While the flare header pressure is normally low (7–21 kPag or 1–3 psig), the maximum design flare pressure that can exist during a major upset, or during an emergency shut down (ESD) can range from 140–340 kPag (20 psig–50 psig).

Therefore, the maximum pressure in the primary seal vent cavity can be equal to the maximum flare pressure plus losses through check valves, orifices, and piping in the vent line.

For services with low or sub-atmospheric suction pressures, even ‘normal’ conditions in the primary vent can cause a reverse differential on the primary seal, unless the system design precludes this possibility.

A significant number of DGS failures in recent years have occurred in low suction pressure refrigeration, and other services. Figure 9.7.1 shows a system that has been used successfully in applications with low suction pressure. A PDCV at each seal controls the seal gas pressure to the inlet cavity at a nominal value of 35–70 kPa (5–10 psi) above the higher of the following values; reference gas pressure and primary seal vent cavity pressure measured upstream of the vent orifice. This is accomplished through the use of a high signal selector device at each seal, which ensures that the primary seal will always have a positive differential of at least 35– 70 kPa (5–10 psid), even if the primary vent pressure increases significantly during an upset or ESD.

Another approach that has been used successfully for large machines, where the seal gas flow represents a relatively small recirculation loss, is to use one PDCV for both seals. In this scheme, the sensing point for the primary seal vent pressure is moved to a location in the primary vent that is common to both seals. The differential pressure controlled by the PDCV is then set high enough (typically 200–350 kPa or 30–50 psid) to ensure that it will always be greater than the maximum pressure drop across the orifice, piping, and any other component between the sensing point and the primary seal vent cavity. This pressure also provides for the maximum allowable primary seal gas leakage flow. Regardless of whether one or two PDCVs are used, the best practice is to ensure that the control valve always operates within the acceptable valve coefficient – known as ‘CV’ – range. For large units where seal gas flows can vary widely, an orifice can be installed in parallel with the PDCV for normal flow conditions. In this arrangement, the PDCV will remain closed unless flow conditions dictate otherwise, and the smaller valve will always operate in the acceptable CV range (10–90%). The key in low-pressure service is to monitor the primary seal differential, as well as the process labyrinth differential, and control the seal gas pressure based on the higher of the two values to ensure that a primary seal pressure reversal is not possible under any anticipated circumstances.

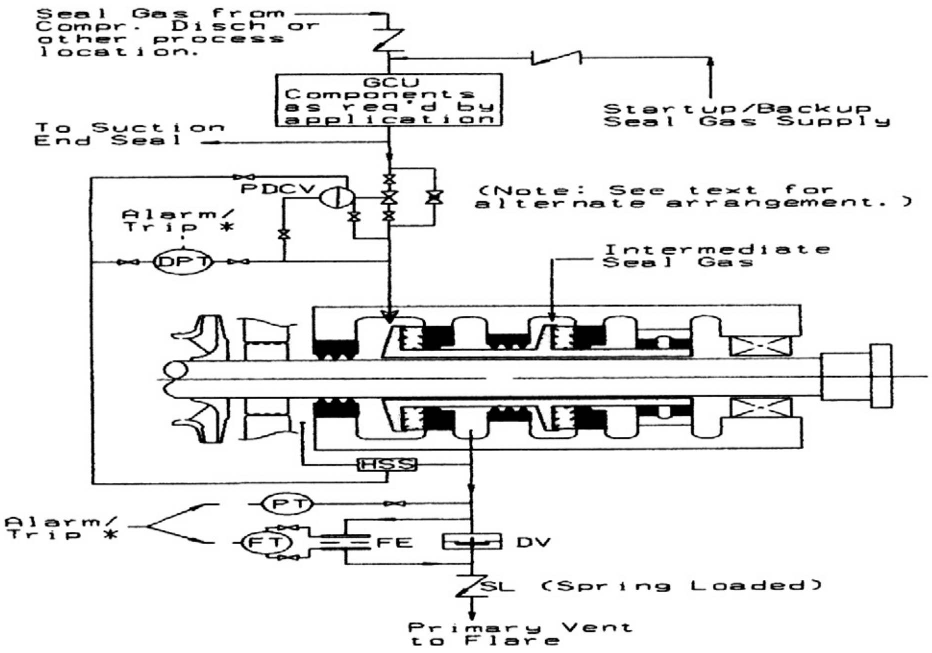


Fig 9.7.1 • Tandem dry gas seal

Best Practice 9.8

Ensure that the low point drain area in a secondary vent is monitored and drained for lube oil carryover to prevent a secondary (outer) seal failure due to oil contamination.

All dry gas seals are provided with a separation seal system to prevent oil from entering into the seal chamber. However, the following issues can arise, which allow lube oil carryover into the secondary (outer) seal port:

- Buffer gas (nitrogen or air) supply upsets
- Control valve malfunction
- Carbon ring floating seal hang-up
- Carbon ring floating seal wear
- Bearing housing oil drain system blockage

Lessons Learned

The majority of dry gas seal separation systems do not positively prevent carryover of oil into the seal chamber. Failure to monitor and drain the secondary (outer) seal chamber can result in sudden secondary seal failure and expose the plant to a gas release.

In 2008, a client had a catastrophic secondary seal failure resulting in a plant gas release. During seal changeout it was discovered that the secondary vent had been totally filled with lube oil up to the top (10 meters + high).

Benchmarks

This best practice has been used since the late 1990s when an outer seal failure was caused by excessive lube oil carryover without being monitored. Since that time, we have always recommended that secondary seal vent chambers be checked periodically for oil carryover, and that a permanent means of monitoring be installed (automatic seal oil drainers – the type used as sour oil drain traps in all oil systems). This best practice has produced dry gas seal MTBFs in excess of 100 months.

B.P. 9.8. Supporting Material

Vent systems between the DGS cartridge and separation seal

Usually referred to as the secondary vent, its purpose is to direct the gas that is present between the DGS (tandem or double arrangement) and the separation seal to a safe location. Most importantly, conditions in this vent can provide information on the health of the outer seal (secondary seal for tandem arrangements and atmospheric seal for double seals) and the separation seal. The majority of installations do not specifically monitor the condition of the secondary or outer seal. Undetected failure of this seal exposes the plant to a process gas release in the event of primary or inner seal failure. If process gas can blow through the separation seal and into the bearing housing, a catastrophic equipment failure could occur. Therefore, the best practice is to

monitor the condition of the secondary or outer seal by measuring one of the following:

- High pressure in the secondary vent – suggested setting: 1–2 kpag (5–10 inches water column)
- Low pressure differential between the separation seal inlet and secondary vent pressure (if the separation gas is controlled at a fixed pressure)
- Low pressure in the primary seal vent – this assumes that a pressure of 30–40 kpag (5 psig) is normally maintained between the primary and secondary seals.

The decision to alarm or trip will depend on the application and the potential daily revenue loss of the plant. Since this vent can also contain oil or oil mist in the event of a separation seal system malfunction, the best practice is to monitor the effectiveness of the separation seal by locating the vent in the seal chamber at the low point (6 o'clock position), and installing a device to indicate oil contamination (level glass as a minimum), with a drain valve to a safe location in the vent line.

Separation systems

Regardless of the type of seal configuration (double or tandem), the function of the separation system is to prevent process gas from entering the bearing housing in the event of a seal failure, and oil from entering the seal cartridge. Entrance of process gas into the bearing housing exposes the plant to catastrophic consequences and extended downtime.

There are several types of separation seals. The choice depends on the availability of the separation gas (usually N_2). The alternatives, arranged in order of highest usage of separation gas, are:

- Labyrinth seals
- Abradable labyrinth seals
- Non-contact carbon seals
- Segmented carbon contact seals

The best practice is to use labyrinth or abradable labyrinth separation seals, if sufficient N_2 is available. This recommendation is based on the reliability of labyrinth-type seals compared to carbon seals, and the fact that the differential pressure across labyrinth seals is not limited, as is the case for most carbon ring seals.

If carbon ring seals are used, the control system must limit the differential pressure to the design maximum. In addition, if carbon contact seals use cryogenic N_2, the best practice is to condition the N_2.

Experience shows that in the case of a catastrophic seal failure, there is a possibility that process gas could enter the bearing housing through the separation seal. For this reason, the best practice is to individually vent each of the bearing housings to a safe location.

The method of separation gas control depends on the type of seal selected. For labyrinth and abradable labyrinth seals, the best practice is to use differential pressure control – seal supply pressure minus secondary vent pressure – to each seal. For carbon ring seals, pressure control could limit the maximum differential pressure across the carbon rings.

The condition of each separation seal can be determined by monitoring and alarming on low differential pressure for labyrinth and abradable labyrinth seals. For carbon ring seals, monitoring and alarming on low pressure is recommended. These parameters should be used as permissive signals to prevent starting the oil system if N_2 gas is not being supplied to the separation seals.

Best Practice 9.9

Require a seal gas conditioning unit (cooler, separator and heater) when there is any possibility that seal gas can be saturated and an external clean dry seal gas source is not available.

If the search for an installed source of clean dry external seal gas has not been successful, there is proven industry experience (since 2000) with the use of a seal gas condition unit that has the following features:

- A seal gas cooler that reduces the seal gas to a minimum of 15°C below saturation temperature at the lowest pressure experienced in the primary vent
- A properly sized separation vessel with automatic level control and a demister
- A seal gas heater that raises the seal gas to a minimum of 15°C above saturation temperature
- A dual coalescer filter with a continuous drain to ensure only clean, dry gas enters the seal chamber
- Note: If the seal gas contains C6+ components, each gas component must be individually included in the gas analysis calculations to ensure the correct saturation temperature at operating pressure conditions.

Lessons Learned

Failure to include a seal gas conditioning unit when the seal gas can be saturated has resulted in low seal reliability (less than 12 months MTBF in some cases).

The failure to design a gas conditioning unit into the seal system, especially in oil and gas applications, has eventually required modification to a GCU (gas conditioning unit) that was justified by multiple seal failures and significant product revenue losses.

Benchmarks

This best practice has been used since 2000 for field modifications, to result in seal MTBFs in excess of 48 months where the previous system (seal gas sent directly to the seal gas filters) could not yield seal MTBFs above 12 months.

B.P. 9.9. Supporting Material

Saturated seal gas, either in the system or entering downstream of the GCU, exposes the DGS to liquid condensation and carryover into the seal chamber and between the DGS faces. The risk of seal damage is high when liquid enters the area between the faces. The best practice is to ensure that the gas is superheated to approximately 15°C (27°F) above the gas condensing temperature at the lowest operating pressure in the primary vent. The addition of a heater to the GCU may be sufficient for this purpose. However, if the required temperature rise could cause polymerization, a cooler, separation vessel and re-heater may be required. If the seal gas contains C6+ components, they must be identified and individually considered in determining saturation conditions.

Best Practice 9.10

Require a primary vent backpressure device to sufficiently pressurize the seal chamber and allow monitoring of secondary seal condition by a low primary pressure alarm transmitter.

The MTBF of the secondary (outside) seal in a tandem seal arrangement can be increased if the differential pressure across the seal faces is greater than 50 kPa (7.25 psi).

Installing a backpressure device (spring loaded check valve or backpressure control valve) will increase the pressure in the seal chamber to above 50 kPa (7.25 psi) and allow the condition of the secondary (outside) seal to be monitored easily. Reduced primary vent pressure, which is the seal chamber pressure, immediately indicates secondary seal increased leakage.

Lessons Learned

Tandem seals that do not use a means of pressurizing the seal chamber above 50 kPa (7.25 psi) have lower MTBFs (less than 70+ months) than tandem seals using a means to pressurize the seal chamber.

Seal vendors concur that increasing the seal chamber pressure in tandem seal applications will prolong the life of the secondary seal, since a greater positive pressure will reduce seal face wear and minimize the possibility of seal face groove contamination.

Benchmarks

This best practice has been recommended for all new projects and field seal system modifications since the early 2000s. This best practice has resulted in increased seal MTBFs.

B.P. 9.10. Supporting Material

A spring-loaded check valve is recommended in the piping at the flare header connection.

This prevents the back-flow of flare gas into the primary vent chamber, and maintains a positive pressure differential across the secondary seal. The check valve is normally designed to exert a minimum back pressure of 35 kPag (5 psig) in the primary vent cavity, ensuring a minimum 35 kPad (5 psid) positive differential over the secondary seal. However, the required secondary seal pressure differential should be determined by the seal vendor, based on anticipated turning gear operation and seal lift-off speed.

Recently, (since 2008) some vendors have been installing a back pressure device in the primary vent which is set at 140 kPa (20 psi) or greater. This approach increases secondary seal face differential pressure and enables monitoring of secondary seal condition (low primary vent pressure indication if secondary seal experiences excessive leakage).

Best Practice 9.11

Require a primary vent orifice bypass device, for tandem seals, whenever the primary vent pressure, during a seal failure can rise above the lowest value of N_2 supply pressure.

If the seal reference pressure is below 400 kPa (60 psig) (Typical minimum plant N_2 header pressure), confirm that the minimum available N_2 pressure at the intermediate seal labyrinth will always be greater than the maximum seal reference pressure to ensure that N_2 gas will fill the seal chamber during a primary seal failure.

If the seal reference pressure can be greater than the lowest value of intermediate N_2 pressure, a weight loaded full stem disc valve set to open at a pressure 20 kPa (3 psi) below the lowest value of N_2 at the seal is recommended. This will ensure that N_2 gas will always be present in the seal chamber and will exit the secondary (outside) seal during primary seal failure and/or upset conditions.

Lessons Learned

Not having a primary seal vent orifice bypass device in high (greater than 400 kPa [60 psig]) seal reference gas applications has led to process gas releases in the plant during a primary seal failure.

Benchmarks

This best practice has been used during new projects since 2000 to result in optimum tandem dry gas seal system safety and reliability (MTBFs greater than 48 months and counting!).

B.P. 9.11. Supporting Material

If a complete primary seal failure can cause the pressure in the primary vent cavity to exceed the pressure that can be delivered by the N_2 system, the best practice is to relieve the vent pressure below the N_2 system pressure. A weight-loaded, stem-guided, full-flow disc valve is recommended. A rupture disc can randomly fail and will require a unit shutdown for replacement.

Best Practice 9.12

Use an 'AND trip' (primary vent high pressure or flow along with indication of secondary seal excessive leakage) to prevent spurious trips while ensuring plant safety whenever plant daily revenue is greater than $1 MM/day.

Historically, dry gas seal vendors have set primary vent trip levels at three times the normal primary seal flow.

However, if the seal gas system is designed with the present (2010) 'best practice' available features, increases in primary vent flow can be safely accommodated.

Best practice features which will ensure system safety during increased primary vent flow are:

- The intermediate N_2 system that is automatically flow controlled.
- The use of a primary vent orifice bypass device (see B.P: 9.11).

If the exposure to daily revenue losses is large (greater than $ 1 MM/day) and the above best practice features are used in the system, an 'AND trip' philosophy is recommended (trip the unit only when excessive primary vent flow and detected secondary seal excessive leakage are present).

A HAZOP review is required for implementation of this best practice.

Lessons Learned

Large (mega) plants have learned that the previous philosophy of primary vent only trips exposes the plant to large revenue losses and the present dry gas seal system best practices enable a safe (HAZOP accepted) alternate trip arrangement that will optimize plant revenue.

Benchmarks

The 'AND trip' approach has been recommended since 2008 and is being used in many new large (over $1 MM/day) revenue plants to ensure plant safety and optimize plant revenue.

B.P. 9.12. Supporting Material

Monitoring and protection

The majority of current DGS installations trip on primary seal failure by measuring primary vent pressure or flow. In the case of a high value product, where availability of the plant is critical, operations may choose to continue running for short periods after the primary seal fails, while preparing for shutdown. In this case, the recommendation is to trip on indicated primary 'AND' secondary seal failure (atmospheric side for double seals). Trip

options are based on the type of seal configuration and secondary seal. They are:

- Tandem seals — trip on 'high-high' primary seal flow or pressure 'AND' high secondary vent pressure
- Tandem seals with carbon contact separation seal — trip on 'high-high' primary seal flow or pressure 'AND' low N_2 separation gas supply minus secondary vent pressure
- Double seals — trip on 'high-high' seal gas supply flow 'AND' vent 'high-high' pressure

For any of the above options, instruments associated with the shutdown circuit should be triple modular redundant. The final decision regarding unit trips is of critical importance to plant safety and reliability, and will require a HAZOP review for each application.

Best Practice 9.13

Do not connect bearing vents of the machinery to a common header in dry gas seal applications. This will ensure that a dry gas seal failure cannot transfer process gas to the bearings of the other machinery in the train.

If a catastrophic tandem seal failure occurs (simultaneous failure of primary and secondary seals) process gas may enter the bearing housing of the compressor.

If the individual bearing housings in the compressor train are connected to a common header, process gas will be transferred to all the other bearing housings in the train.

If a turbine driver or non-intrinsically safe motor is contained in the train, exposure to an explosion or fire is present in hydrocarbon services.

Lessons Learned

A recent (2007) plant fire was started when a total dry gas seal failure in a steam turbine driven compressor train transferred process gas through the common bearing housing vent system to the inlet steam end bearing of a VHP (very high pressure – greater than 10,000 kPa) steam turbine.

Benchmarks

This best practice has been recommended since the reliability issue identified above for all new and existing installations that have been audited by our organization since 2007.

B.P. 9.13. Supporting Material

Experience shows that in the case of a catastrophic seal failure, there is a possibility that process gas could enter the bearing housing through the separation seal. For this reason, the best practice is to individually vent each of the bearing housings to a safe location.

Best Practice 9.14

Control separation nitrogen pressure using a differential pressure transmitter sensing secondary vent pressure to ensure a nitrogen atmosphere between the secondary seal and separation seal to positively prevent a hydrocarbon release to the plant environment.

Frequently called the 'disaster bushing', the separation seal is the last item in the dry gas seal chamber that can prevent a gas release to the plant environment.

If the separation gas system (usually N_2) is designed with a differential pressure controller to regulate supplied separation gas above the secondary seal vent pressure 20 to 40 kPad (3 to 6 psi), positive sealing will be ensured.

Lessons Learned

Failure to use a differential pressure controller for the separation seal system has resulted in process gas releases to the plant environment during tandem seal reliability issues.

Benchmarks

This best practice has been used since 1990 in projects to optimize plant safety and reliability. (no known process gas releases for the subject projects and seal MTBFs greater than 70 months).

B.P. 9.14. Supporting Material

Separation systems

Regardless of the type of seal configuration (double or tandem), the function of the separation system is to prevent process gas from entering the bearing housing in the event of a seal failure, and oil from entering the seal cartridge. Entrance of process gas into the bearing housing exposes the plant to catastrophic consequences and extended downtime.

There are several types of separation seals. The choice depends on the availability of the separation gas (usually N_2). The alternatives, arranged in order of highest usage of separation gas, are:

- Labyrinth seals
- Abradable labyrinth seals
- Non-contact carbon seals
- Segmented carbon contact seals

The best practice is to use labyrinth or abradable labyrinth separation seals, if sufficient N_2 is available. This recommendation is based on the reliability of labyrinth-type seals compared to carbon seals, and the fact that the differential pressure across labyrinth seals is not limited, as is the case for most carbon ring seals. If carbon ring seals are used, the control system must limit the differential pressure to the design maximum. In addition, if carbon contact seals use cryogenic N_2, the best practice is to condition the N_2.

Experience shows that in the case of a catastrophic seal failure, there is a possibility that process gas could enter the bearing housing through the separation seal. For this reason, the best practice is to individually vent each of the bearing housings to a safe location.

The method of separation gas control depends on the type of seal selected. For labyrinth and abradable labyrinth seals, the best practice is to use differential pressure control – seal supply pressure minus secondary vent pressure – to each seal. For carbon ring seals, pressure control could limit the maximum differential pressure across the carbon rings.

The condition of each separation seal can be determined by monitoring and alarming on low differential pressure for labyrinth and abradable labyrinth seals. For carbon ring seals, monitoring and alarming on low pressure is recommended. These parameters should be used as permissive signals to prevent starting the oil system if N_2 gas is not being supplied to the separation seals.

Best Practice 9.15

Color code the following dry gas seal sub-systems, using an OME team (operations, engineering and maintenance) to ensure complete understanding of the system functions and optimize safety and reliability:

- Seal gas system
- Primary vent system
- Intermediate nitrogen system
- Secondary vent system
- Separation seal system

Dry gas systems are critical to compressor safety and plant revenue (since a dry gas seal failure can impact plant safety and daily revenue).

Enabling the plant personnel DGS system function understanding (operations, maintenance and engineering) significantly contributes to increased plant safety and reliability.

Using a 'team' approach to color code each DGS sub-system (paint – new units or color tape – existing units) will lead to understanding and ownership of these important systems.

Lessons Learned

Lack of operator, maintenance and engineering awareness of DGS system function and operation has led to reduced dry gas seal system safety and low seal MTBFs (less than 12 months). Using a site OME team to color code these systems will correct these issues.

Benchmarks

This best practice has been recommended since 2000, initially on a project that used the first dry gas seal system in an existing chemical plant that previously had used oil seal systems. The color coding of the dry gas seal system by operations and engineering greatly increased awareness and understanding of this system.

10

The Post-Shipment Phase: Installation, Pre-Commissioning, Commissioning and Start-up Best Practices

Introduction

The safety and reliability of properly specified, designed and tested critical machinery over the 30+ year plant life is directly related to the execution of the post-shipment phase activities. Unfortunately, it is common practice that the required procedures, materials and spare part storage facilities are not sufficiently prepared in advance, which results in reducing the safety and reliability of the plant for its entire life.

This chapter therefore presents the post shipment phase best practices that will optimize machinery safety and reliability for each day of plant life.

Best Practice 10.1

Review and finalize machinery installation details well in advance of site construction activities to ensure optimum safety and reliability for the plant life cycle.

The relevant areas to concentrate on are as follows:

- Prepare, review and approve all site installation procedures (See ‘Supporting Material’ below for specific procedures).
- Confirm spare part orders, delivery dates, storage facilities and preservation compounds.
- Review and approve all machinery instruction manuals prior to shipment, and require manuals be shipped at the same time as the machinery.
- Review the special tool list during the engineering phase and require their shipment at the same time as the machinery.
- Establish a ‘material receipt’ procedure to confirm that all required material has been received on or before the promised date.

Confirm that there is full coordination between the engineering/procurement (EP) and construction (C) groups to ensure the above best practice guidelines are executed well before the construction phase.

Lessons Learned

Critical machinery safety and reliability is not ensured only by proper specification, design and testing but depends on effective planning, management and implementation during the project construction phase.

The following examples show how the reliability of properly specified, designed and tested machinery was challenged by inadequate construction phase execution:

- Preservation procedures not prepared in advance, general in nature and not implemented
- Instruction manuals not on site when required and when received, not complete
- Spare parts not on site when required resulting in start-up delays
- Special tools missing and not available resulting in start-up delays

Benchmarks

This best practice has been used since the mid-1980s for large petrochemical projects, and requires that all machinery construction details be addressed during the engineering phase of the project. This best practice has resulted in smooth, timely start-ups for large machinery, and critical machinery reliabilities of maximum safety and reliability (greater than 99.7%).

B.P. 10.1. Supporting Material

Regardless of the quality of design and manufacture; regardless of a successful test and efficient shipment; installation will determine the amount of maintenance required and the resulting revenue of the process unit. Figure 10.1.1 shows the general site considerations that are required for a successful field installation. Each one of these items will now be covered.

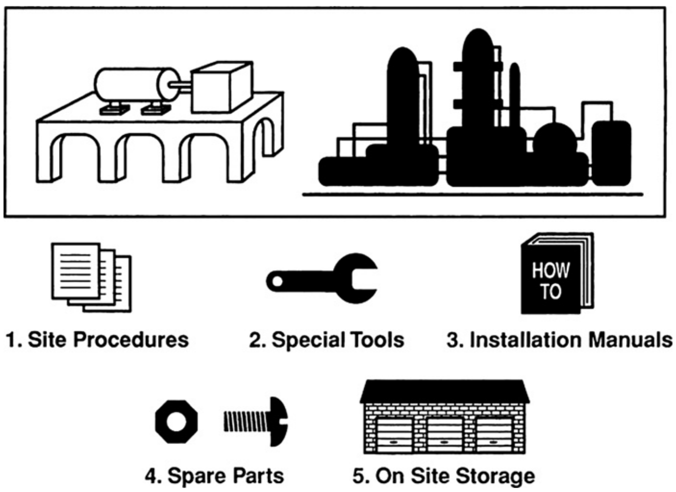


Fig 10.1.1 • General site considerations

Site procedures

The importance of site installation procedures cannot be overemphasized. Figure 10.1.2 shows the most commonly required site installation procedures.

It must be remembered that the objectives of the construction contractor and of the end user are identical in terms of profit. However, they are dissimilar in the means used to achieve these common objectives. The contractor’s objective is to construct a safe and reliable process unit within the budget and on time. This objective is opposed to the end user’s objective, which is to operate the process unit for thirty years or more at maximum profit and thus requires maximum reliability of the installed equipment. The only leverage that the end user has in meeting his objectives is to require practical, proven site installation procedures that will result in the most reliable, safe and cost effective installation of his equipment. Frequently the contactor will rely on the equipment vendor to provide most of the site procedures. It is strongly recommended that the end user, early in the project, require approved procedures for every major site installation milestone. These procedures include, but are not limited to:

- Equipment preservation
- Equipment installation
- Grouting
- Alignment
- Flushing
- Functional checks
- Initial run in of equipment

It must be mentioned that preservation procedures are often ignored early in a construction project and become written and

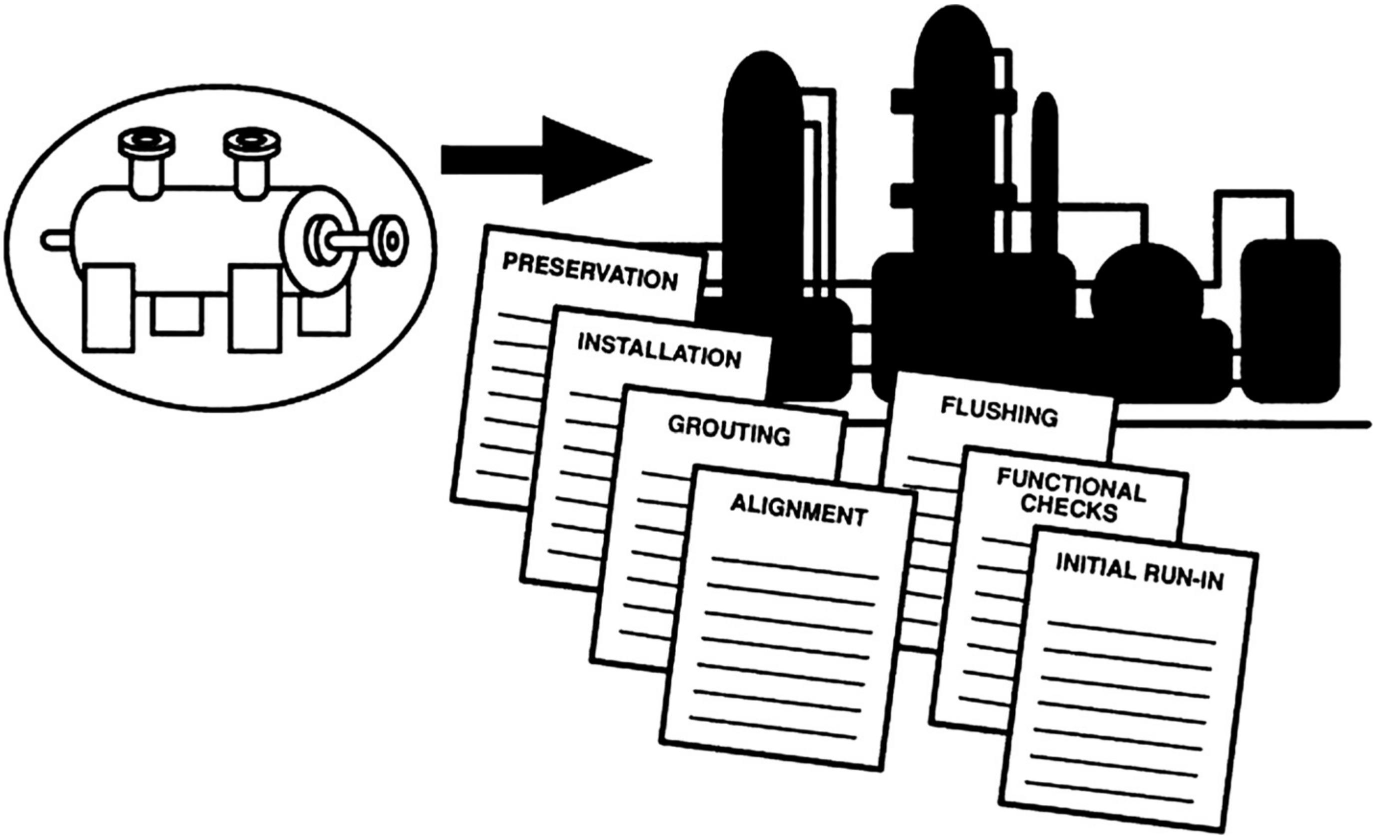


Fig 10.1.2 • Site installation procedures

implemented too late to effectively prevent equipment deterioration due to corrosion. Again, require procedures be written and approved well in advance in the start of construction.

A specific 'initial run in procedure' for each major piece of rotating equipment should be reviewed by the end user well in advance. It is recommended that the end user review these procedures during the shop test with vendor field service engineers to ensure that the run in procedure is in accordance with vendor and plant best practices. An example of not performing a pre-run in review and the consequences of not doing this concerned a large, high-pressure, condensing, steam turbine cold start-up time vs. speed curve. Since the turbine was designed for an automated sequenced start, the cold start-up curve was programmed into the PLC without detailed review by the end user. The curve was simulated and did not take the specific parameters of the application into account. The result was a severe rub that damaged rotor and stationary internals, and resulted in over a 40 day plant start-up delay. The daily profit of this plant was approximately 0.25 $MM.

Construction special tools

Most of the equipment that is installed will be custom designed. Therefore, special tools will be shipped with the equipment that can only be used for that particular item. Consequently, these tools must be listed and stored in a proper location that will allow maintenance personnel to easily locate them when required. Many of the tools, such as hydraulic jacks, special mounting devices, etc. also will require preservation during storage to prevent corrosion. Many times this requirement is overlooked. Be aware!

An example of not properly storing special tools and spare parts is a high humidity, tropical island installation where the spare rotor and coupling hydraulic mounting tools were to be used for a turnaround. They were not inspected prior to the turnaround and the result was that the spare rotor could not be installed due to excessive corrosion. The coupling hydraulic mounting adapter had to be replaced with a new adapter due to excessive corrosion. It should be noted that this equipment was stored in a sealed container with a nitrogen purge but unfortunately the seal was faulty and the nitrogen purge pressure was not monitored.

Installation manuals

Like site procedures, installation manuals frequently arrive after they are first required. The installation manual will contain valuable information pertaining to receipt of equipment, preservation, interim storage and of course, installation. It is recommended that the end user again require that manuals be approved and received well in advance of the start of any installation activity.

In an effort to ensure that the instruction manual contains information that is accurate and specific to the job, we have written into the job specification that the instruction book shall be completely reviewed at the time of the shop test and

shipped with the unit. Almost every such review has uncovered incorrect information and/or general information that is not specific to the particular project that would have resulted in confusion during disassembly/assembly of the equipment, which in turn may have lead to possible equipment reliability problems.

Spare parts

The end user must require the contractor to be responsible for the proper storage of spare parts on site and most importantly, the receipt of all required start-up spares, operating spares and capital spare parts well in advance of the start of construction activity. Often, spare parts are required prior to the initial operation of equipment, since components are broken during shipment.

It has been our experience that many of the spare rotors for major, unspared equipment are not properly inspected and maintained upon receipt from the vendor. A specific rotor container inspection procedure should be written by the contractor, and approved by the vendor and end user, to ensure that all rotor preservation will be maintained from initial receipt date on site. There have been many cases where the rotor containers have never been inspected prior to the intended use date and could not be used and had to be returned to the vendor for rework.

On site storage

Prior to the arrival of any equipment on site the contractor should review the manufacturer's requirements and provide extended on site storage facilities that meet or exceed those requirements. In addition, the contractor shall order and have available the required preservation compounds.

When selecting the preservative compounds, special care should be given to the selection of compounds for the specific site environment and should be based on local experience. In humid, seacoast and offshore environments, special care should be given to ensure the preservative compounds can resist high moisture/salt environments. In addition, components should be checked frequently to confirm that the compounds are providing the required protection.

Best Practice 10.2

Ensure that the all preservation materials are on site and that the agreed site machinery preservation program is fully implemented.

An important initial site activity is to review the previously approved machinery preservation procedure (see B.P. 10.1) with the construction contractor to confirm:

- All preservation materials are on site
- Preservation materials are correct for the site environment
- Schedule of preservation activities and checks for each machinery item
- Method of record keeping to ensure the activity is performed on time
- Personnel assigned to the preservation activity are sufficient in number
- Day and time for monthly preservation status meetings

Involvement with plant construction activities has revealed the following issues with plant preservation programs:

- Insufficient preservation material ordered thus extending the actual intervals between preservation activity
- False reporting of preservation checks (marking off checks without performing them)
- Insufficient manpower for the preservation program leading to extended or missed preservation activities

Lessons Learned

Delayed machinery start-ups are often caused by insufficient preservation programs

Benchmarks

This best practice has been used since the mid-1980s in all projects to ensure sufficient machinery preservation during construction and on time plant start-ups. Implementation of these best practices has resulted in high bearing and seal MTBFs (in excess of 100 months).

B.P. 10.2. Supporting Material

See B.P: 10.1 for supporting material.

Best Practice 10.3

Ensure that site visits by vendor service representatives are effective, by taking appropriate action at the time of the vendor assistance request.

Steps to ensure effective site visits are as follows:

- Send detailed problem statements for each machinery item to be covered
- Send detailed workscope for each machinery item to enable vendor representatives to be totally prepared
- Require vendors to submit two nominees (main and alternate) along with contact reference phone numbers for last two field visits made by the nominees
- State in vendor visit document that weekly visit reports are required while on site and a final visit report is to be submitted prior to leaving site
- Include any specific training requirements for the representative to present during the visit

Following all the guidelines noted above will optimize visit effectiveness and establish a good working relationship with the vendor representative.

Lessons Learned

Vendor representative visits, particularly in remote areas, are not totally effective unless detailed site problem information and requirements are sent well in advance.

I have witnessed and been told of many ineffective vendor representative visits, and I have discovered that none of the noted best practices above had been followed.

Benchmarks

This best practice has been used since 1984 during a large petrochemical complex construction project and has since that time consistently produced effective vendor representative visits.

Best Practice 10.4

Follow these critical machinery foundation preparation best practices to ensure optimum machinery life cycle reliability:

Review the contractor's machinery foundation procedure prior to the start of any equipment installation to confirm:

- Proper foundation preparation procedures
- The contractor's procedure, types of bolts and bolt arrangement, meet or exceed the equipment vendor's requirements and bolt arrangements
- The proper location of all foundation bolts to prevent baseplate bolt hole machining
- Proper preparation of foundation bolts (stress relieving, etc.)
- Proper leveling of machinery

Having a critical machinery foundation preparation procedure that has been reviewed and accepted by the construction contractor prior to field construction will ensure a critical machinery foundation of the highest reliability, and optimize the overall machine train reliability.

Lessons Learned

Installing critical machinery without a foundation procedure that is based on proven installation experience and vendor procedures will reduce the machinery train's life cycle reliability.

I have been involved with project delays of over a month that have resulted from not having a foundation preparation procedure for critical compressor trains. In this case, the subject compressor foundation bolt location was not coordinated with the location of the reinforcing bars, which required complete excavation of the foundation to achieve the proper location of the foundation bolts.

Benchmarks

This best practice has been used since the issue noted above was experienced (1988), and requires that a critical machinery foundation procedure be prepared and reviewed by all project critical machinery vendors prior to the project construction phase. This best practice has resulted in critical machinery foundations that have not required rework and have produced critical machinery trains of the highest reliability (99.7%+).

B.P. 10.4. Supporting Material

Foundations

The installation of properly designed and constructed foundations plays an important part in the long term availability of equipment. This section will cover the major aspects of sound foundations.

General considerations

After the proper civil work is done, and the foundation is designed in accordance with specifications, there are certain general considerations required. These are shown in Figure 10.4.1.

Foundations must be rough enough to allow grout to adhere. The elevation of the top surface should allow at least one inch of grout under the base plates or sole plates. When machinery is mounted directly on the foundation, sole plates must be

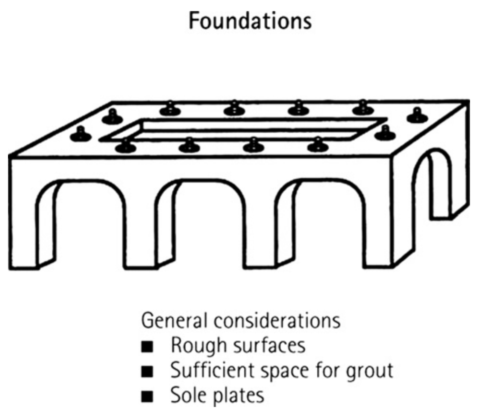


Fig 10.4.1 • Foundations

provided. It is wise to epoxy-grout sole plates to facilitate easy removal and installation of equipment during maintenance. Sole plates must be level in themselves and all other planes.

Foundation bolts

Each equipment manufacturer has foundation bolt requirements. It is a good idea to review the contractor's foundation bolt arrangements prior to any equipment installation and ensure that the contractor's procedure, types of bolts and bolt arrangement, meet or exceed the equipment vendor's requirements. Many a project has been delayed by not incorporating this requirement. Figure 10.4.2 shows three typical installation arrangements for anchor bolt installations.

A case history for the installation of a large reciprocating compressor in a refinery shows how poor planning can cause significant construction delays. The foundation re-bar pattern was not coordinated with the foundation bolt pattern for the crankcase and crossheads. After the foundation was set, with the foundation bolts in place, it was discovered that the bolt locations had moved from the original positions, and that the crankcase could not be positioned over the foundation bolts. The re-bar had interfered with the foundation bolts causing them not to be correctly positioned. The result was complete foundation rework to correctly position the foundation bolts that resulted in a delay of one month to the construction schedule. At that time the lost revenue was approximately $1 MM per day.

One last word regarding foundation bolts. It is easy to mislocate the bolt locations relative to the machinery base plate holes. Before bolt holes are randomly elongated to facilitate misplaced location bolts, all facts should be discussed with both the contractor and the vendor of the equipment. Irresponsible action regarding elongation of bolt holes has caused machinery problems. Fabricated foundation bolts on which welding is used in the fabricated assembly must be stress relieved after welding.

Leveling

Figure 10.4.3 presents the basics of leveling of equipment.

Equipment must be leveled within a tolerance of 0.05 mm per meter and confirmed with a calibrated engineer's level. Any special leveling instructions given by the vendor must be followed. In the case of reciprocating equipment, it is important that shims straddle hold down bolts. When jacking screws are used for leveling equipment it is not necessary to remove them after grouting, but they must be backed off at least two turns and the hold down bolts must be retorqued to their correct value after grout has adequately cured. There should be a minimum of one jacking bolt for each hold down bolt.

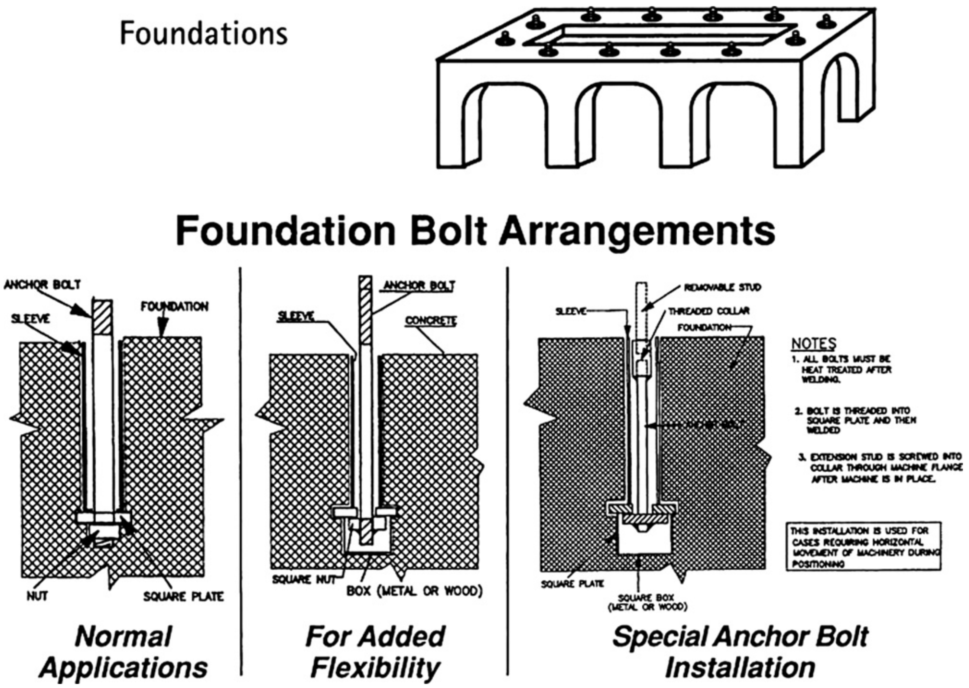


Fig 10.4.2 • Anchor bolt installations (Courtesy of ME Crane, Consultant)

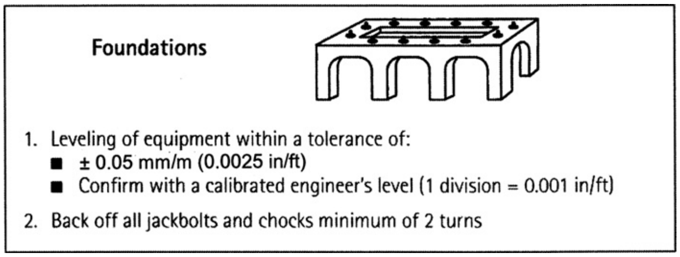


Fig 10.4.3 • The basics of leveling of equipment

Best Practice 10.5

Use epoxy grout for all machinery foundations, in accordance with the epoxy grout vendor's procedures, and employ an experienced epoxy grout contractor to produce a grout foundation that will last for the life of the installed equipment.

Only epoxy grout, when properly applied, will ensure a machinery baseplate support that will last for the life of the installed equipment.

Ensure that prior to the construction phase of the project:

- Epoxy grout will be used for all machinery.
- An accepted epoxy grout procedure is approved by all parties.
- The selected grout contractor has experience in epoxy grout applications in the geographical area (environment) where the plant is located.

Lessons Learned

Failure to specify the use of epoxy grout for all machinery installations, have an approved grout procedure in place and use an experienced epoxy grout contractor has caused significant project delays, and produced foundations that required re-grouting before or during the first scheduled plant turnaround.

Benchmarks

This best practice has been used since the mid-1980s when significant epoxy grouting issues were experienced. The final solution, after a project delay, was to require that the epoxy grout vendor representative come to the site and conduct epoxy grout training for the grout contractor, who did not have any experience in epoxy grout installation in the geographical area in which the plant was located. Since that time, we have required that the site grout procedure be reviewed and approved prior to start of construction and that the contractor experience be confirmed.

B.P. 10.5. Supporting Material

Grout (general)

The grouting plays an important role in the availability of equipment. Improper grout type and application has caused many unscheduled shutdowns in the field. Figure 10.5.1 presents some general grout considerations that have been proven through many long, hard construction projects.

1. Approved grouting procedure
2. Epoxy grout for:
   - Greater than 75 kw (100 hp)
   - All axial, centrifugal or reciprocating compressor trains
3. Special environmental conditions
   - Temperatures > 50°C (140°F)
4. Proper surface preparation
   - Clean
   - Chipped
   - Water free (for epoxy grout)
   - Grease anchor bolts, jackbolts, chocks

Fig 10.5.1 • General grout considerations

Most important is an approved grouting procedure. This is not a simple procedure that states the type of grout and how much will be used, but a detailed procedure specifying the equipment used for proper grout pours, the forms, the form preparation, the details concerning depth of pour, specifications for grout, etc. It has been our experience that contractors are not experienced in proper grouting procedures. Remember, the installation phase is only a short period in the life of equipment. The decisions made during grouting will affect the equipment for its entire lifetime.

Epoxy grout is usually required for equipment greater than 75 kW (100 hp) and all reciprocating types of rotating equipment. Although epoxy grout is much more expensive than conventional grouts, it certainly pays out in the long run since it is impervious to oil and resists cracking. In the application of any grouts, ambient conditions are very important. Be sure that the site grouting procedure takes the local ambient conditions into account.

Like most jobs, proper preparation significantly affects the quality of the finished product. Clean, chipped, water-free (for epoxy grout) foundations are a necessity. Also anchor bolts, jack

bolts and chocks should be greased for ease of operation once the grout starts to cure.

Epoxy grout

Epoxy grout is clearly the grout of choice for critical (un-spared) equipment installation, since it lasts the longest and is impervious to most external sources. Figure 10.5.2 presents some epoxy grout considerations.

It is important that epoxy grout be poured in accordance with the grout manufacturer's recommendations. Most contractors need experience in epoxy grout installation. In fact, it has been our experience that an on-site demonstration of epoxy grout by the epoxy grout manufacturer is a worthwhile expenditure, and saves countless repours and project delays. Some other epoxy grout considerations are:

- Wax or grease all forms
- Limit thickness of pour to approximately 10 cm (4 in)
- Ensure that proper mixing and pouring tools are available
- Fill bolt sleeves or pockets to ensure that grout does not spill into these areas
- Check for voids and fill with epoxy pressure grout when required
- Seal grout holes in metal base plates

1. In accordance with grout manufacturers' procedures
2. Wax or grease all forms
3. Limit thickness of pour to 10 cm (4")
4. Fill bolt sleeves or pockets
5. Check for voids. Fill with epoxy pressure grout
6. Seal grout holes

Fig 10.5.2 • Epoxy grout considerations

Non-shrink or cementous grout

Frequently for cost considerations, contractors will attempt to use non-shrink or cementous grout on large pieces of equipment. In certain instances this is acceptable, as in the case of large oil console foundations, which are usually installed with cementous grout for reasons of mass. This action solidifies the console base and minimizes pipe and component vibration.

It should be mentioned that some types of non shrink grout incorporate metal filings. It has been found in some instances that these filings will corrode with time, and cause separation of grout from the foundation. Prior to application of any grout, a proper procedure and details of the grout must be defined. In the event of any doubts ask the original equipment manufacturer regarding his considerations.

Best Practice 10.6

Require that the following best practices be executed to ensure proper process piping to machinery flange alignment and foundation continuous support (soft foot):

- With process piping disconnected, confirm that the baseplate does not rise more than 0.05 mm (0.002 in) when the foundation bolts adjacent to a dial indicator are loosened.
- Process piping is floated (not connected) to the machinery flanges resulting in a maximum mating to machinery flange parallel difference of 0.25 mm (0.010 in) at four locations 90 degrees apart.
- All flange bolts can be installed without any external forces applied to the process piping.
- When the process piping is connected to the machinery, two dial indicators reading shaft or coupling hub movement, mounted at 90 degrees, do not register movement greater than 0.05 mm (0.002 in). Note: Dial indicators must be mounted off of the machine (on the coupling guard or an independent support).

The above action will produce on-schedule start-ups and optimum reliability for the life of the machinery.

Lessons Learned

Improper process piping attachment and foundation procedures cause prolonged machinery component (bearings, seals and internal rub) problems that have resulted in low component MTBFs (less that 12 months).

Benchmarks

This best practice has been used since the mid-1980s to result in optimum installed machinery component MTBFs (> 100 months).

B.P. 10.6. Supporting Material

Have you ever been called into your supervisor's office and questioned on how to properly install the equipment or a component? Have you ever had the experience of installing a component (bearing) only to have it fail repeatedly over the following months and became a 'bad actor'? What is the problem? Is it your assembly procedures, the installation, the equipment or the process?

The subject of this section is equipment pipe stress and soft foot. Without a doubt, these factors are prime contributors to 'bad actors'. They both are relatively easy to prove; however, they can be very difficult to correct. The purpose of this section is to present the reasons why excessive pipe stress and soft foot cause bad actors, how to prove these problems exist and the most cost-effective method to correct them.

Before we can understand *how* pipe stress and soft foot can cause equipment component failures, we must know *what* pipe stress and soft foot are! Figure 10.6.1 presents these facts.

Naturally, all equipment cases are designed to accommodate reasonable pipe loads and minimal load due to soft foot.

Pipe stress and soft foot exert failure producing forces on the equipment casing from:

- Top, side or bottom flanges (pipe loads)
- Support feet (soft foot)

Fig 10.6.1 • Pipe stress and soft foot

However, Figure 10.6.2 shows what the equipment designer assumes in this regard.

- Under the limit on external pipe force (on assembly dwg)
- Under the limit on external pipe moments (on assembly dwg)
- All support feet are flat and in the same plane
- Foundation under all support feet has been leveled (shimmed if necessary with stainless steel shims)
- All external pipe(s) and support feet are properly connected

Fig 10.6.2 • External force design assumptions

How pipe stress and soft foot can cause component failure

Figures 10.6.3 and 10.6.4 show a typical single stage overhung pump and a general purpose steam turbine respectively.

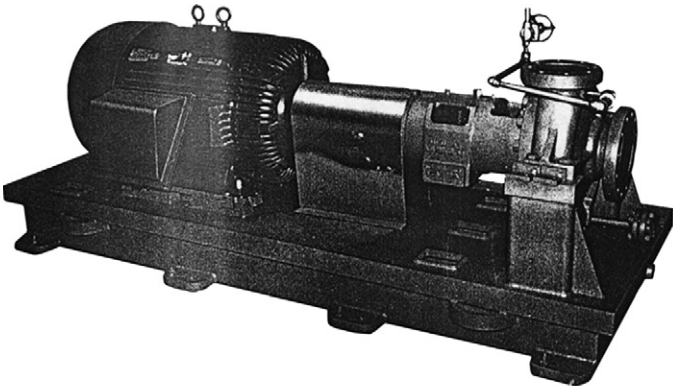


Fig 10.6.3 • Single stage overhung pump (Courtesy of Union Pump Co).

In both figures, the process pipes are not connected. If, in addition, both the pump and steam turbine were not coupled or bolted to their bases, what would cause the load (force) on the bearings?

Hopefully your answer was the rotor. Let's use the pump in the following discussion (Figure 10.6.3). However, everything discussed will equally apply to the steam turbine or any other type of equipment.

Please refer to Figure 10.6.5, which shows a typical anti-friction bearing that would be used for the pump radial bearing.

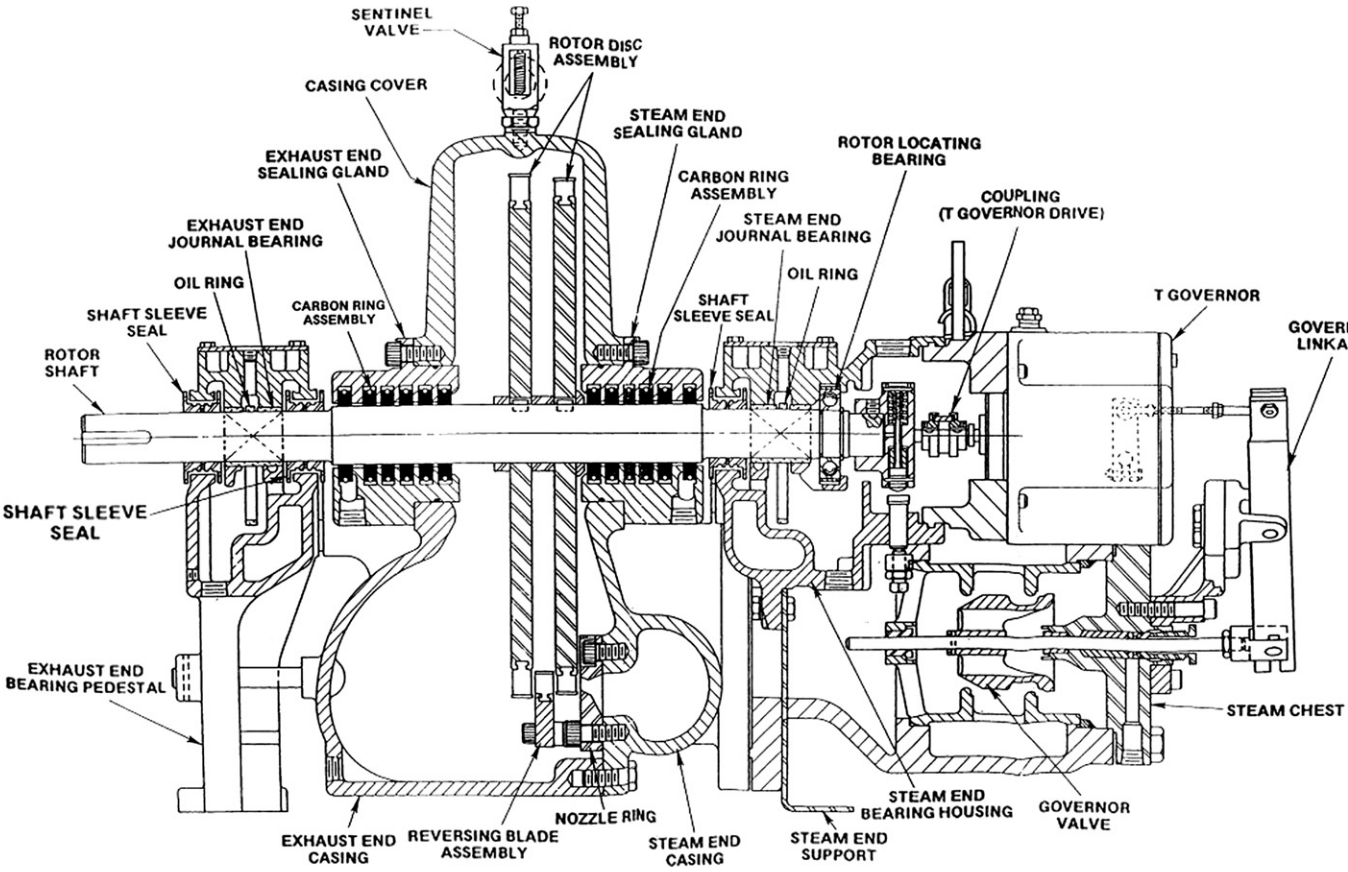


Fig 10.6.4 • General purpose steam turbine (Courtesy of Elliott Co.)

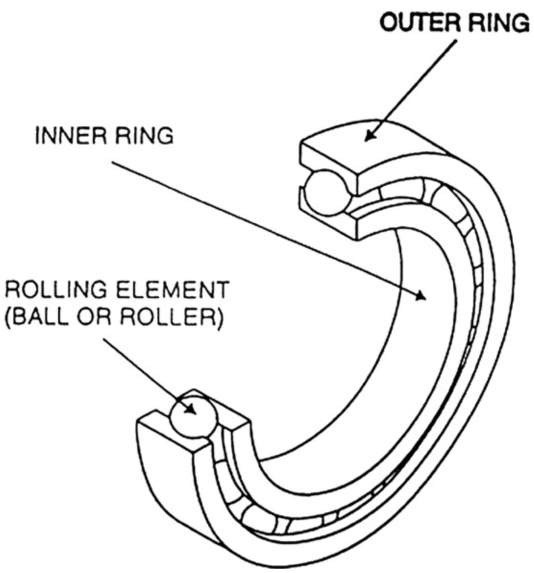


Fig 10.6.5 • Radial bearing

Figure 10.6.6 presents the sources of the forces on any radial and/or thrust bearing regardless of the bearing type (anti-friction or sleeve).

For the pump in Figure 10.6.3, please describe the forces that the designer takes into account during the bearing *selection*. (Remember – anti-friction bearings are not custom designed.) Circle the forces in Figure 10.6.6 that should be considered during the bearing selection.

- Increased process pipe forces and moments
- Foundation forces ('soft' foot, differential settlement)
- Fouling or plugging of impeller
- Misalignment
- Unbalance
- Rubs
- Improper assembly clearances
- Thermal expansion of components (loss of cooling medium, excessive operating temperature)
- Radial forces (single volute – off design operation)
- Poor piping layouts (causing unequal flow distribution to the pump)

Fig 10.6.6 • Sources of forces

Now please refer to Figure 10.6.7 which describes the relationship to determine the life of any anti-friction bearing.

As an exercise, let's determine the bearing life for the following cases; Case 1 – no excessive external load, Case 2 – additional soft foot load, and Case 3 – additional pipe stress load. We will note this information in Figure 10.6.8.

In Figure 10.6.8, Case 1 represents a bearing selected in accordance with industry standards that was installed correctly. That is, the predicted life of the bearing is in excess of 25,000 hours or 3 years' continuous operation.

Cases 2 and 3 are a different story! Observe the dramatic effect of additional forces on the equipment casing from either soft foot or piping forces.

'B' or 'L' - 10 life is defined as the life in hours that 9 out of 10 randomly selected bearings would exceed in a specific application.

$$\text{'B' or 'L' - 10 life} = \frac{16700}{N}\left[\frac{C}{F}\right]^3$$

Where: N = RPM
C = Load in lbs that will result in a bearing element life of 1,000,000 revolutions
F = Actual load in lbs

Fig 10.6.7 • 'B' or 'L' - 10 life

Case	1	2	3
N (speed)	3600 rpm	3600 rpm	3600 rpm
C (bearing dynamic load factor – lbs)	3000	3000	3000
F (total actual bearing load – lbs)	170	500	1000
Condition	As designed	Additional soft foot forces	Additional pipe load forces
L-10 life years	25,495 hours	1002 hours	125 hours
Cause of early failure	No early failure specified minimum life was exceeded	Excessive soft foot forces	Excessive pipe load forces

Note: use the relationship in Figure 10.6.7 to determine the bearing L-10 life.

Fig 10.6.8 • External loads on equipment example

If your manager or an operator had to complete this exercise, he probably would have listed the 'machinist' as the cause of failure! Hopefully, this exercise has clearly demonstrated why components, especially bearings, can suddenly fail for no apparent reason. These facts are presented in Figure 10.6.9.

Figure 10.6.10 has been modified to show the force path from excessive discharge flange loadings to the bearing bracket.

Figures 10.6.11 and 10.6.12 show the orientation of external flange forces and moments are referred to in the table shown in Table 10.6.1.

Tables 10.6.1 and 10.6.2 show that the allowable forces and moments for most pumps are very low!

They exert forces in excess of design limits on the components:
- Casing
- Bearings
- Rotor
- Seals
- Wear rings

The path of the forces is from the external force through the casing, bearing bracket to the components.

Fig 10.6.9 • How excessive pipe stress and soft foot forces cause equipment component failure

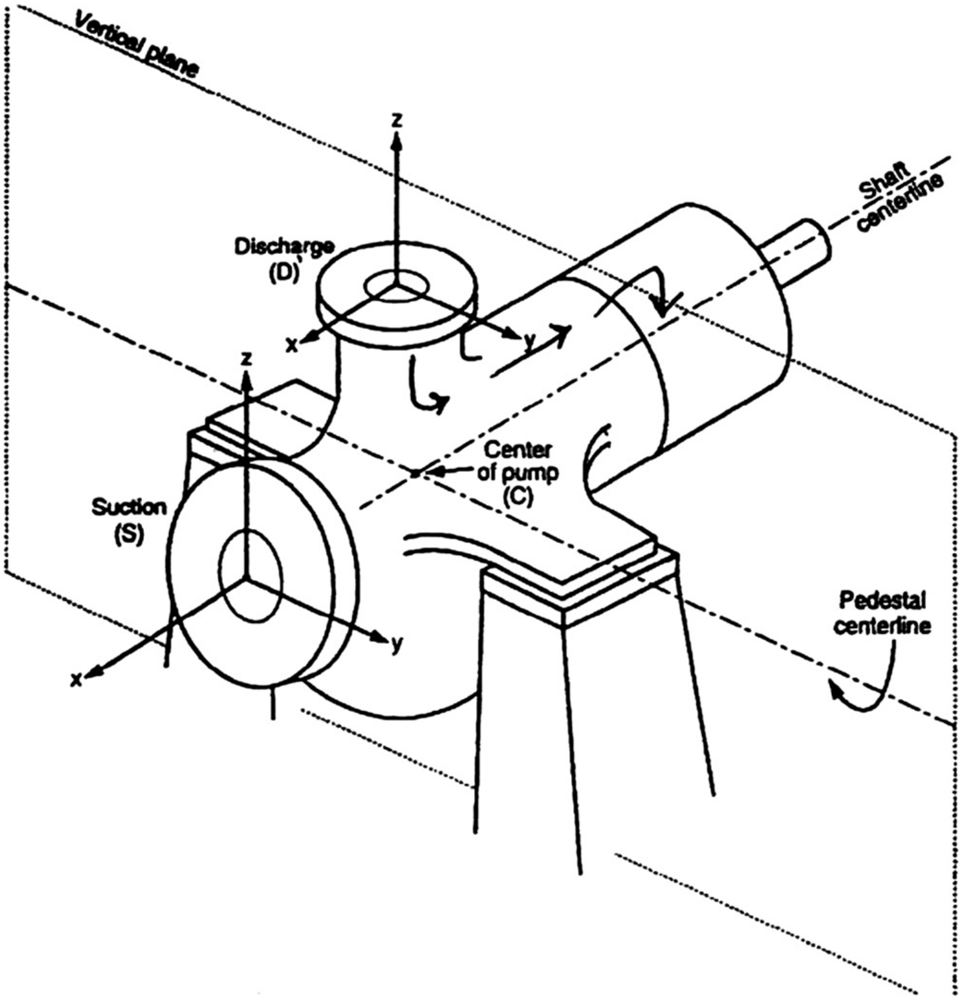


Fig 10.6.10 • Force path from excessive discharge flange loadings to the bearing bracket (Courtesy of API)

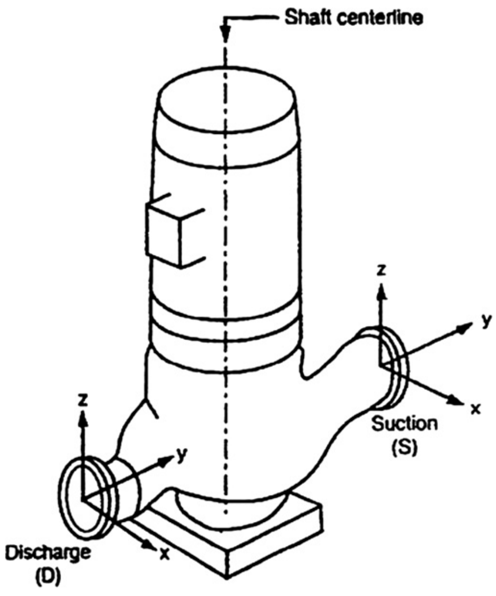


Fig 10.6.11 • Vertical in-line pump (Courtesy of API)

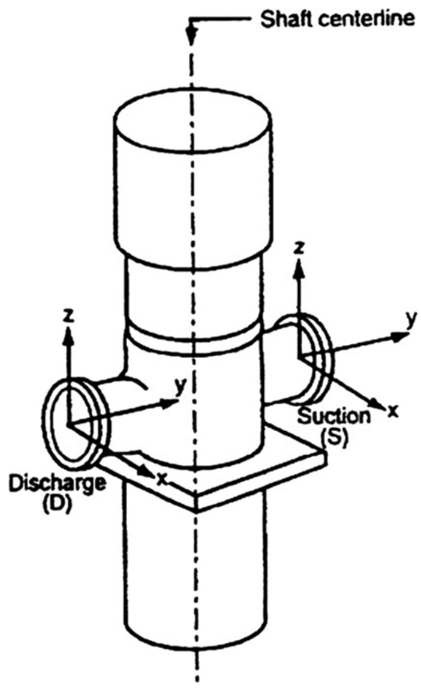


Fig 10.6.12 • Vertically suspended double-casing pump (Courtesy of API)

Table 10.6.1 Nozzle loadings (Courtesy of API)
Table A — Nozzle loadings (SI units)

Note: each value shown below indicates a range from minus that value to plus that value; for example 710 indicates a range from −710 to +710

	Nominal size of flange (NPS)								
Force/moment	**2**	**3**	**4**	**6**	**8**	**10**	**12**	**14**	**16**
Each top nozzle									
FX	710	1070	1420	2490	3780	5340	6670	7120	8450
FY	580	890	1160	2050	3110	4450	5340	5780	6670
FZ	890	1330	1780	3110	4890	6670	8000	8900	10230
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850
Each side nozzle									
FX	710	1070	1420	2490	3780	5340	6670	7120	8450
FY	890	1330	1780	3110	4890	6670	8000	8900	10230
FZ	580	890	1160	2050	3110	4450	5340	5780	6670
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850
Each end nozzle									
FX	890	1330	1780	3110	4890	6670	8000	8900	10230
FY	710	1070	1420	2490	3780	5340	6670	7120	8450
FZ	580	890	1160	2050	3110	4450	5340	5780	6670
FR	1280	1930	2560	4480	6920	9630	11700	12780	14850
Each nozzle									
MX	460	950	1330	2300	3530	5020	6100	6370	7320
MY	230	470	680	1180	1760	2440	2980	3120	3660
MZ	350	720	1000	1760	2580	3800	4610	4750	5420
MR	620	1280	1800	3130	4710	6750	8210	8540	9820

Note 1: F = force in Newtons; M = moment in Newton meters; R = resultant. See Figures 10.6.11 and 10.6.12 for orientation of nozzle loads (X, Y and Z)
Note 2: Coordinate system has been changed from API Standard 610, 7th edition, convention to ISO 1503 convention.

The root causes of excessive pipe stress and soft foot

Refer to Figure 10.6.13, the machinery environment.

An associate of mine has a favorite quote regarding the machinery environment.

'Stand at the equipment unit and rotate yourself 360°. Everything that you see can and will affect the reliability of this piece of machinery'.

As shown in the last section the cause of component failure is the excessive forces exerted on the equipment from:

- The piping
- The foundation (soft foot)

What then are the possible causes? There are many. We will divide the possible causes into the following categories:

- Design
- Construction
- Plant conditions

The possible root causes are presented in Figures 10.6.14, 10.6.15 and 10.6.16.

Please refer to foundation and grout best practices presented in B.P: 10.4 and B.P: 10.5 for additional information.

The rotating equipment environment
- Process condition change
- Piping and foundation change
- 'Unit' (driven, driver, transmission, auxiliaries)
- Ambient conditions

Fig 10.6.13 • The rotating equipment environment

Table 10.6.2 Nozzle loadings (Courtesy of API)
Table B – Nozzle loadings (US units)

Note: Each value shown below indicates a range from minus that value to plus that value; for example 160 indicates a range from −160 to +160.

	Nominal size of flange (NPS)								
Force/moment	**2**	**3**	**4**	**6**	**8**	**10**	**12**	**14**	**16**
Each top nozzle									
FX	160	240	320	560	850	1200	1500	1600	1900
FY	130	200	260	460	700	1000	1200	1300	1500
FZ	200	300	400	700	1100	1500	1800	2000	2300
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each side nozzle									
FX	160	240	320	560	850	1200	1500	1600	1900
FY	200	300	400	700	1100	1500	1800	2000	2300
FZ	130	200	260	460	700	1000	1200	1300	1500
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each end nozzle									
FX	200	300	400	700	1100	1500	1800	2000	2300
FY	160	240	320	560	850	1200	1500	1600	1900
FZ	130	200	260	460	700	1000	1200	1300	1500
FR	290	430	570	1010	1560	2200	2600	2900	3300
Each nozzle									
MX	340	700	980	1700	2600	3700	4500	4700	5400
MY	170	350	500	870	1300	1800	2200	2300	2700
MZ	260	530	740	1300	1900	2800	3400	3500	4000
MR	460	950	1330	2310	3500	5000	6100	6300	7200

Note 1: F = force in pounds; M = movement in foot pounds; R = resultant. See Figures 10.6.11 and 10.6.12 for orientation of nozzle loads (X, Y and Z)
Note 2: Coordinate system has been changed from API Standard 610, 7th edition, convention to ISO 1503 convention.

- Pipe stress calculation error
- Improper spring hanger selection
- Improper soil analysis assumptions
- Improper foundation design

Fig 10.6.14 • Possible causes for excessive pipe stress and/or soft foot (design)

- Using equipment as a 'pipe support'
- Improper installation of fixed spring supports
- Poor foundation and/or grout preparation
- Poor foundation and/or grout pour

Fig 10.6.15 • Possible causes for excessive pipe stress and/or soft foot (construction)

- Settling pipe support foundations
- Cracked grout and/or foundation (concrete)
- Locked spring supports
- Improperly installed new spring supports
- Shim corrosion under pipe supports
- Shim corrosion between equipment feet and baseplate
- Shims vibrating loose under pipe supports

Fig 10.6.16 • Possible root causes for excessive pipe stress and/or soft foot (plant conditions)

There have been numerous examples, especially in the Middle East, of poorly prepared foundations and grouting. Careful attention must be paid to the quality of the water used, the type of grout and the method of grouting followed.

It is strongly recommended that an epoxy grout be used for all rotating equipment and a grouting procedure, approved by a reputable epoxy grout manufacturer, be utilized.

Figure 10.6.17 has an important message: 'If you suspect excessive pipe stress and/or soft foot forces, get out and thoroughly walk around the affected machine'.

Condition monitoring indications of excessive pipe stress and soft foot

At this point, we have covered the function of the two most important components in rotating equipment:

- Bearings
- Seals

The components most commonly affected by excessive piping and/or foundation (soft foot) forces are the bearings and couplings, although seal reliability can be affected along with the bearings in extreme cases. Figures 10.6.18 and 10.6.19 present the parameters to monitor, as well as the limits for anti-friction and sleeve type radial bearings.

As previously stated, excessive pipe strain and soft foot exert forces beyond the design limits on equipment components. In the case of bearings, the forces will be transmitted from the source (pipe and/or foundation) through the casing to the bearing housing, to the bearing. Figure 10.6.20 presents the facts concerning how to determine by condition monitoring if we have a pipe stress and/or soft foot problem.

Bearing (anti-friction)

Parameter	Limits
1. Bearing housing vibration (peak)	0.4 inch/sec (10 mm/sec)
2. Bearing housing temperature	185°F (85°C)
3. Lube oil viscosity	off spec 50%
4. Lube oil particle size	
• non metallic	25 Microns
• metallic	any magnetic particle in the sump
5. Lube oil water content	below 200 ppm

Fig 10.6.17 • Condition monitoring parameters and their alarm limits

Bearing (hydrodynamic)

Parameter	Limits
1. Radial vibration (peak to peak)	2.5 mils (60 microns)
2. Bearing pad temperature	220°F (108°C)
3. Radial shaft position*	> 30° change and/or 30% position change
4. Lube oil supply temperature	140°F (60°C)
5. Lube oil drain temperature	190°F (90°C)
6. Lube oil viscosity	off spec 50%
7. Lube oil particle size	> 25 microns
8. Lube oil water content	below 200 ppm

*except for gearboxes where greater values are normal from unloaded to loaded

Fig 10.6.18 • Condition monitoring parameters and their alarm limits

- More than one (1) bearing failure, rotor breakage, or coupling failure per year
- Unexplained high vibration (usually indicating misalignment)
- Unexplained high bearing housing temperature
- Pipe supports close to equipment *not* vibrating, equipment is vibrating

Fig 10.6.19 • Condition monitoring indications of excessive pipe forces and/or soft foot

Confirming excessive pipe stress and/or foundation forces (soft foot)

Once the root causes of machinery component failure are suspected, they must be confirmed. Once confirmed, a cost-effective action plan must be developed that will ensure implementation at the earliest opportunity. This section presents this important information.

Confirming excessive pipe stress and/or foundation forces

In order to implement any action, we had better be sure our suspected root causes are correct. If they are not, we will always have a difficult time obtaining approval for any future recommendation.

Figures 10.6.21 and 10.6.22 present the guidelines for confirmation of excessive piping stress and/or soft foot.

Correcting excessive pipe stress and foundation forces on equipment

Of all the different problems with rotating equipment, the resolution of excessive pipe stress is the most difficult.

Why? Correction can involve extensive work that will require a significant amount of safety permits and may even require process unit shutdown.

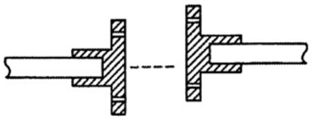
Figure 10.6.23 shows the suggested excessive pipe stress solution procedure. It is naturally arranged in a cost-effective order (simplest, least costly action first).

Correcting soft foot problems can be extremely simple if equipment support feet are not level to the foundation. In this case, stainless steel shims can be added. However, in some cases, baseplates can become distorted and/or the foundation can experience differential settlement over a period of time. This case usually requires that a complete new foundation be designed and installed at the next T&I. A short-term fix can be to install stainless steel shims to temporarily correct the problem.

Implementation of the action plan

Correction of pipe stress problems is usually the most difficult problem to obtain action plan implementation for.

Shaft alignment

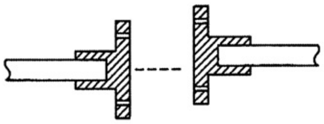


Preliminary considerations

1. Obtain thermal shafts and machine growth calc's establish "cold offsets"
2. Coupling hubs installed in accordance with OEM's procedures
3. Set proper B.S.E. dimension
4. Test for "soft foot" 0.05 mm (0.002") maximum allowable differential rise
5. Use only stainless shims – minimize number of shims
6. Shims must straddle hold down bolts

Fig 10.6.20 • Shaft alignment – preliminary considerations

Shaft alignment



Alignment change limits when connecting piping

1. Mount dial indicators independent of machinery horizontally and vertically
2. Reading on coupling flanges
3. Zero out indicators
4. Tighten using specified sequence and torque values
5. Maximum shaft movement = 0.05 mm (0.002")
6. Dowell if required by OEM

Fig 10.6.21 • Shaft alignment – alignment change limits when connecting piping

I Confirm excessive pipe stress[1] (refer to Figure 10.6.24). Also confirm pipe bolting can be removed without a 'come along'

II Walk piping system and confirm proper installation per piping isometrics
- Proper pipe support shims
- Spring supports free to move
- No obvious pipe misalignment

III Correct excessive pipe stress by:[2]
- Attempting rebolting at the next flange
- Using 'Dutchman' with flexitallic gaskets (each side)
- Heating of pipe for alignment
- Pipe modification at next T&I

Notes: [1]Work permits required
[2]All items in III must be confirmed correct per Box 10.6.23.

Fig 10.6.22 • Suggested excess pipe stress solution procedure

Why? It is costly, exposes the plant to possible safety problems and can result in a process unit shutdown. Usually, it should be planned for a T&I. I have found that the guidelines in Figure 10.6.24 provide the best probability of implementation.

1. Float pipe to machine. Machine is not a pipe support
2. Lock pins on spring or pipe supports adjacent to machinery kept in place until system filled
3. Flanges parallel on machined surfaces and no come alongs! (Bolts can be removed without excessive force)
4. Flange faces within 0.0010"

Fig 10.6.23 • Pipe considerations

1. Clearly stating impact of problem on plant profit
2. Prepare a brief statement of:
   - The problem
   - Action plan and confirmation of success (past experience)
   - Cost of failure to date
   - Cost of solution
   - The impact on plant profit (loss)
3. Being confident!
4. Being professional!
5. Providing timely updates and final report on completion

Fig 10.6.24 • Obtain and maintain management support by

Best Practice 10.7

Implement the following 'best practice' oil flushing procedure in your plant to produce optimum system cleanliness and minimize flushing time.

Optimal procedures have the following features:

- Nitrogen bubbling
- Isolation of oil coolers for initial flush
- Temporary valved bearing and seal housing bypasses which maximize oil velocities
- Means of ensuring drain lines do not flood during flushing
- Minimizing flush screen number and suggested locations
- Flush screen details to ensure integrity
- Flushing log details for optimum flush effectiveness

A proven effective oil flush procedure incorporating the above and details is contained in the supporting material for this best practice below.

Lessons Learned

Most oil flush procedures take too long and are not effective.

Oil flush procedures are rarely performed on time, or result in filter cartridge changes (greater than one year multiple when filter differential exceeded 1.5 bar).

Benchmarks

This best practice has been used in varying degrees since nitrogen bubbling was demonstrated in the mid-1970s after a refinery fire. It has resulted in oil flush times for large equipment trains (greater than 20 MW) of less than 2 weeks and filter change-out periods of greater than 1 year.

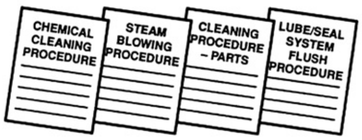
B.P. 10.7. Supporting Material

Flushing of lube oil and seal oil systems

All critical equipment incorporates bearings that continuously support the shaft on an oil film approximately 15–20 microns thick. Even though the lubricating and lube oil systems incorporate many components responsible for supply of clean, cool oil at proper pressure, temperature and flow conditions, fine debris existing in pipes between flanges and gaskets, in voids of coolers and other vessels can supply fine metallic and non-metallic particles that can cause significant damage to bearings and to the equipment. It is therefore imperative that a cost effective flushing procedure be implemented in the field. We have included a cost-effective, proven, oil system flushing procedure at the end of this section that has repeatedly saved valuable construction time, and resulted in the cleanest possible oil system that will minimize filter changes during operation.

Figure 10.7.1 provides some considerations regarding proper flushing procedures. Again, it must be pointed out that the objectives of the contractor and the end user are different.

Cleaning of equipment and associated piping



Flushing of lube and seal oil systems

Procedure outline (refer to Appendix)

1. Hand clean major components (lint free cloths)
2. Maintain flushing log
3. Use auxiliary system components
4. Add temporary jumpers and hand valves
5. Minimize screens (suggested locations)
   - Before and after cooler (1st flush)
   - Supply lines (at jumpers)
   - Return (initial flushes only)

Fig 10.7.1 • Cleaning of equipment and associated piping – flushing of lube and seal oil systems

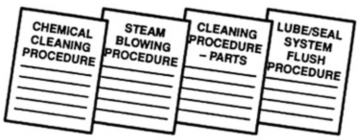
Maintaining a flushing log is important to ensure that all flushing procedure requirements are carried out. In addition, periodic inspection of flushing screens is suggested to ensure the job is proceeding smoothly. It is a common occurrence that the length of a flushing cycle frequently exceeds its predicted time. The procedure provided in the appendix and the guidelines presented in this section will ensure that flushing will be accomplished in the minimum amount of time. However effective a procedure is, it is only as effective as its implementation. Therefore, monitoring the flushing operation is imperative.

Figure 10.7.2 presents additional considerations regarding flushing.

One final word regarding flushing: it has come to our attention that in an effort to ensure totally clean systems, too many flushing screens are employed, resulting in excessive flushing times. A common example of this practice is the installation of flushing screens in drain lines. Since drain lines are designed to operate only half full, a flushing screen will eventually become partially plugged. When it does, the level of the oil in the affected drain line will rise and will very effectively clean the top of the pipe. Since this is an abnormal occurrence, and since one function of the reservoir is to contain the sludge and prevent it from re-entering the system, excessive use of screens in drain lines should be discouraged.

A practical recommendation would be to install one screen in the main return to the reservoir for only the initial phases of flushing. A convenient way to ensure that this screen is not becoming plugged is to connect a temporary piece of plastic tubing to the bottom of the drain line close to the flushing screen and monitor the level of the oil. When the level begins to rise in the plastic tube, shut down and clean the screen before the level becomes excessive. Once the debris on the screen has leveled out, it is suggested that the screen be removed.

Cleaning of equipment and associated piping



Flushing of lube and seal oil systems (cont.)

Procedure outline (refer to Appendix)

6 Screens (100 mesh SS, 60 mesh backup)

7 Heat cycle

8 Nitrogen bubble

9 Final flush
- white cloth
- acceptance criteria
  - non metallic
  - less than 74 micron

10 Remove all screens, visually inspect, replace all components, install new cartridges, fill reservoir

Fig 10.7.2 • Cleaning of equipment and associated piping – flushing of lube and seal oil systems (cont.)

Final inspection and start-up checks

The contractor and the customer must mutually agree on an acceptable hand over and sign off procedure. As a minimum the following items should be completed:

- All installation and long term storage requirements
- All auxiliary piping, installation, insulation and painting
- All instrument and electrical loop checks complete and proven operable
- All spare parts received and equipment documentation available

First start, run-in and initial operation

After the equipment has been handed over, first start, run-in and initial operation should be performed in accordance with proven start-up procedures and planned operation procedures. Figure 10.7.3 provides these considerations.

Included at the end of this section are the following additional documents that contain valuable guidelines for site pre-commissioning and commissioning:

- Site pre-commissioning best practice list for your use
- Functional lube/seal system test procedure outline
- Electro-hydraulic governor functional test procedure outline
- Steam turbine solo run functional test procedure outline

Auxiliary system flushing procedure

This information is provided courtesy of M.E. Crane, Consultant.

The following procedure is presented as a guide for field flushing of lube and seal systems. In order to be fully productive, it is recommended that all requirements noted herein be strictly followed.

1. General

1.1 Flushing operation will be carried out by the designated party (contractor or owner).

1.2 Cleanliness of oil console, equipment skid, overhead seal oil tanks, piping systems and screens shall be determined by mutual agreement between equipment vendor, contractor and owner.

1.3 Owner and vendor shall keep a log for general review of flushing progress. Master flow sheets shall be kept by owner and updated to progress. An entry shall be made during each shift.

1. Final inspection and hand-over accepted (all work complete, all punch list items complete, loop checks complete)
2. Start-up procedure (optional) approved
3. Air run (compressors) optional
   - Procedure
   - Safety considerations
   - Limits of operations

Fig 10.7.3 • First start, run-in and initial operation

1.4 In general the oil flush shall be performed using selected permanent auxiliary equipment which is part of the vendor supply package. This will include the following:
- Auxiliary oil pump (electrical) and main oil pump if possible.
- Main oil filter to be in position for all flushing.
- Main oil reservoir, degassing tank, overhead seal oil tanks and seal oil traps.
- Skid piping.
- Selected instruments and controls.

2. Preparation

2.1 Any residual oil from factory testing must be removed from the reservoir and filters. Relief valves must have been checked prior to flushing.

2.2 The reservoirs, degassing tanks and filter casing must be wiped clean, inspected and approved by the owner. All cleaning must be carried out using a lint free cloth. When filters are open for cleaning, special care should be taken to avoid contaminates falling into 'clean' side of filter housing.

2.3 The filters must be verified to be in place and satisfactory for flushing. Examination of filters will include checks for bypassing, inside or outside of filter housing.

2.4 All lube and seal oil interconnecting piping will be installed consistent with normal operating conditions in accordance with bypass piping arrangements as mutually agreed upon between vendor and owner.
- All equipment supply and drain piping is required to be flushed during entire flushing operation. Location of all valves, bypasses and screens shall be in accordance with marked-up P&IDs of lube oil, seal oil and control oil system.
- Add hand valves suitable to meet the operating requirements during flushing to *all* piping supply points.
- All lines to the steam turbine throttle valve, servo motors and dump devices will be flushed in accordance with a manufacturer's requirements.
- Overhead seal oil tanks will be flushed by jumpering to compressor reference gas lines between compressor, overhead tanks and drainers are required to be flushed.

2.5 Stainless steel 100-mesh screens with back-up 60-mesh screens with a number of spares must be fabricated with retaining gaskets and installed at selected lube oil piping flanges. This fabrication involves cutting and fitting the screen to the gaskets. Screens must be clearly and permanently tagged for ease of identification. Location for screens must be agreed with by vendor and owner. Basically, they should be positioned at all inlets to the machine. Locations immediately after risers must be avoided. Additionally, 100-mesh screens with 60-mesh back up screens will be installed at the main oil return, degassing tank inlet and return and reservoir oil fill connection. These screens will be in place during all flushing operations. A drain should be fitted at lower point of return lines ahead of screen in order to deal with a blocked screen. Also, a pressure device (manometer – i.e: simple length of plastic tubing) is to be installed at this point to monitor any pressure build-up due to blockage.

2.6 The reservoir shall be filled with the lube oil specified for permanent plant operation unless directed otherwise by the owner.

2.7 Lube oil flush should not occur until the compressor skid has been fully grouted.

2.8 The compressor rear bearing port cover and the load coupling guard must be installed in order to minimize any oil spill during flushing.

2.9 The vent piping must be checked out for mechanical completeness.

2.10 Check for correct operation of the auxiliary lube oil pump on/off/auto switch and the oil high temperature alarm before commencing flushing.

2.11 The instruments required for the flushing operation shall be identified on a P&ID mark-up for flushing and will be calibrated for normal operation prior to starting the oil system.

2.12 Add nitrogen bottles or instrument air connections at suitable tapping points downstream of lube and seal oil filters.

2.13 Water to oil coolers shall be provided.

2.14 An auxiliary boiler will be provided to heat the oil to approximately 82°C (180°F).

3. Flushing procedure

3.1 The range of temperatures for the hot oil circulation flush shall be 50°C – 82°C (120°F – 180°F). Before initial circulation the oil should be heated to approximately 50°C (120°F).

3.2 The following parameters shall be documented in the log on an hourly basis:
- Pump discharge pressure
- Bearing header pressure
- Oil reservoir temperature
- Bearing header temperature
- Oil filter differential pressure
- Filter in use
- Sections of piping being flushed
- Start time

3.3 The drain oil sight flow gauges shall be continually monitored for flow at all places. The complete filling of the sight glass indicates a flow blockage. Immediate action shall be taken to stop circulation and clean filter screens. The debris obtained on the screen shall be collected into plastic bags, identified by screen location, machine number and time.

3.4 The following schedule shall be used for flushing:

- Add 100-mesh screen with 60-mesh back-up screens at oil reservoir and degassing tank if applicable as stated in paragraph 2.5 and flush through total system at 15 minute intervals until screens are reasonably clean. Monitor closely the return lines on a continuous basis to ensure system is not backing up with oil.
- Flush through total system at intervals of one hour until screens are reasonably clean.
- Flush total system, alternating through systems section until screens are clean. Supply lines shall be alternated to ensure a greater than 150 percent oil flow is maintained at all times.
- Add 100-mesh screens with 60-mesh back-up screens to lube and seal oil inlet lines to all equipment. Also add 100/60-mesh screens to overhead and seal oil tanks, reference lines and coupling guard feed lines (if furnished).
- Repeat 'flush total systems' above.
- Alternate flushing through each filter/cooler section, control valves and their bypasses, overhead tanks, seal oil traps and reference gas lines. *Record*
  Note: A differential pressure of 100 kPa (15 psig) across either filter indicates the need for filter change.
- Bubble nitrogen through system at regular intervals. *Record*
- Flush through all instrument connections. *Record*
- Flush through all pressure control valve impulse lines. *Record*
- Thermoshock the system by use of the lube oil coolers at regular intervals (varying oil temperature between 50 °C – 82 °C [120°F – 180°F]). *Record*
- Rap exposed piping with a fibre hammer at one hour intervals. *Record*

3.5 When the 100-mesh screens meet criteria in paragraph 3.9/3.10 reinstate all instrumentation, orifices, pipe spools, etc. Arrange entire system in normal, complete configuration with all controls, alarms, etc., in operation.

3.6 Add 100-mesh white cloth (backed with 60-mesh screen) to all lube and seal oil supply points on equipment bearings, seals and control oil inlets. Flush until the criteria as stated in paragraph 3.9/3.10 is achieved, but for a minimum period of 24 hours.
Note: During final flush, alternate flushing through each filter/cooler section, control valves and their bypasses.

3.7 Whenever practical, circulation shall be continued from the time of start-up until completion on a 24 hour per day basis. Any irregularities shall be immediately reported to the vendor representatives and the applicable owner representative as designated, which will be posted on the accessory skid control panel.

3.8 Owner's quality control representative will monitor all operations for compliance and verify all records, test parameters and acceptance criteria on a surveillance basis. Final screen particle count will be verified and recorded by quality control.

3.9 The oil system acceptance criteria which shall be the basis for witness approval parameters for contractor and/or owner shall be as follows: Screen contamination shall be within the particle count limits according to the size of pipe; it will be determined as follows:
20 non-metallic particles on pipe 2.5 to 5 cm (1 to 2 in)
50 non-metallic particles on pipe 5 to 10 cm (2 to 4 in)

3.10 Particles shall not be metallic and shall display random distribution on the screen.

3.11 The acceptance of the system shall be after installation/inspection 100-mesh cloth covered screens, then circulating the lube oil an additional four hours and re-inspecting the screens. The final four hours' flush should be with valves open allowing full flow through the system at operating temperature. Final acceptance requires an oil analysis to determine metal content, viscosity and water content.

3.12 Following acceptance the system shall be restored by the owner for normal operation including the following actions:

- Remove all screens
- Visually inspect overhead seal oil tanks, degassing tanks, drainer modules, if required, wipe clean with lint-free cloth.
- Replace all components. Clean filter cartridges (less than 34.5 kPa [5 psi] pressure drop) can remain subject to owner approval. If new cartridges are fitted, the cleanliness of the system must be rechecked.
- Restore reservoir oil to normal operating level.
- Obtain oil analysis.

System identification ______________________________

Oil type ______________ Req'd circulation temp ___________

Minimum duration req'd ______________________________

Pre-flush checklist	Owner	Vendor
1. System configuration per reference drawings.	________	________
2. Old oil has been removed and reservoir/filter housing wiped clean.	________	________
3. Filters and screens installed for flushing.	________	________
4. All required instrumentation calibrated.	________	________
5. Lube/seal oil filled to capacity.	________	________
6. Bearing housings ready to accept flow.	________	________
7. Initial oil temperature reached.	________	________
8. Approval to initiate oil flush.	________	________

Oil flush acceptance

The oil flush of the subject system has been conducted in accordance with site procedures and accepted.

Date of flush _________ Time started _________ Time completed _________

Inspected and accepted by _________ Date _________

Owner _________

Vendor representative _________

Oil Flush Log

Sheet __of __

System identification ___________ Inspector ___________

Date	Time	Circulating pump No.	Discharge pressure	Bearing/seal header pressure	Bearing/seal header temperature	Reservoir temperature	Filter in circulation	Comments

Best Practice 10.8

Perform auxiliary system functional tests prior to start-up, to confirm proper installation; that site operation is in agreement with vendor requirements, project specifications and to obtain a baseline signature.

The following tests should be performed:

- Lube oil system
- Control oil system (if applicable)
- Seal oil system (if applicable)
- Dry gas seal system (if applicable)
- Governor system (if applicable)

The functional test agendas should be prepared based on vendor factory test requirements and field operating requirements to confirm satisfactory steady state and transient operation.

Lessons Learned

Failure to completely check the functional operation of all critical equipment train auxiliary systems prior to start-up will result in delays and can reduce the reliability of the critical equipment train up to the first scheduled turnaround.

Benchmarks

This best practice has been used since 1984; confirm the proper functional operation of all critical equipment auxiliary systems prior to start-up, and record a detailed record of initial settings and operational characteristics for each auxiliary system. See an example of a baseline auxiliary system record in the supporting information below.

B.P. 10.8. Supporting Material

Functional testing

Having satisfactorily installed and flushed the auxiliary system, all auxiliary equipment should be functionally tested, and all instruments and controls checked for proper setting prior to operation of equipment. A functional test outline and procedure is included at the end of this section. We will highlight the major considerations of the procedure at this time.

It is recommended that the console vendor and/or the equipment purchaser prepare a detailed field functional test procedure and calibration check form. The format of this procedure can follow the factory test procedure if this is acceptable. As a minimum, the auxiliary system, bill of material, and schematic should be thoroughly checked, in order to include the calibration and functional test of each major component in the auxiliary system. That is, components on consoles, and up at unit interfaces. A detailed record should be kept of this functional test procedure – this will help significantly during the operation of the unit. The functional test procedure should be carried out initially without the critical equipment running, and then with the auxiliary system running as closely as possible to design operation conditions. The functional test procedure should first require that all instrumentation be properly calibrated before proceeding. Each specific functional test requirement should then be performed and the results noted. If they do not meet specified limits, testing should stop, and components should be corrected at this point. Each step should be followed thoroughly to ensure each component meets all requirements. It is recommended that the operators assigned to this particular unit should assist in functional testing to familiarize themselves with the operation of the system. In addition, site training courses should be conducted prior to the functional test to familiarize operators with the system's basic functions. This training, again, significantly increases understanding of the equipment and ensures unit reliability.

Satisfactory acceptance of a functional test then ensures that the unit has been designed, manufactured, and installed

correctly such that all system design objectives have been obtained and that equipment reliability is optimized.

One remaining factor to be proven is the successful operation of the system with the critical equipment unit in operation. During initial start-up, it is recommended that the functional test be re-performed with the unit operating. While this advice may seem dangerous, unless the unit operators are assured that the subject system has the ability to totally protect critical equipment while operating, auxiliary equipment will never be tested while the unit is in operation.

Remember, critical equipment is designed for 30 years' life, or greater. The components that comprise the auxiliary system are many, and have characteristics that will change with time. Therefore, their reliability can only be maintained if the systems are totally capable of on-line calibration and functional checks. The functional pre-commissioning procedure should be modified to include an on-line periodic functional checking procedure.

At this point, we can clearly see that the major determination of continued equipment reliability rests with the operation, calibration and maintenance of the equipment. In order to assume maximum continued auxiliary equipment reliability, periodic on-line functional checks and calibrations must be performed. How can this be done? The only way is by convincing plant operations of the safety and reliability of the procedure for on-line checking. This can be reinforced during pre-commissioning, by including operators in functional testing checks and on-site training sessions, to show the function of the system. A site training course modified for the specific equipment would prove immensely valuable in achieving this.

Only by involving unit operators in the pre-start check-ups can it be hoped to establish a field functional checking procedure that will be utilized and followed through. Remember, a pressure switch of less than $400 in cost could cause equipment shutdown that could reduce on-site revenue by around one to two million USD per day. The pressure switch selected could be the best, the highest quality in the world, and be properly installed and set. If its calibration is not periodically checked, it could cause an unnecessary shutdown of equipment and result in this revenue loss.

Functional lube/seal system test procedure outline

Objective:	To confirm proper functional operation of the entire system prior to equipment start-up
Procedure Format:	Detail each test requirement. Specifically note required functions/set points of each component. Record actual functions/set points and *all* modifications made.
Note:	All testing to be performed *without* the unit in operation.

I Preparation
- **A** Confirm proper oil type and reservoir level
- **B** Confirm system flush is approved and *all* flushing screens are removed
- **C** Confirm all system utilities are operational (air, water, steam, electrical)
- **D** Any required temporary nitrogen supplies should be connected
- **E** All instrumentation must be calibrated and control valves properly set
- **F** Entire system must be properly vented

II Test procedure
- **A** Oil reservoir
  1. Confirm proper heater operation
  2. Check reservoir level switch and any other components (TIs, vent blowers, etc.)
- **B** Main pump unit
  1. Acceptable pump and driver vibration
  2. Absence of cavitation
  3. Pump and driver acceptable bearing temperature
  4. Driver governor and safety checks (uncoupled) if driver is a steam turbine
- **C** Auxiliary pump unit
  Same procedure as item B above.
- **D** Relief valve set point and non-chatter check
- **E** Operate main pump unit and confirm all pressures, differential pressures, temperatures and flows are as specified on the system schematic and/or bill of material
- **F** Confirm proper accumulator pre-charge (if applicable)
- **G** Confirm proper set point annunciation and/or action of *all* pressure, differential pressure and temperature switches
- **H** Switch transfer valves from bank 'A' to bank 'B' and confirm pressure fluctuation does not actuate any switches
- **I** Trip main pump and confirm auxiliary pump starts without actuation of any trip valves or valve instability
  Note: Pressure spike should be a minimum of 30% above any trip settings
- **J** Repeat step I above but slowly reduce main pump speed (if steam turbine) and confirm proper operation
- **K** Simulate maximum control oil transient flow requirement (if applicable) and confirm auxiliary pump does not start
- **L** Start auxiliary pump, with main pump operating and confirm control valve and/or relief valve stability
  Note: Some systems are designed to *not* lift relief valves during two pump operation

III Corrective action
- **A** Failure to meet any requirement in Section II requires corrective action and retest
- **B** Specifically note corrective action
- **C** Sign-off procedure as acceptable to operate

Electro-hydraulic governor functional test procedure outline

Objective:	To confirm proper system functional operation prior to equipment start-up.
Procedure Format:	Detail each test requirement. Specifically note required functions/set points. Record actual functions/set points and all modifications made. (Note: All testing to be performed *without* the unit in operation)

I Preparation
- Confirm all shut down contracts are in the normal condition
- Confirm all power supplies are on
- Secure necessary test equipment
- Pressure sources (nitrogen bottles) for pressure simulation at transmitters
- Frequency generators for simulating speed signals

II Test procedure
- Take required action to put system in 'run' mode
- Open trip and throttle valve *only after* ensuring the main steam block valve is closed
- Simulate turbine start, slow roll and any start sequence 'hold' points up to minimum governor operating point
- Confirm proper operation of 'raise' and 'lower' speed buttons
- Connect external process signal inputs (one at a time) and confirm proper governor action to input signal variation
- Check overspeed override feature
- Confirm automatic transfer to and from backup governor 'position control' for each of the following cases:
  - Loss of main governor power supply
  - Zero external input signal
  - Failure of 'final driver' (internal governor component)
  - Zero speed inputs
- Confirm manual transfer to and from backup governor and 'emergency override'
- Check raise and lower speed controls while in backup governor mode
- Confirm governor shutdown (trip) operation under the following conditions:
  - Overspeed setting
  - Failure of both main and backup governor controls

III Corrective action
- Failure to meet any requirement in Section II requires corrective action and retest
- Specifically note corrective action
- Sign off procedure as acceptable to operate

Steam turbine solo run functional test procedure outline

Objective:	To confirm acceptable mechanical operation of the steam turbine, governor system and safety (trip) system
Procedure Format:	Detail each test requirement. Specifically note required test limits (Note: shop test data should be used to define acceptable limits). Record actual test values using appropriate instrumentation and note all modifications made.

I Preparation
- Confirm all auxiliary system tests are complete (governor, lube system, etc.)
- Confirm all inlet steam lines have been cleaned and signed off
- Confirm all installed instrumentation is calibrated
- Secure *all* required instruments
- Calibrated pressure and temperature gauges
- Occiloscope(s)
- Vector filters
- Amplifier(s)
- Spectrum analyzer
- Tape recorder or information gathering module
- 'Walk' all steam inlet, extraction and exhaust lines. Confirm *all* spring hangers are released (unlocked) and safety valves are installed

II Test procedure
- Confirm *all* auxiliary systems are operational and at proper conditions (lined out)
- Lube/control oil system
- Governor/trip system
- Turning gear (if applicable)
- All warming lines drained and operational
- Condensing system including condensate pumps (if applicable)
- Extraction system (if applicable)
- Steam seal system
- Condition monitoring systems (vibration, temperature, etc.)
- Steam conditions within allowable vendor limits
- Slow roll and start unit per vendor's instructions. (refer to cold start-up speed vs. time chart)
- Demonstrate manual trip (panic button) at low speed (500 rpm)
- Reset trip, accelerate back to desired speed – listen for rubs, etc.
- Gradually increase speed to next speed step. Record the following data for each vendor required speed step up to minimum governor speed
- Overall vibration at each vibration point (record frequency if specified limits are exceeded)
- Bearing oil temperature rise at each bearing
- Bearing pad temperature (axial and radial) at each point (if applicable)
- Turbine speed
- Axial shaft displacement
- Turbine exhaust temperature
  Note: Use shop test data for comparison
- After confirming stable operation at minimum governor speed, accelerate carefully to overspeed trip setting and trip the turbine three times. Each trip speed should fall within the vendor's trip speed set point allowable range
- Return to minimum governor speed and confirm satisfactory manual and automatic speed control. Also confirm automatic transfer from main to backup governor
- Connect vibration recording instruments, reduce turbine speed to 500 rpm and record shaft vibration (at each vibration monitoring point) and phase angle while gradually increasing speed to maximum continuous speed. Repeat step in reverse direction (maximum continuous speed to 500 rpm)
- Note: This data will be reduced to Bode, Nyquist and Cascade plots and should be compared to shop test data.
- Increase turbine speed to maximum continuous speed and run for four (4) hours or until bearing temperatures stabilize
- Finally trip the turbine using a system trip switch (simulate low oil pressure, etc.)

III Corrective action
- Failure to meet any requirement in Section II requires corrective action and retest
- Specifically note any corrective action
- Sign off equipment as acceptable to operate
- Typical initial field functional lube/seal system test procedure baseline document

Table 10.8.1 Lube/seal system test procedure

Item:		
Reference DWGS:	Turbine utility	P&ID _____
	Compressor utility	P&ID _____
Purpose:	To fully prove functional operation of entire lube/seal system, including all permissive, alarm and shutdown functions prior to initial operation of the unit.	
Note:	All testing to be performed without the unit in operation. When specified values are not satisfied correct and retest	
Preparation:	Prior to testing of the system, confirm and sign off that the following has been checked:	
	■ Oil reservoir at proper level ■ Specified oil is used ■ Oil heater in operation ■ Oil cooler water supply on ■ System clean (all test screens out) ■ Instrument air in operation at all instruments ■ Temporary N_2 supply connected ■ All instrumentation noted on attached list has been calibrated to specified values ■ Steam lines to console blown ■ Entire system vented	

Table 10.8.2 Lube system check list (example)

Item	Description	Specified value	Actual value	Witnessed by
1.	Record reservoir temp. rise in four (4) hours with heater on. Read on T.I.	Record actual ΔT	______	
2.	Check reservoir level switch setting. Read on Annun.	High level 14 " from top. low level 45" from top	High ______ Low ______	
3.	Energize aux. lube pump check:			
3A.	Pump vibration	.2 in/sec	______	
3B.	Motor vibration	.2 in/sec	______	
3C.	Cavitation	None	______	
3D.	Pump/motor brg. Temp	165°F	______	
4. (see note 1)	Block in aux. pump using pump discharge valve-set relief valve. Valve chatter is not acceptable.	3130 Kpa	______	
5.	Confirm the 'Aux. pump running' Annun. is actuated by switch by shutting off pump and restarting while reading pressure on PI	690 Kpa rising	_____	
6.	Allow system to heat up to 49°C downstream of coolers and adjust the following items (if required) to attain specified values:			
6A.	Back press regulator Read value on P.I.	2262 Kpa	_____	
6B.	Lube oil supply valve PCV Read value on P.I.	124–138 Kpa	_____	
6C.	Control oil valve PCV Read value on P.I.	883 Kpa	_____	

(Continued)

Table 10.8.2 Lube system check list (example)—Cont'd

Item	Description	Specified value	Actual value	Witnessed by
6D.	L.P. case seal oil differential valve *PDCV*. Supply N_2 press, of *5PSIG* at gas reference side of *PDT* and read differential press on PDI	241 Kpa	_____	
6E.	HP case seal oil differential valve. Supply N_2 pressure of *30 PSIG* at gas reference side of PDT and read differential press on PDI	241 Kpa		
7.	Record the following:			
7A.	Pump disch. press on P.I.	Approx. 2600	_____	
7B.	Oil temp. upstream on T.I.	Less than 65 °C	_____	
7C.	Oil temp. downstream on T.I.	49°		
7D.	Cooler/filter ΔP on PDI	Less than		
	Switch transfer valve and record:	70 Kpa	_____	
	Bank ‘A’ ΔP		_____	
	Bank ‘B’ ΔP			
7E.	Lube press at console on P.I.	*Approx. 283 Kpa	_____	
7F.	Control press at console on P.I.	*1091 Kpa	_____	
7G.	LP case on ΔP PDI	241 Kpad	_____	
7H.	HP case on ΔP PDI	241 Kpad	_____	
7I.	Lube press at unit on P.I.	124 to 138 Kpa	_____	
7J.	Control press at unit P.I.	883 Kpa	_____	
7K.	All ‘sight’ glasses show oil flow		_____	
7L.	Lube oil head tank is full	–	_____	
7M.	Turbine accumulator press on P.I.	*Approx. 900 Kpa	_____	
	*Adjust as required to attain proper values at the unit			
8.	Record seal oil drainer level for each drainer for one hour.			
	Drainer A	2 fills per hour	_____	
	Drainer B		_____	
	Drainer C		_____	
	Drainer D		_____	
9.	Switch transfer valve from ‘A’ to ‘B’ bank and observe press fluctuation on P.I. and confirm that PSL does not actuate.		_____	
10.	Bleed low side of filter switch PD SH and confirm PDAH actuates at specified value.	241 Kpa Rising	_____	
11.	Bleed pressure off PSL (low oil press) and confirm @ unit annun. *PAL* actuates at specified value. Read P.I.	90 Kpa Falling		
12.	Increase temporary N_2 press on *PDSL* (L.P. case seal low ΔP alarm switch) and confirm Annun. *PDAL* actuates at specified value. Read PDI	207 Kpa Falling	_____	
13. (see note 2)	Repeat above for *PDSL* (H.P. case seal low ΔP alarm) confirm Annun. *PDAL* actuates read on PDI	Falling	_____	
14.	Bleed pressure off PSL (low oil trip switch) and observe:	76 Kpa Falling	______	
	A. Annun. PALL functions Read PI			
	B. T&T valve closes			
	C. Valve rack closes			
15. (see Note 2)	Increase temporary N_2 press on *PDSLL* (L.P. case seal low ΔP trip switch) and observe that:			
	A. Annun. *PDALL* functions. Read on PDI	138 Kpa Falling	______	
	B. Action occurs as in 14 above			

Table 10.8.2 Lube system check list (example)—Cont'd

Item	Description	Specified value	Actual value	Witnessed by
16.	Repeat above for *PDSLL* (H.P. case seal low ΔP trip) and observe Annun. PDALL functions at specified value. Read PDI	138 Kpa Falling	 ______	
17.	Shut off aux. oil pump and confirm all valves are stable		______	
18.	Time rundown of oil tank. Observe level switch *LSL* actuates Annun. *LAL*	Approx. 4" above Tang. Line	______	
19.	A. Disconnect main lube oil pump from turbine. B. Drain steam inlet line to main lube pump turbine. C. Drain turbine. D. Confirm turbine bearing cooling water is on. E. Confirm trip is reset.			
20.	Open inlet valve, gradually bring turbine up to speed and confirm the following: A. Rated speed (Strobe Tac) B. Inlet press P.I. C. Inlet temp. T.I. *Temp* D. Exhaust press P.I. *Temp* E. Vibration (using metrix vibration instrument)	3600 RPM 2274 Kpa 327 °C 517 Kpa .2 in/sec.	______ ______ ______ ______	
21.	Disable governor and check overspeed trip 3 times.	 4140 RPM	 ______	
22.	Manually trip turbine using hand trip.			
23.	Check pump/turbine alignment and couple up main pump.			
24.	Slowly bring pump up to speed and check: 24A. Pump vibration 24B. Cavitation 24C. Pump brg. temp.	 .2 in/sec None 165°F	 ______ ______ ______	
25.	Adjust governor so all valves are as noted in step 7. Record speed RPM.		 ______	
26.	Block in pump using pump discharge valve. Set relief valve *PSV*. Valve chatter is not acceptable.		______	
27.	With turbine operating at speed noted in step 25 and disch. block valve open, manually trip turbine and observe: A. Aux. pump starts. B. Min. press spike on P.I. lube P.I. Control PDI LP Case ΔP seal PDI HP Case ΔP seal C. Alarm and trip switches connected with lube & seal oil are not actuated. D. All valves are stable.	 90 Kpa T&T valve does not close 207 Kpa 207 Kpa	 ______ ______ ______ ______ ______	
28.	Restart pump turbine and dump control oil pressure using hand valve at turbine. Observe that no alarm or trip lights are actuated and that all valves remain stable.		______	
29.	Start aux. oil pump with turbine operating and observe: A. RVs do not lift B. All control valves are stable C. Oil pump turbine does not hunt (speed remains stable)		 ______ ______	

(Continued)

Table 10.8.2 Lube system check list (example)—Cont'd

Item	Description	Specified value	Actual value	Witnessed by
30.	Stop aux. pump motor and observe: A. All control valves remain stable. B. Min. press. spike on P.I. lube P.I. Control PDI LP Case ΔP PDI LP Case ΔP	 90 Kpa T&T valve does not trip 207 Kpa 207 Kpa	 _____ _____ _____ _____ _____	
31.	With turbine operating. Bleed press, from PSL and observe aux. pump starts and that all valves remain stable.		_____	
32.	Having satisfactorily completed all items above, secure both aux. and main pump and sign off as being acceptable for operation.		_____	
	NOTE: At this point, elect one driver and continue to operate the console 24 hours a day.			
NOTE 1	RVs will continuously pass a small stream of oil. Actual setting will be that pressure at which stream volume increases. Observe by unbolting FLGS at reservoir. Accumulation value is with pump discharge block fully closed.			
NOTE 2	Block out gas signal to diff. control valve during this step.			

Best Practice 10.9

Perform critical compressor air or nitrogen runs during pre-commissioning to confirm that all phases of construction have been successfully completed.

The following phases should be checked:

- Preservation
- Foundation preparation
- Grouting
- Soft foot check
- Process pipe connection
- Final alignment
- Functional auxiliary system checks

The air or nitrogen run will ensure that the mechanical behavior of the critical machinery train matches that of the FAT (factory acceptance test) mechanical test, thus confirming that all construction issues were properly addressed.

Lessons Learned

Failure to include an air or inert gas run in pre-commissioning exposes the plant to reduced safety and reliability.

Whenever the project team was able to justify an air or inert gas run prior to plant start-up, items were identified that would have impacted the safety and reliability of the critical equipment train up to the first turnaround.

Benchmarks

This best practice has been recommended since 1985 to achieve plant start-ups with minimum delays and optimum plant safety and reliability.

B.P. 10.9. Supporting Material

The site air or inert gas run procedure should be developed along with the machinery vendor to ensure that all operational differences (power requirements, dry gas seal conditions, pressures and temperatures, etc.) are considered and planned for.

The test objective is to completely confirm the mechanical operation of the machinery and the associated auxiliary systems prior to plant operation.

Process piping may have to be disconnected for these runs to prevent overheating. In some applications, temporary temperature measuring devices may have to be installed to curtail testing when casing temperatures exceed design limits (approximately 200 °C [400°F]).

Best Practice 10.10

Perform driver solo runs to confirm that all the phases of construction have been completed successfully, and that the control and protection systems operate properly.

These runs will also confirm:

- Proper functional operation of all auxiliaries
- Overspeed protection system check
- Accumulator precharge check*
- Trip valve manual exercise check*
- Governor system check

* = where applicable

In addition to ensuring proper operation of the driver and associated auxiliary systems, the solo runs are an excellent training opportunity for site personnel.

Awareness of the function and design basis of the various components of the driver and support systems by site personnel will result in a greater understanding of condition monitoring requirements and increase driver safety and reliability.

Lessons Learned

Failure to completely check out driver operation prior to plant start-up will result in start-up delays and expose the plant to lower machine reliability until the first scheduled turnaround.

In the rush to complete commissioning activities and start plant production, pre-start-up checks are often minimized. Any critical (unspared) machinery reliability issues that are not identified prior to plant operation will expose the plant to corresponding safety, reliability and revenue losses until they can be corrected (usually during the first plant turnaround four or five years after start-up).

Benchmarks

This best practice has been used since 1984 to identify and correct critical machinery driver potential safety and reliability issues prior to plant start-up to optimize critical machinery train reliability (> 99.7%).

B.P. 10.10. Supporting Material

Electro-hydraulic governor functional test procedure outline

Objective:	To confirm proper system functional operation prior to equipment start-up.
Procedure Format:	Detail each test requirement. Specifically note required functions/set points. Record actual functions/set points and all modifications made. Note: All testing to be performed *without* the unit in operation

I Preparation
- Confirm all shut down contracts are in the normal condition
- Confirm all power supplies are on
- Secure necessary test equipment:
  - Pressure sources (nitrogen bottles) for pressure simulation at transmitters
  - Frequency generators for simulating speed signals

II Test procedure
- Take required action to put system in 'run' mode
- Open trip and throttle valve *only after* insuring the main steam block valve is closed
- Simulate turbine start, slow roll and any start sequence 'hold' points up to minimum governor operating point
- Confirm proper operation of 'raise' and 'lower' speed buttons
- Connect external process signal inputs (one at a time) and confirm proper governor action to input signal variation
- Check overspeed override feature
- Confirm automatic transfer to and from backup governor 'position control' for each of the following cases:
  - Loss of main governor power supply
  - Zero external input signal
  - Failure of 'final driver' (internal governor component)
  - Zero speed inputs
- Confirm manual transfer to and from backup governor and 'emergency override'
- Check raise and lower speed controls while in backup governor mode
- Confirm governor shutdown (trip) operation under the following conditions:
  - Overspeed setting
  - Failure of both main and backup governor controls

III Corrective action
- Failure to meet any requirement in Section II requires corrective action and retest
- Specifically note corrective action
- Sign off procedure as acceptable to operate

Steam turbine solo run functional test procedure outline

Objective:	To confirm acceptable mechanical operation of the steam turbine, governor system and safety (trip) system
Procedure Format:	Detail each test requirement. Specifically note required test limits (Note: shop test data should be used to define acceptable limits). Record actual test values using appropriate instrumentation and note all modifications made.

I Preparation
- Confirm all auxiliary system tests are complete (governor, lube system, etc.)
- Confirm all inlet steam lines have been cleaned and signed off
- Confirm all installed instrumentation is calibrated
- Secure *all* required instruments:
  - Calibrated pressure and temperature gauges
  - Oscilloscope(s)

- Vector filters
- Amplifier(s)
- Spectrum analyzer
- Tape recorder or information gathering module

■ 'Walk' all steam inlet, extraction and exhaust lines. Confirm *all* spring hangers are released (unlocked) and safety valves are installed

II Test procedure

■ Confirm *all* auxiliary systems are operational and at proper conditions (lined out):
- Lube/control oil system
- Governor/trip system
- Turning gear (if applicable)
- All warming lines drained and operational
- Condensing system including condensate pumps (if applicable)
- Extraction system (if applicable)
- Steam seal system
- Condition monitoring systems (vibration, temperature, etc.)
- Steam conditions within allowable vendor limits

■ Slow roll and start unit per vendors instructions. (Refer to cold start-up speed vs. time chart)

■ Demonstrate manual trip (panic button) at low speed (500 rpm)

■ Reset trip, accelerate back to desired speed – listen for rubs, etc.

■ Gradually increase speed to next speed step. Record the following data for each vendor required speed step up to minimum governor speed:
- Overall vibration at each vibration point (record frequency if specified limits are exceeded)
- Bearing oil temperature rise at each bearing
- Bearing pad temperature (axial and radial) at each point (if applicable)
- Turbine speed
- Axial shaft displacement
- Turbine exhaust temperature

  Note: Use shop test data for comparison

■ After confirming stable operation at minimum governor speed, accelerate carefully to overspeed trip setting and trip the turbine three times. Each trip speed should fall within the vendors' trip speed set point allowable range

■ Return to minimum governor speed and confirm satisfactory manual and automatic speed control. Also confirm automatic transfer from main to backup governor

■ Connect vibration recording instruments, reduce turbine speed to 500 rpm and record shaft vibration (at each vibration monitoring point) and phase angle while gradually increasing speed to maximum continuous speed. Repeat step in reverse direction (maximum continuous speed to 500 rpm)

■ Note: This data will be reduced to Bode, Nyquist and Cascade plots and should be compared to shop test data.

■ Increase turbine speed to maximum continuous speed and run for four (4) hours or until bearing temperatures stabilize

■ Finally trip the turbine using a system trip switch (simulate low oil pressure, etc.)

III Corrective action

■ Failure to meet any requirement in Section II requires corrective action and retest

■ Specifically note any corrective action

■ Sign off equipment as acceptable to operate

Best Practice 10.11

Obtain a CCM (component condition monitoring) initial start-up baseline for critical machinery trains.

This is done by obtaining information concerning the condition of the following train components:

- Rotor (performance information)
- Journal bearings
- Thrust bearings
- Seals
- Auxiliary systems

Machinery condition monitoring using the CCM approach ensures total monitoring of each major component in the machinery train (see the supporting information below for complete CCM details).

We recommend that all parameters for each monitored component be obtained from the plant process information system and downloaded to Excel spreadsheets. This will allow simultaneous trending of each component parameter to enable rapid interpretation of a condition change in any component for prompt corrective action. We have included examples of CCM Excel spreadsheets in the supporting material for this best practice.

Lessons Learned

Failure to obtain detailed component baseline conditions for critical machinery trains will lead to extended periods taken up with obtaining additional information to determine the root causes of machinery component parameter changes, with the results being reduced machinery safety and reliability and corresponding revenue losses.

Many site root cause analyses are inconclusive and need to obtain additional 'baseline' information after failed or worn components are replaced. If 'baseline' conditions at initial start-up had been obtained, root causes could have been immediately determined.

Benchmarks

This best practice has been recommended since the late 1980s for all new projects and during all site machinery reliability audits to enable site machinery specialists to immediately identify component parameter abnormal conditions. This information can then be discussed with operations to plan for appropriate process changes while attempting to extend any required maintenance to a planned shutdown.

B.P. 10.11. Supporting Material

The major machinery components

Think of all the machinery that you have been associated with and ask, 'What are the major components and systems that are common to all types of rotating equipment?'

- Pumps
- Steam turbines
- Compressors
- Motors
- Gas turbines
- Fans, etc.

Figure 10.11.1 presents the major component classifications for any type of machinery.

Regardless of the type of machinery, monitor these components and you will know the total condition of the machine.

Component condition monitoring

As previously stated, component and system functions must first be defined and the normal values for each component listed. These facts are presented in Figure 10.11.2.

Once the function of each component is defined, each major machinery component can be monitored as shown in Figure 10.11.3.

- Rotor
- Radial bearing
- Thrust bearing
- Seal
- Auxiliary systems

Figure 10.11.1 • Major machinery components and systems

- Define the function of each affected component
- Define the system in which each affected component operates
- List the normal parameters for each affected component and system component

Fig 10.11.2 • Component and system functions

- Define each major component
- List condition monitoring parameters
- Obtain baseline data
- Trend data
- Establish threshold limits

Fig 10.11.3 • Component condition monitoring

If you don't know where you started, you do not know where you are going!

Fig 10.11.4 • Baseline condition

Baseline

Having defined all condition parameters that must be monitored, the next step in a condition monitoring exercise is to obtain baseline information. It is important to obtain this information as soon as is physically possible after start-up of equipment. However, operations should be consulted to confirm when the unit is operating at rated or lined-out conditions. Obtaining baseline information without conferring with operations is not suggested since faulty information could be obtained, which would lead to erroneous conclusions being drawn for predictive maintenance. Figure 10.11.4 states the basics of a baseline condition.

It amazes us how many times baseline conditions are ignored. Please remember Figure 10.11.4 and make it a practice to obtain baseline conditions as soon as possible after start-up.

Trending

Trending is simply the practice of monitoring a parameter condition with time. Trending begins with baseline condition and will continue until equipment shutdown. In modern day thought, it is often conjectured that trending must be performed by micro-processors and sophisticated control systems. This is not necessary! Effective trending can be obtained by periodic, manual observation of equipment, or using equipment available to us in the plant which will include DCS systems, etc. The important fact is to obtain the baseline and trends of data on a periodic basis. When trending data, threshold points should also be defined for each parameter that is trended. This means that when the parameter pre-established value is exceeded, action must be taken regarding problem analysis. Setting threshold values at a standard percentage above normal value is recommended. Typically values are on the order of 25–30% above baseline values. However, these values must be defined for each component based on experience. Figure 10.11.5 presents trending data for a hydrodynamic journal bearing. All of the parameters noted in Figure 10.11.5 should be monitored to define the condition of this journal bearing.

Specific machinery component and system monitoring parameters and their limits

The following sections contain information concerning what parameters should be monitored for each major machinery component to determine its condition. In addition, typical limits

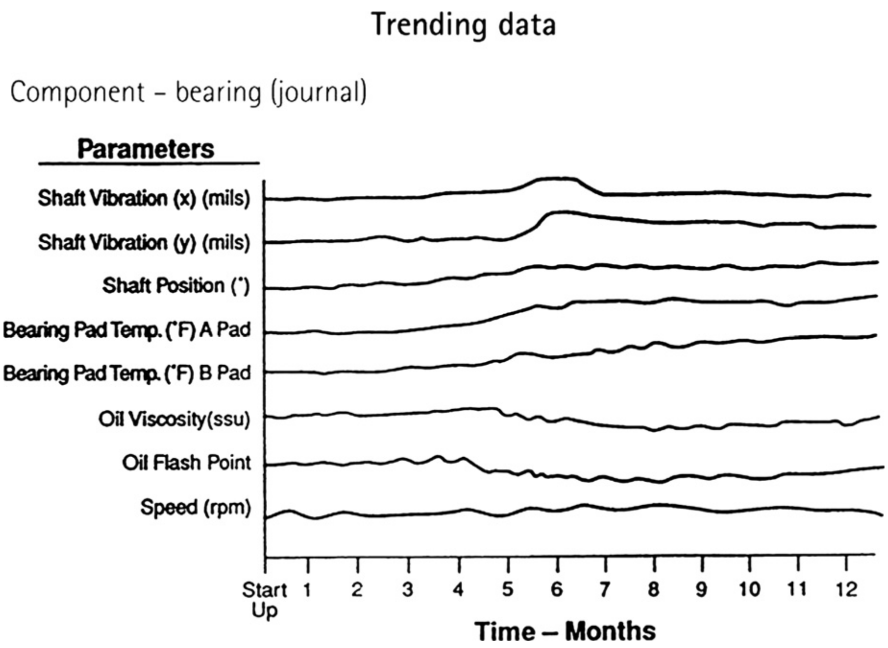


Figure 10.11.5 • Trending data

are noted. These limits represent the approximate point at which action should be planned for maintenance. They are not intended to define shutdown values.

The rotor

Rotor condition defines the performance condition (energy and efficiency) of the machine. Figure 10.11.6 presents this value for a pump.

Radial bearings

Figures 10.11.7 and 10.11.8 present the facts concerning anti-friction and hydrodynamic (sleeve) radial or journal bearing condition monitoring.

1. Take value at minimum flow (shut off discharge valve)
2. Measure:
   - P_1 ■ Driver bhp
   - P_2 ■ Specific gravity

Where: P_1 and P_2 = psig, bhp = brake horsepower.

1. Calculate:

A. head produced

$$\frac{m-kgf}{kgM} = \frac{\Delta P \text{ x}.102}{S.G.}\left(\frac{ft-lb_f}{lbm} = \frac{\Delta P \text{ x } 2.311}{S.G.}\right)$$

B. pump efficiency (%)

$$= \frac{hd \text{ x } m^3/hr \text{ x } \rho \text{ x } g}{3.6 \text{ x } 10^6 \text{ x } kW}\left(\frac{hd \text{ x } gpm \text{ x } S.G.}{3960 \text{ x } bhp}\right)$$

2. Compare to previous value; if > −10% perform maintenance

Fig 10.11.6 • Pump performance monitoring

Journal bearing (anti-friction)

Parameter	Limits
1. Bearing housing vibration (peak)	0.4 inch/sec (10 mm/sec)
2. Bearing housing temperature	180°F (85°C)
3. Lube oil viscosity	off spec 50%
4. Lube oil particle size	
• non metallic	25 microns
• metallic	any magnetic particle in the sump
5. Lube oil water content	below 200 ppm

Fig 10.11.7 • Condition monitoring parameters and their alarm limits

Journal bearing (hydrodynamic)

Parameter	Limits
1. Radial vibration (peak to peak)	2.5 mils (60 microns)
2. Bearing pad temperature	220°F (108°C)
3. Radial shaft position*	> 30° change and/or 30% position change
4. Lube oil supply temperature	140°F (60°C)
5. Lube oil drain temperature	190°F (90°C)
6. Lube oil viscosity	off spec 50%
7. Lube oil particle size	> 25 microns
8. Lube oil water content	below 200 ppm

* Except for gearboxes where greater values are normal from unloaded to loaded

Fig 10.11.8 • Condition monitoring parameters and their alarm limits

Thrust bearings

Figures 10.11.9 and 10.11.10 show condition parameters and their limits for anti-friction and hydrodynamic thrust bearings.

Seals

Figure 10.11.11 presents condition parameters and their limits for a pump liquid mechanical seal.

Auxiliary systems

Condition monitoring parameters and their alarm limits are defined in Figures 10.11.12 and 10.11.13 for lube and pump flush systems.

Figures 10.11.14, 10.11.15 and 10.11.16 present condition monitoring parameters and limits for dynamic compressor performance, liquid seals and seal oil systems.

Thrust bearing (anti-friction)

Parameter	Limits
1. Bearing housing vibration (peak)	
• radial	0.4 in/sec (10 mm/sec)
• axial	0.3 in/sec (1 mm/sec)
2. Bearing housing temperature	185°F (85°C)
3. Lube oil viscosity	off spec 50%
4. Lube oil particle size	
• non metallic	> 25 microns
• metallic	any magnetic particles with sump
5. Lube oil water content	below 200 ppm

Fig 10.11.9 • Condition monitoring parameters and their alarm limits

Thrust bearing (hydrodynamic)

Parameter	Limits
1. Axial displacement*	> 15–20 mils (0.4–0.5 mm)
2. Thrust pad temperature	220°F (105°C)
3. Lube oil supply temperature	140°F (60°C)
4. Lube oil drain temperature	190°F (90°C)
5. Lube oil viscosity	off spec 50%
6. Lube oil particle size	> 25 microns
7. Lube oil water content	below 200 ppm

*and thrust pad temperatures > 220°F (105°C)

Fig 10.11.10 • Condition monitoring parameters and their alarm limits

Pump liquid mechanical seal

Parameter	Limits
1. Stuffing box pressure	< 25 psig (175 kpa)**
2. Stuffing box temperature	Below boiling temperature for process liquid
3. Flush line temperature	+/− 20°F (10°C) from pump case temp
4. *Primary seal vent pressure (before orifice)	> 10 psi (70 kpag)

*On tandem seal arrangements only
**Typical limit – there are exceptions (Sundyne Pumps)

Fig 10.11.11 • Condition monitoring parameters and their alarm limits

Lube oil systems

Parameters	Limits
1. Oil viscosity	off spec 50%
2. Lube oil water content	below 200 ppm
3. Auxiliary oil pump operating yes/no	operating
4. Bypass valve position (P.D.pumps)	change > 20%
5. Temperature control valve position	Closed, supply temperature > 130 55°C)
6. Filter ΔP	> 25 psid (170 kpag)
7. Lube oil supply valve position	change > +/−20%

Fig 10.11.12 • Condition monitoring parameters and their alarm limits

Pump sealflush (single seal, flush from discharge)

Parameter	Limits
1. Flush line temperature	+/−20°F (+/−10°C) of pump case temperature
2. Seal chamber pressure	< 25 psi (175 kpa) above suction pressure

Fig 10.11.13 • Condition monitoring parameters and their alarm limits

1. Calibrated: pressure and temperature gauges and flow meter
2. Know gas analysis and calculate k, z, m.w
3. Perform as close to rated speed and flow as possible
4. Relationships:

$$\text{A. } \frac{N-1}{N} = \frac{LN\frac{(T_2)}{(T_1)}}{LN\frac{(K_2)}{(K_1)}} \qquad \text{B: EFFICIENCY}_{poly} = \frac{\frac{k-1}{k}}{\frac{n-1}{n}}$$

$$\text{C. HEAD}_{poly} = \left(\frac{m-kgf}{kgM}\right) = \frac{847.4}{MW} \times T1 \times \frac{n}{n-1} \times Z_{avg}\left(\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1\right)$$

$$\left(\frac{Ft-lb_f}{Lb_m}\right) = \frac{1545}{MW} \times T1 \times \frac{n}{n-1} \times Z_{avg}\left(\left(\frac{p2}{p1}\right)^{\frac{n-1}{n}} - 1\right)$$

5. Compare to previous value, if decreasing trend exists greater than 10%,inspect at first opportunity

Fig 10.11.14 • Compressor performance condition monitoring

Compressor liquid seal

Parameter	Limits
1. Gas side seal oil/gas ΔP	
• bushing	< 12 ft. (3.5m)
• mechanical contact	< 20 psi (140 kpa)
2. Atmospheric bushing oil drain temperature	200°F (95°C)
3. Seal oil valve* position	> 25% position change
4. Gas side seal oil leakage	> 20 gpd per seal

*supply valve = + 25%

return valve = − 25%

Note this assumes compressor reference gas pressure stays constant

Fig 10.11.15 • Condition monitoring parameters and their alarm limits

Compressor liquid seal oil systems

Parameters	Limits
1. Oil viscosity	off spec 50%
2. Oil flash point	below 200°F (100°C)
3. Auxiliary oil pump operating yes/no	operating
4. Bypass valve position (P.D. pumps)	change > 20%
5. Temperature control valve position	closed, supply temperature 130°F (55°C)
6. Filter ΔP	25 psid (170 kpag)
7. Seal oil valve position	change > 20% open (supply) > 20% closed (return)
8. Seal oil drainer condition	(proper operation)
9. Constant level (yes/no)	level should be observed
10. Observed level (yes/no)	level should not be constant
11. Time between drains	approximately 1 hour (depends on drainer volume)

Fig 10.11.16 • Condition monitoring parameters and their alarm limits

Best Practice 10.12

Conduct mini information sessions for operators on new equipment monitoring requirements, and any reliability issues that produce a ‘bad actor’ (more than one machinery failure per year).

Identify any significant reliability issues that have occurred during start-up and/or plant operation.

Team up with the appropriate disciplines to develop a mini (1 hour) information PowerPoint presentation that will present the design and operation principles connected with the reliability issue.

Present these mini information sessions to each shift of plant operators to ensure functional awareness of the affected components and of corrective measures to prevent recurrence of the problem.

Lessons Learned

Lack of operator awareness of machinery design basis leads to reduced machinery safety and reliability. Short, practical and informative ‘function awareness’ sessions are appreciated by operations and result in increased machinery safety and reliability.

Benchmarks

This best practice has been used since the mid-1990s to increase operational awareness of machinery component function, which has resulted in optimum safety and reliability of machinery (reliability > 99.7%).

B.P. 10.12. Supporting Material

Contained in this section are two examples of mini information sessions that were presented to operations in a refinery after these issues had caused reduced machinery safety and reliability.

Typical Refinery Mini Information Session –

Hydraulic Disturbances

Presented by

Bill Forsthoffer

October 11,1998

Fig 10.12.1 • ■

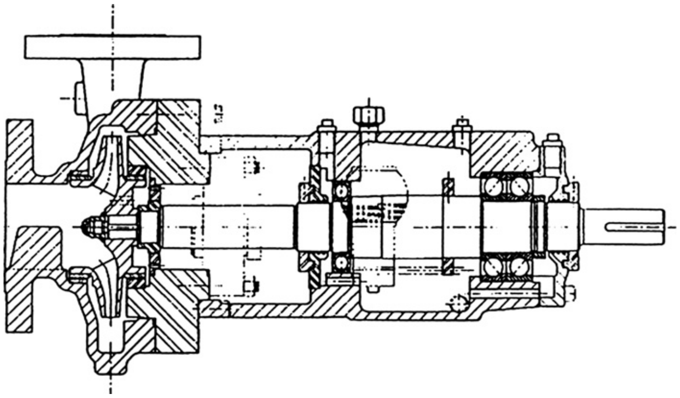


Fig 10.12.2 • ■

- The Heart of the Pump Curve
- What causes a change in the flow rate?
- What is Cavitation?
- NPSH?
- What on earth is Recalculation?

Fig 10.12.3 • Session topics

- Please refer to Figure 10.12.5. If the operator can keep from messing the pump, it will be reliable!
- Guess what? It's not him but the process which cannot always be controlled!
- If discharge pressure increases (closed valve), suction pressure decreases(a plugged screen) or S.G. changes, so does flow!

Fig 10.12.4 • The heart of the curve

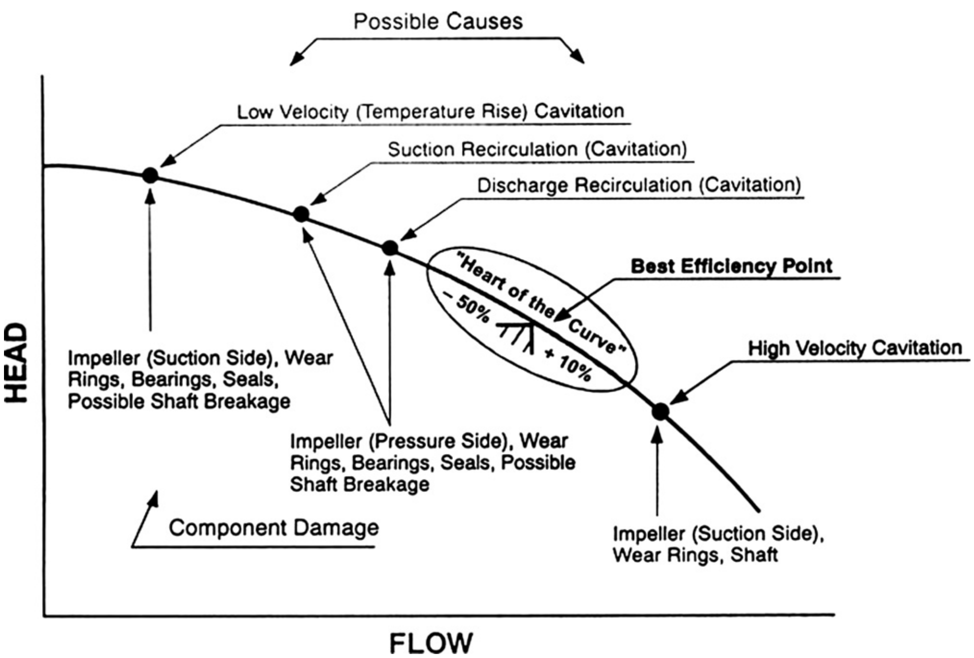


Fig 10.12.5 • ■

- A change in:
- Discharge Pressure
- Suction Pressure
- Specific Gravity
- What can happen in the process to cause these changes?
- Change flow and pump damage is possible!

Fig 10.12.6 • What causes a change in pump flow?

- Cavitation is possible only if a liquid vaporizes?
- What can cause a liquid to vaporize? (Fig.)
- After a liquid vaporizes, it will change back to a liquid when ? (look at Fig 10.12.11.)
- This is cavitation, it sounds like rocks and can cause much damage if water is the liquid. Oils act like shock absorbers.

Fig 10.12.7 • What causes cavitaiton and what is cavitation?

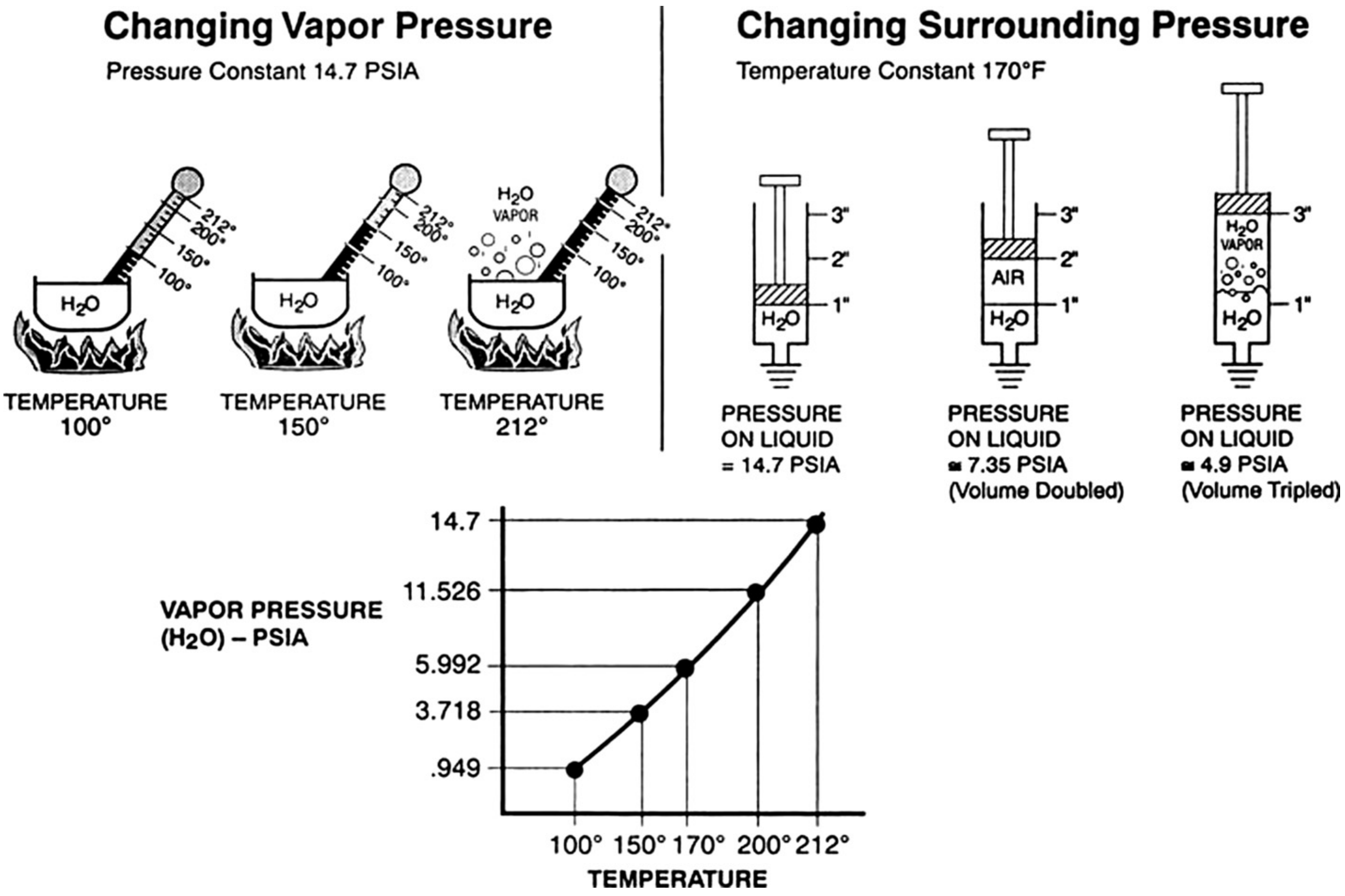


Fig 10.12.8 • ■

- A very confusing term indeed (see Fig 10.12.11.)
- If the pressure drop in the suction part of the pump reduces the liquid pressure below it's suction pressure, what happens?
- To predict if-----------will occur, NPSHa is used to describe how much above the vapor pressure is in feet (pressure). NPSHr is the pressure drop inside the pump suct in feet

Fig 10.12.9 • NPSH anyone?

- Therefore if the pressure drop (NPSHr) is greater than the pressure above the liquid's vapor pressure at the suction flange of the pump (NPSHa), we will have--------------?
- That's why if NPSHr (Net Positive Suction Head required) is > than NPSHa, it's bad!!
- What can cause cavitation?-------,-------,------------,--------,--------

Fig 10.12.10 • NPSH? cont

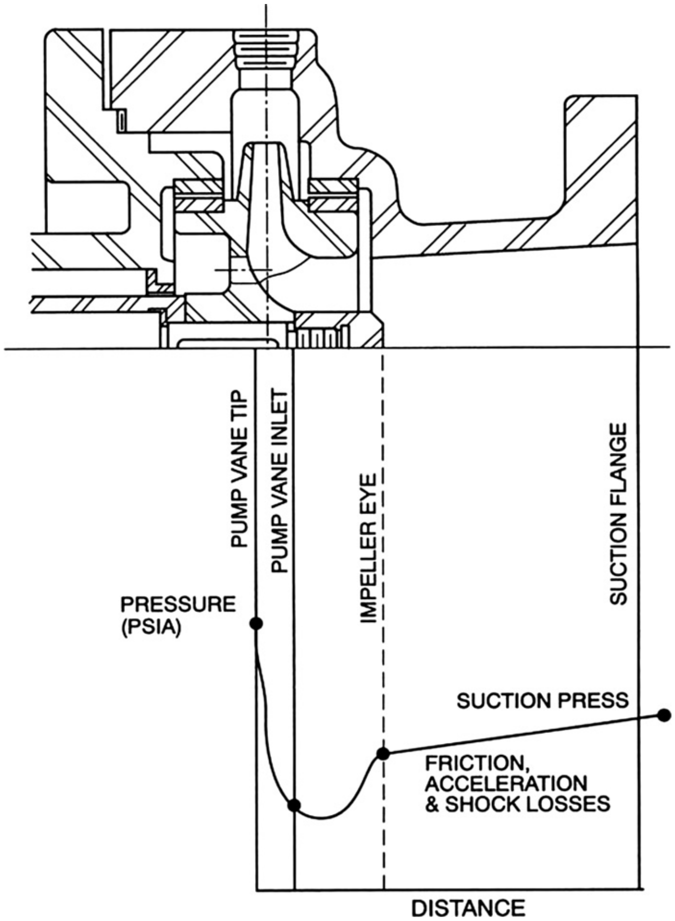


Fig 10.12.11 • ■

- What is the root cause of Cavitation?
- What can cause----------?
- Hint --it happens at low flow rates
- Look at Fig. This is a picture taken in a plastic pump with a strobe (disco) while the pump is operating at low flow. What do you see? That's right--recirculation. It's a type of cavitation and can be very bad!

Fig 10.12.12 • What is Recirculation?

- If a pump operates in the heart of the curve, no problem.
- If a pump does not operate in the heart of the curve, the seal, bearing, shaft, impeller or case can be damaged and vibration increases. Keep your eye on flow (if you have a flow meter) or what else tells you flow changed? ------,-----------,-----------

Fig 10.12.13 • In Conculsion

Pump in Classic Recirculation

Fig 10.12.14 • ■

Typical Refinery Mini Information Session -

Steam Turbines

Presented by Bill Forsthoffer

October 11,1998

Fig 10.12.15 • ■

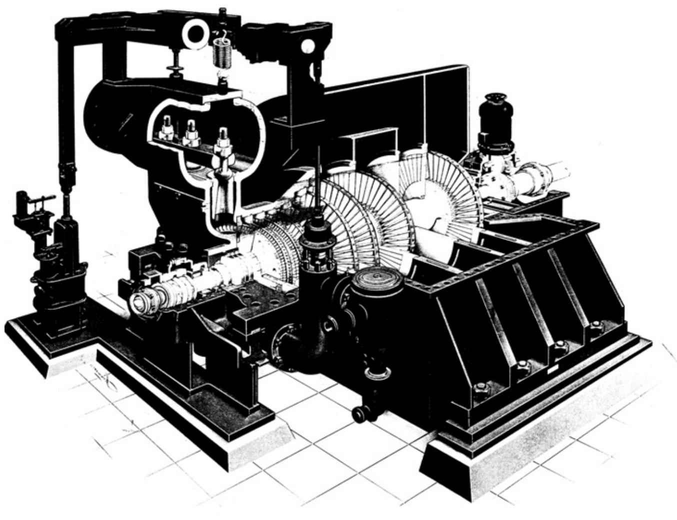


Fig 10.12.16 • ■

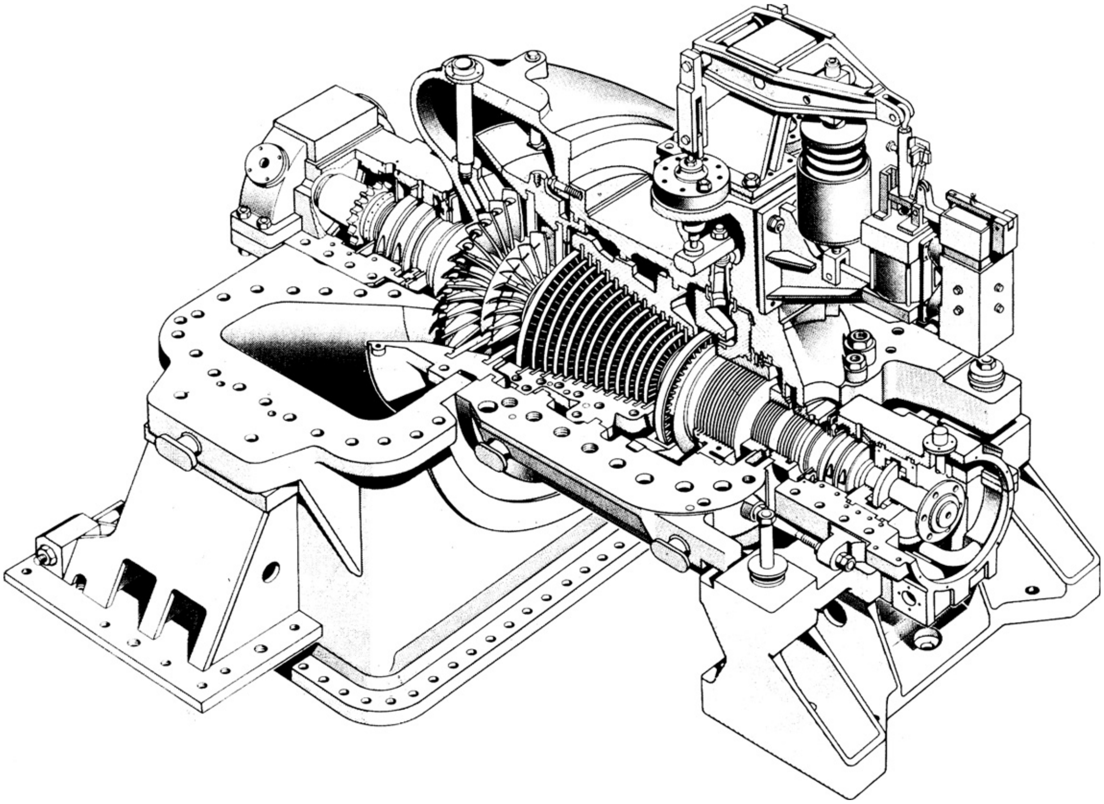


Fig 10.12.17 • ■

- Principles of operation
- Component functions:
- Steam Chest & Valves
- Nozzles & Blades
- Rotor & Bearings
- Steam Seals
- Start-up and Shut-down precautions

Fig 10.12.18 • Session Topics

- Please refer to Fig.10.12.16
- An expansion turbine uses high velocity gas (1100 KM/HR) to turn the rotor blades at approx.500 KM/HR.
- Since the most common gas is water vapor (steam)! However, steam is hot (800°F) and we must slowly heat up and cool down the turbine to prevent rubs

Fig 10.12.19 • Principles of Operation

- The Steam chest contains and guides the steam into the turbine
- The Steam control valve (not T&T valve) admits steam to the turbine with a minimum of pressure drop and guides the steam to a specific portion of the nozzle block

Fig 10.12.20 • Component Functions

- The nozzles accelerate the steam velocity from approx. 300KM/HR to 1100 KM/HR
- The steam enters the rotor blades (buckets) at 500 KM/Hr and turns the rotor
- The higher the pressure drop across the turbine, the higher the produced power and vice versa
- The lower steam temp., greater erosion

Fig 10.12.21 • Component Functions cont.

- The rotor transfers power to the coupling and the load (compressor, generator, etc.)
- The journal bearings support the rotor. A rotor operating at 3600 RPM rotates 60 times per second, 6,000,000 times per day and 6,000,000,000 times between T&'s, all on less than 0.001" oil film! So----

Fig 10.12.22 • Component Functions cont.

- The steam seal limits and directs steam leakage from the shaft to a desired location **and not the bearings!**
- If the steam seal is worn or the steam seal system is not functioning, water will contaminate the oil and cause premature bearing failure and steel component corrosion

Fig 10.12.24 • Component Functions cont.

- The thrust bearing supports the rotor axially.
- The condition monitoring of the thrust bearing is most important since the rotor will be machined by the nozzles and case if the thrust bearing fails--- and it will be fast! That's why thrust pad temp. should not exceed 240°F for any bearing

Fig 10.12.23 • Component Functions cont.

- Since the rotor is much less mass (thinner) than the case, it will heat up and cool down at a different rate
- Therefore, all steam turbines have **required start up and shut down heating and cooling rates**
- **Never rapidly start or stop a steam turbine! (Refer to instruction book)**

Fig 10.12.25 • Start-up and Shut-down

Best Practice 10.13

Implement the principle of CCM (component condition monitoring) for all site machinery to completely monitor all machinery trains and identify maintenance requirements for the next turnaround.

At the present time (2010), many plants still do not have an effective machinery condition monitoring program, and still rely on either preventive or breakdown maintenance.

Using the principle of CCM for all site machinery will optimize machine safety and MTBFs and minimize PM (preventive maintenance) and breakdown maintenance.

Effective communication and integration of maintenance and operation personnel will enable many maintenance tasks to be moved to a planned turnaround.

This approach should also be followed for spare equipment, since maintenance performed on spare equipment during plant operation renders the spared equipment critical (un-spared).

Lessons Learned

Absence of an effective site predictive maintenance (PDM) program exposes the plant to lower machinery safety and reliability. Implementation of a component condition based PDM will optimize machinery safety, reliability and daily revenue for critical equipment.

Benchmarks

This best practice has been used since 1990 to achieve optimum machinery safety, component MTBFs and reliability in all installations that have implemented this best practice.

This best practice is currently being used in the following industries to achieve optimum machinery train reliability (compressor trains > 99.7%):

- Oil and gas
- Chemical
- Refinery
- LNG

B.P. 10.13. Supporting Material

See B.P: 10.11 for supporting information.

11

Preventive and Predictive Maintenance Best Practices

Introduction

More plant revenue is lost by ineffective maintenance programs than by unreliable critical equipment (reliabilities below 99.5%).

Many end users still have a PM (preventive maintenance) heavy program, and do not use all their machinery monitoring instrumentation to effectively monitor each major component of their critical equipment trains.

Where plants still employ a PM heavy maintenance program, the majority do not extend their PM intervals by imputing the actual component condition from their predictive maintenance program (PDM).

Today (2010) the typical critical machinery train condition monitoring instrumentation accounts for 25% or more of the train capital cost. Process information systems (PI, PHD, Aspen, etc.) allow continuous monitoring and trending in the distributed control system (DCS).

The 'best of the best' plants make total use of all machinery instrumentation and their DCS system to trend the condition of each major component in their critical machinery trains. In addition, they use their PDM program results to maximize any required PM intervals and to extend all maintenance activities to the next scheduled shutdown if possible.

This chapter will therefore present PM and PDM machinery best practices, to enable your plant's maintenance program to be among the 'best of the best'.

Preventive Maintenance Best Practices – PM

While some machinery PM – Preventive Maintenance (time based) – is necessary, use the site PDM – predictive maintenance program (condition based) – to extend and optimize PM intervals.

Predictive Maintenance Best Practices – PDM

Experience indicates that all plants can extend the scope of their PDM – predictive (condition based) maintenance program. Most plants do not monitor performance or integrate machinery performance results with their mechanical condition monitoring program.

It appears that approximately 80% of the root causes of machinery component failure are related to process changes and yet many plant condition monitoring programs remain mechanical condition based.

Integrating machinery performance (efficiency, flow rate, head and power) trends with mechanical component condition trends will result in optimum machinery safety and reliability.

This section therefore will present PDM – predictive maintenance best practices – with the hope that your facility will optimize the use of PDM over PM maintenance practices to increase the safety and reliability of all installed machinery.

Best Practice 11.1

Always minimize machinery PMs and extend PM intervals by using condition monitoring (PDM) results.

Use plant, company and industry lessons learned to identify those PM (preventive maintenance) practices that can be eliminated by having an effective PDM (predictive maintenance program) in place.

Where PMs cannot be eliminated, utilize the results of the PDM – predictive maintenance program – to extend the PM intervals.

Lessons Learned

Preventive maintenance (time based) programs do not increase machinery reliability, are costly, eventually become ineffective, and expose the plant to un-scheduled shutdowns.

The following facts demonstrate PM effectiveness issues:

- Many plants 'talk' predictive maintenance (PDM) but still rely on time-based preventive maintenance (PM).
- The costs of PM maintenance for machinery are very large and require additional manpower.
- Periodic and repetitive maintenance routines can reduce the effectiveness of even superior maintenance teams.
- Whenever a spared unit is taken out of service for PM, the operating unit becomes a critical (un-spared) unit, thus exposing the process unit to reduced revenue of a process unit shutdown.

Benchmarks

We have recommended reduced PM and PM interval extensions since 1990 to all clients. Even today (2010), many plants do not eliminate critical equipment internal PMs by performance trending and schedule overhauls of the equipment regardless of performance parameter trends (head, efficiency, flow rates and power consumption). Where our PM reduction programs have been implemented, clients have experienced significantly increased machinery MTBFs (>100 months).

B.P. 11.1. Supporting Material

The five machinery failure classifications are presented in Figure 11.1.1.

The details concerning each of these failure classifications were discussed in the previous chapter. How can these failure causes be prevented?

- Process condition changes
- Improper assembly/maintenance/installation
- Improper operating procedures
- Design deficiencies
- Component wearout

Fig 11.1.1 • Failure classifications

Re-examination of the details concerning each failure classification shows that the solution to the prevention of each failure cause is identical. This fact is presented in Figure 11.1.2.

- Component function awareness (what should it do?)
- Component condition monitoring (what is it doing?)
- Using predictive maintenance techniques
- Teamwork – reliability is everyone's responsibility

Fig 11.1.2 • Prevent machinery failures by...

Let's now examine each of the action items noted above in detail.

Component function awareness – 'What should it do?'

Component (machinery part) function awareness allows you to determine what the component is supposed to do. It is obvious that a certain amount of knowledge is required to accomplish this fact. Remember, you may not have all of the knowledge required – so OBTAIN IT! Figure 11.1.3 presents sources of where the information may be obtained.

Thoroughly understand the function of each component by:

- Reading the instruction book
- Asking questions of:
  - Site reliability group
  - Site technical group
  - Machinists
  - Operators
- Referring to reference books
- Organizing 'mini' information sessions for operators and machinists

Fig 11.1.3 • What is it supposed to do?

Figure 11.1.4 presents the important principle of knowledge base. The greater this base, the more effective predictive maintenance and root cause analysis procedures will be.

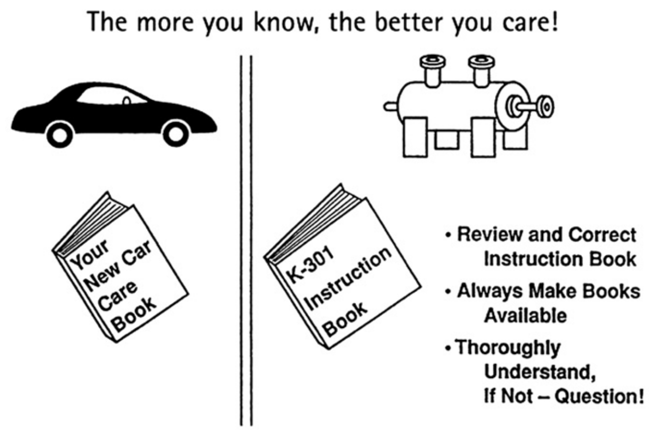


Fig 11.1.4 • The more you know, the better you care!

One final word. Do not be afraid to admit to management that you do not know certain aspects of a problem. But be sure to state that you will find out. After all, management must understand that this is a learning process and does require time.

To aid in the understanding of component function definition, we have included an example for an anti-friction bearing in Figure 11.1.5.

An anti-friction bearing continuously supports all *static and dynamic forces* of a rotor by providing sufficient *bearing area* and requires *oil flow* to remove the generated *frictional heat*.

This statement then defines the items that must be monitored to determine the bearing's condition:

- Static and dynamic forces
- Bearing area
- Oil flow
- Frictional heat

Fig 11.1.5 • Component function example

Naturally, we cannot measure directly all of the items noted in Figure 11.1.5. However, based on the instruments and measuring devices available on site, what can be measured to ensure the component (bearing) is performing correctly?

Component condition monitoring – 'What is it doing?'

In reference to the anti-friction bearing example, the component condition monitoring parameters are presented in Figure 11.1.6.

- Bearing housing vibration
- Bearing housing temperature
- Lube oil condition
  - Viscosity
  - Water content
  - Oil particle content

Fig 11.1.6 • Component condition monitoring parameters (for anti-friction bearing)

A similar exercise can be conducted for all of the major components and systems in any piece of equipment.

What are the major components and systems of any piece of rotating equipment? How many are there? And are the same components contained in any type of rotating equipment? The answers to these questions will be discussed in a later chapter of this book and form the principle of component condition monitoring.

Preventive (PM) and predictive maintenance (PDM)

At this point, the distinctions between preventive maintenance (PM), predictive maintenance (PDM) and troubleshooting must be discussed.

Preventive maintenance

Preventive maintenance is performed at predetermined intervals. It is time based. A very common preventive maintenance step is an automotive oil change. The objective of this action is to remove the oil from the engine before oil contamination and deterioration cause excessive wear to the engine components. Figure 11.1.7 presents the components of a typical site preventive maintenance program.

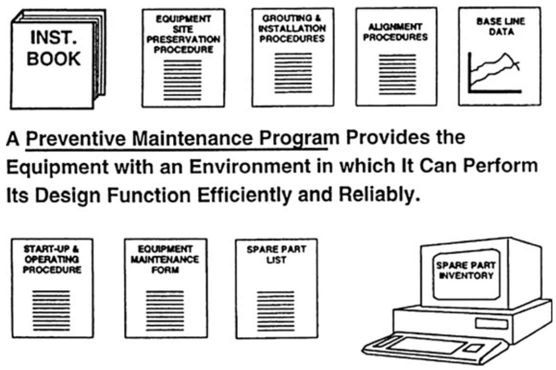


Fig 11.1.7 • A typical preventive maintenance program (Courtesy of M.E. Crane, Consultant)

In our experience, a well-planned preventive maintenance program can truly be effective. However, the question must be asked, 'Is the maintenance performed always necessary?' Refer to Figure 11.1.8.

- Preventive maintenance prevents but ... takes time.
- Is it always necessary?

Fig 11.1.8 • Preventive maintenance

What is the basis for replacing components? Unnecessary component replacement exposes the machinery unit to a failure classification (improper assembly of components, improper installation, component malfunction, improper component storage procedures, etc.).

In addition, preventive maintenance can cause a mindset that automatically determines maintenance at every turnaround regardless of component condition. This can be a costly practice.

A case history also demonstrates where preventive maintenance can lead if not properly monitored. A centrifugal compressor in a large refinery was scheduled for maintenance during the upcoming turnaround. Maintenance planning had scheduled bearing inspection and change if necessary. During the turnaround when bearings were inspected, excessive clearances and signs of deterioration were found. Naturally the bearings were replaced. However, because the bearings were replaced, it was decided that the seals, which are more difficult to remove and inspect, should be observed. Upon seal removal the seals were also in a distressed condition and needed to be replaced. Now the tough decisions had to be made. It was decided that the compressor would be disassembled, to inspect the interior

condition for possible causes of seal and bearing failure. Upon disassembly, no significant abnormalities were found within the compressor and it was consequently reassembled.

This case history demonstrates how a standard preventive maintenance approach can lead to unnecessary maintenance and significant loss of revenue to the operating unit. In this case, the operating unit did not make use of site instrumentation. Nowhere had people answered the question 'What changed?'. This approach therefore led to the unnecessary disassembly of the compressor. If bearing parameters (temperature, vibration, etc.) and seal parameters (inner and outer seal leakage) had been monitored for change, the conclusions that only the bearing and seal had changed would have been made without unnecessary disassembly. Remember, to disassemble a compressor, significant additional tools and materials are required, and it can easily reach one week. It can be seen therefore, that the effective way to perform any maintenance activity is to thoroughly plan that activity based on condition changes to equipment. This leads us to the discussion of predictive maintenance.

Predictive maintenance

Predictive maintenance is based on component condition monitoring and trending. Figure 11.1.9 presents the definition of predictive maintenance.

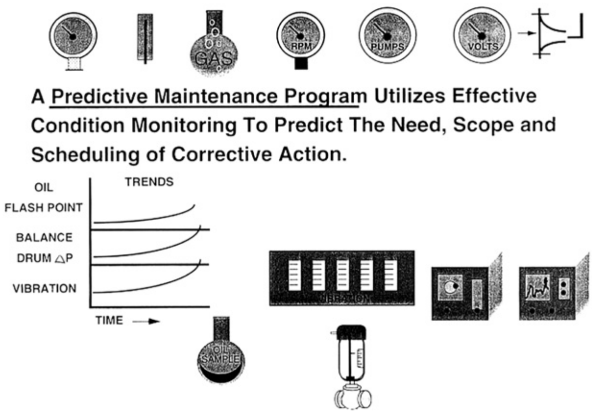


Fig 11.1.9 • A predictive maintenance program (Courtesy of M.E. Crane, Consultant)

Troubleshooting

Wherever I travel, worldwide, troubleshooting is the 'keyword'. More recently, other 'keywords' have emerged:

- Failure analysis
- Root cause analysis
- Reliability centered maintenance (RCM)

Regardless of the 'keyword', it's still troubleshooting. This term is defined in Figure 11.1.10.

What are the requirements to accomplish an effective troubleshooting exercise? These facts are presented in Figure 11.1.11.

Do these requirements sound familiar? They certainly should. These are the requirements for predictive maintenance!

Troubleshoot – to discover and eliminate (root) causes of trouble

Fig 11.1.10 • Definition

- Troubleshooting requires that all abnormal conditions be defined
- However, to determine abnormal conditions, the normal conditions must be known
- Therefore baseline (normal) conditions must be known

Fig 11.1.11 • An effective troubleshooting exercise

The differences between these two terms are presented in Figure 11.1.12.

- Predictive maintenance requires baseline and trend data to predict the root cause of the change in condition.
- Troubleshooting requires baseline and trend data to predict the root cause of failure

Fig 11.1.12 • Predictive maintenance and troubleshooting

Therefore, if we use site-wide predictive maintenance techniques, we can *potentially* detect a *change in condition* before failure. Please refer to Figure 11.1.13.

Is predictive maintenance after a failure!

Fig 11.1.13 • Troubleshooting...

Notice that in the discussion above, the word 'potentially' was in italics.

Remember that the majority of rotating equipment in any plant is general purpose or spared equipment that is not continuously monitored in the control room DCS system. This equipment is also the source of most reliability problems ('bad actors'). How can this equipment be effectively monitored?

Reliability, everyone's responsibility

Please refer again to Figure 11.1.4 of this chapter and review the analogy between your vehicle and site machinery. You'll have fewer problems if you and the mechanic (operators and machinists) know more and – work as a team!

Reliability must be everyone's responsibility. The entire plant; operations, maintenance and engineering departments

must be aware of the reliability program philosophy and must be able to implement it. Having operators and machinists equipped and trained in the use of simple vibration instruments (vibration pens), oil condition monitors and laser temperature guns will significantly increase the reliability of general purpose (spared) equipment through the implementation of an effective predictive maintenance program.

Best Practice 11.2

Maximize pump unit bearing bracket oil change intervals by using the principles of component condition monitoring (CCM) to extend pump bearing bracket oil change intervals from 13 weeks to as much as 104 weeks.

The principle of component condition monitoring addresses the five major components of any type of machinery to completely monitor its condition. (See B.P. 10.11 supporting material for details.)

Using these principles to trend rotor performance, bearing, seal and auxiliary system conditions, clients have extended the previous 13 week oil change out time in 13 week intervals from 13 to 104 weeks!

The savings in manpower and materials was tremendous and the exposure to reliability issues was significantly reduced.

Lessons Learned

13 or 26 week pump unit bearing bracket oil change intervals, when all component parameters are acceptable, expose the plant to additional maintenance, shutdowns and do not increase reliability.

Comparison of pump MTBFs before and after bearing bracket oil change intervals were increased from 13 weeks to over 100 weeks shows higher MTBFs for the extended interval pumps.

Benchmarks

This best practice has been recommended since the late 1990s, following experience with a number of refinery machinery reliability audits. Since that time, this best practice has produced pump MTBFs in excess of 100 months where they were less than 48 months prior to the extension of oil change intervals.

B.P. 11.2. Supporting Material

See B.P:11.1 for supporting material.

Best Practice 11.3

Optimize pump unit component MTBFs by changing over pumps every 3 to 6 months to minimize pump start-ups while ensuring stand-by pump unit availability.

Excessive pump start-ups expose components (especially mechanical seals) to reduced MTBFs.

Most plants do not have or follow a specified pump changeover program. Many plants change over pumps frequently (as often as once per week) while other plants do not change over pumps on a regular basis.

We recommend that a pump changeover program be developed, based on site pump wear experience. With the exception of firewater (mandatory weekly operation) and seawater (frequent operation to control corrosion) we recommend:

- The program initially require three month interval changeover
- The three month interval be extended month by month based on pump CCM (see B.P.10.11)
- The length of operation of the started pump be determined by:
- Pump driver – if the main pump is steam turbine driven to prevent electrical shutdowns and the started pump is motor driven – operate for four hours and ‘CCM’ this pump unit
- Plant spare pump philosophy:
- Equal wear philosophy – operate started pump for changeover interval
- ‘New spare’ philosophy – operate started pump for four hours and ‘CCM’ this pump unit

Lessons Learned

Plants that do not have an active and monitored pump changeover program experience significantly lower MTBFs.

I have never visited a plant where I was not asked my opinion of pump changeover philosophy.

Discussion always resulted in a recommendation consistent with the information in this B.P.

Unfortunately, follow-up usually has shown mixed results in implementation of the recommendations.

Benchmarks

This best practice has been recommended to all plants visited since the late 1980s. Where it has been followed, plant pump MTBFs have increased. There is an example of pump MTBFs being increased from less than 36 months to over 70 months in a process unit where the changeover program results are continuously monitored and implemented.

Best Practice 11.4

PM critical (un-spared) machinery train oil system and dry gas seal system valves and sensing line hardware at every turnaround to achieve optimum oil system and train reliability (>99.7%).

The operation of control valves, during steady state and transient conditions, in critical (un-spared) machinery train oil and dry gas seal systems directly affects the train reliability.

For this reason, we recommend that every control valve in the oil, dry gas seal and associated buffer gas systems be overhauled (diaphragm, packing, controller [if applicable]) and then checked for proper response as well as position indication.

If associated pulsation suppression devices (ball check/needle valve, needle valve and/or orifice) are installed, these devices must be inspected, cleaned and replaced if necessary. Note: Pulsation suppression devices must be installed in the proper orientation. Always match mark these devices to ensure they are re-installed in the proper direction.

Lessons Learned

Even though industry specifications require isolation valves and a manual bypass around each oil and gas system control valve, valve replacement during plant operation exposes the plant to an unscheduled shutdown and corresponding large revenue losses.

During site auxiliary system audits I have observed isolated control valves and the associated manual bypass valve in operation. Problems had existed with the valve and the plant was waiting until the next shutdown for replacement. Needless to say, any transient condition (main pump trip, two pumps operating, etc.) would expose the plant to a shutdown.

Benchmarks

This best practice has been recommended since the late 1980s during turnaround planning meetings for a large petrochemical complex. Since that time, this best practice recommendation, where followed, has resulted in optimum compressor train reliabilities (> 99.7%).

B.P. 11.4. Supporting Material

In order to prevent the potential reliability issues noted below, we recommend that this best practice be executed during every turnaround for critical equipment oil and dry gas seal systems.

System reliability considerations

Concerning auxiliary system control and instrumentation, a number of reliability considerations are worthy of mention.

Control valve instability

Control valve instability can be the result of many factors, such as improper valve sizing, improper valve actuators, air in hydraulic lines or water in pneumatic lines. Control valve sensing lines should always be supplied with bleeders to ensure that liquid in pneumatic lines or air in hydraulic lines is not present. The presence of these fluids will usually cause instability in the system. Control valve hunting is usually a result of improper controller setting on systems with pneumatic actuators. You are advised to consult instruction books to ensure that proper settings are maintained. Direct acting control valves frequently exhibit instabilities (hunting on transient system changes). If checks for air prove inconclusive, it is recommended that a snubber device (mentioned previously) be incorporated in the system to prevent instabilities. Some manufacturers install orifices which sufficiently dampen the system. If systems suddenly act up where problems previously did not exist, any snubber device or orifice installed in the sensor line should be checked immediately for plugging.

Excessive valve stem friction

Control valves should be stroked as frequently as possible to ensure minimum valve stem friction. Excessive valve stem friction can cause control valve instabilities or unit trips.

Control valve excessive noise or unit trips

Squealing noises suddenly produced from control valves may indicate valve operation at low travel (C_v) conditions. Valves installed in bypass functions that exhibit this characteristic may be signaling excessive flow to the unit. Remember the concept of control valves being crude flow meters. Observation of valve travel periodically during operation of the unit will indicate any significant flow changes.

Control valve sensing lines

Frequently, plugged or closed control valve sensing lines can be a root cause of auxiliary system problems. If a sensing line that is dead-ended is plugged or closed at its source, a bypass valve will not respond to system flow changes and could cause a unit shutdown. Conversely, if a valve sensing line has a bleed orifice back to the reservoir (to ensure proper oil viscosity in low temperature regions), plugging or closing the supply line will cause a bypass valve to fully close, rendering it inoperable, and may force open the relief valve in a positive displacement pump system.

Valve actuator failure modes

Auxiliary system control valve failure modes should be designed to prevent critical equipment shutdown in case of actuator failure. Operators should observe valve stem travel and pressure gauges to confirm valve actuator condition. In the event of

actuator failure, the control valve should be designed for isolation and bypass while on-line.

This design will permit valve or actuator change out without shutting down the critical equipment. During control valve on-line maintenance, an operator should be constantly present to monitor and modulate the control valve manual bypass as required.

Best Practice 11.5

Use the results of the plant component condition monitoring program (PDM) to extend oil lubricated coupling PM maintenance intervals.

Many plants have a PM (preventive maintenance) procedure in place for the yearly inspection and cleaning of oil lubricated couplings.

This activity is usually unnecessary at these intervals, and can be extended or postponed to a turnaround by reviewing the vibration trends and frequency trends of the PDM program (vibration monitoring program).

We recommend the following procedure for extension of lubricated coupling inspection intervals:

- Review the overall vibration and vibration frequency trends prior to the yearly oil lubricated coupling inspection
- Inspect and clean the coupling as required by the procedure
- Observe the overall vibration and vibration frequency trend after start-up
- If the vibration trends are not different, continue to trend the coupling vibration and perform the next coupling inspection on a condition basis (vibration values reach or exceed alarm setting)
- Try to extend maintenance activity to the turnaround – discuss with site vibration specialist, vendor and operations to determine action plan

Lessons Learned

Yearly inspection of oil lubricated couplings exposes the plant to unnecessary shutdowns.

Recent (2010) involvement with a site root cause failure analysis for a steam turbine overspeed incident required a review of all of the predictive (condition based) and preventive (time based) maintenance procedures with the affected train. An oil lubricated gear coupling was installed and had a yearly PM for inspection and cleaning. Review of vibration information, which did not show any significant change, led the recommendation of this best practice. The recommendation was accepted.

Benchmarks

This best practice has been recommended since the late 1980s to result in extended, and in most cases postponed, oil lubricated gear coupling inspections until the next turnaround.

B.P. 11.5. Supporting Material

Gear couplings

Gear-type couplings are shown in Figures 11.5.1 and 11.5.2. They usually include two separate gear mesh units, each consisting of an external gear which fits closely into an internal gear. The internal gear can either be part of the coupling hub assemblies or mounted on each end of the coupling spacer assembly. If the internal gears are hub-mounted, then the external gears are spacer mounted and vice-versa.

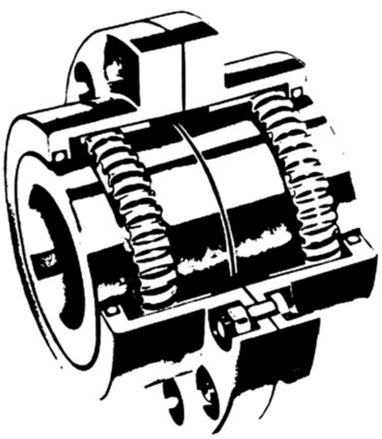


Fig 11.5.1 • Gear tooth coupling (grease packed) (Courtesy of Zurn Industries)

Grease pack couplings (see Figure 11.5.2) are normally designed with hub-mounted external gears, and the internal gears are part of a sleeve-type spacer which serves as a retainer for the grease lubrication. The flange joint of the sleeve is either precision ground to avoid lubrication leaks, or has a gasket between the two flange faces. The sleeve ends are fitted with ‘O’ ring seals to keep dust out and lubrication in.

In recent years, flexible element couplings have been used almost exclusively. However, many older gear-type couplings are still in use. They are the most compact coupling for a given amount of torque transmission of all the coupling designs. For this reason, they also have the least overhung weight. In addition, the gear coupling can adapt more readily to requirements for axial growth of the driver and driven shafts. Axial position change tolerances are on the order of 12.7mm ($^1/_2$") or greater.

There is a common disadvantage in all gear-type flexible couplings. Any gear mesh has a break-away friction factor in the axial direction. This is caused by the high contact force between the two sets of gear teeth. The result is that the forces imposed on the driver and driven shafts are not totally predictable, and are sometimes higher than desired due to the quality of the tooth machine surfaces and the inevitable build up of sludge or foreign material in the tooth mesh during

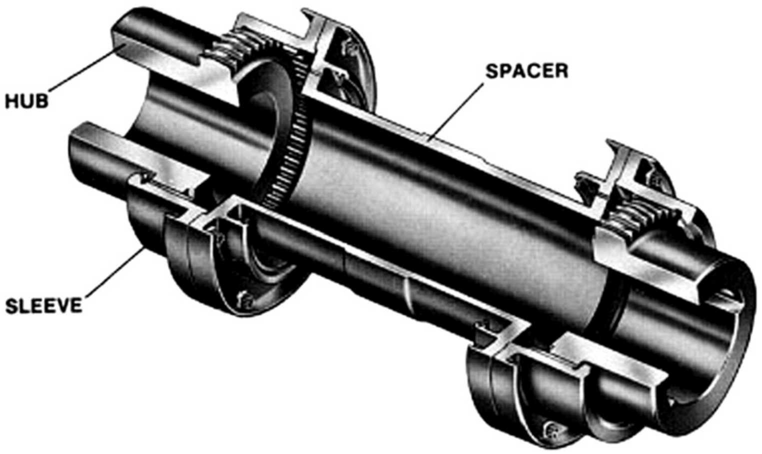


Fig 11.5.2 • Continuously lubricated gear type coupling with spacer (Courtesy of Zurn Industries)

extended service. These forces are detrimental to the ability of the coupling to make the required corrections for misalignment but, more importantly, can have a disastrous effect on the ability of the coupling to correct for thermal or thrust force changes between the driver and driven machines.

Both coupling manufacturers and users have long been aware of this problem and have used many methods to minimize the effect. Some of these methods are:

- Reduction of the forces between the gear teeth by increasing the pitch diameter of the gear mesh. This is often self-defeating in that it results in increased size of the coupling and the coupling weight.
- Reduction of the break-away friction factor by the use of higher quality gear tooth finish and better tooth geometry and fit.
- Reduction of sludge and foreign material build up in the gear mesh by finer filtration of the coupling lubricant.
- Reduction of sludge and foreign material build up in the gear mesh by incorporating self flushing passages and ports in the coupling to allow any contaminants to pass through in the lubricant without being trapped in the gear mesh area.

These steps have been only partially successful and the problem still exists in many applications.

Coupling manufacturers are asked to quote the design break-away friction factor of their coupling as built and shipped from the factory. Machinery train designers then use this figure to calculate the maximum axial force that the coupling would be expected to exert on the connected shafts. From this information, the designers can decide if the thrust bearings adjacent to the coupling are adequate to handle the axial loads within the machine plus the possible load from the coupling resistance to any external forces.

There has been much discussion and some disagreement regarding the friction factor to be used when calculating possible thrust forces which can be transmitted by the coupling. When the coupling is in reasonably good condition, factors from 0.15 to 0.30 have been considered reasonable. Since the factor reflects the total force relationship, the coupling design can have a significant effect on the factor used. The factor is a function of the number of teeth in contact and the contact areas of each tooth plus the quality of the tooth contact surface. If we assume that the factor to be used is 0.30, then the axial force which must be exerted in order to allow the coupling to correct for axial spacing changes can be calculated as:

$$F_a = \frac{0.30 \times T}{Dp/2}$$

Where: F_a = Required axial force in pounds
T = Design torque in in/pounds
Dp = Pitch diameter of gear mesh in inches

We can assume then, that if we use a coupling with a six inch pitch diameter gear mesh transmitting 25,000 in/lb of torque and a break-away friction factor of 0.30, the axial force required to move the gear mesh to a new axial position would be 2,500 lb. Adjacent thrust bearings must be capable of handling this force in addition to the machine's normal calculated thrust forces. Machinery train designers and users must be aware of this and make provisions for it in the built-in safety factors of thrust bearings and machinery mounting design.

The machinery user must know that the same phenomenon has an effect on machinery vibration when machinery is operated with excessive misalignment. The gear mesh position must change with each revolution of the shaft to correct for the misalignment. This results in counter axial forces on a cyclic basis since the mesh is moving in opposite directions at each side of the coupling. Vibration detection and monitoring instrumentation will show that the resulting vibration will occur at twice the running frequency of the shafts. Although the primary force generated is axial, the resultant can show up as a radial vibration due to the lever arm forces required on the coupling spacer to make the gear meshes act as ball and socket connections. Axial or radial vibration in rotating machinery which occurs at twice the frequency of the shaft rotational speed will normally be an indication of misalignment between the two machines.

Figure 11.5.2 shows a continuously lubricated, spacer gear-type coupling. Spacers are usually required for component removal (seals, etc). They also provide greater tolerance to shaft misalignment. A common spacer size used for unspared (critical) equipment is 18 inches.

Best Practice 11.6

Always perform transient oil system tests (main oil pump forced trip and aux. pump start without system instability or shut down) prior to shutdown to ensure optimum oil system operation and reliability.

These tests performed during equipment shutdown (at turnarounds) do not duplicate operational characteristics and lead to false conclusions regarding system transient response.

Follow the procedure noted below (see Supporting Material) immediately before the turnaround and take appropriate action to correct issues during the turnaround.

Lessons Learned

Oil system transient tests performed during a turnaround have shown acceptable results only to find that the system did not properly respond during plant operation resulting in a unit shutdown on low oil pressure.

If it is suspected, from past experience, that the transient test will not be successful it is recommended that advance engineering be performed for installation of a properly sized and instrumented dual stainless steel accumulator assembly during the turnaround. See B.P. 7.14.

Benchmarks

This best practice has been recommended since the late 1980s, to identify and resolve long term critical oil system reliability issues that had previously caused large unit revenue losses over the life of the critical machinery train.

B.P. 11.6. Supporting Material

Functional testing

Having satisfactorily installed and flushed the auxiliary system, all auxiliary equipment should be functionally tested and all instruments and controls checked for proper setting prior to operation of equipment. A functional test outline and procedure is included in the end of this section. We will highlight the major considerations of the procedure at this time.

It is recommended that the console vendor and/or the equipment purchaser prepare a detailed field functional test procedure and calibration check form. The format of this procedure can follow the factory test procedure if this is acceptable. As a minimum, the auxiliary system, bill of material, and schematic should be thoroughly checked in order to include the calibration and functional test of each major component in the auxiliary system – that is, components on consoles and at unit interfaces. A detailed record should be kept of this functional test procedure. This will help significantly during the operation of the unit. The test procedure should be accomplished without the critical equipment running initially and then with the auxiliary system at design operation conditions as closely as possible. The functional test procedure should first require that all instrumentation is properly calibrated before proceeding. Each specific functional test requirement should then be performed and results noted. If they do not meet specified limits as noted, testing should stop and components should be corrected at this point. Each step should be followed thoroughly to ensure each component meets all requirements. It is recommended that operators assigned to this particular unit assist in functional testing to familiarize themselves with the operation of the system. In addition, site training courses should be conducted prior to functional test to familiarize operators with system's basic functions. This training, again, significantly increases understanding of the equipment and ensures unit reliability.

Satisfactory acceptance of a functional test then ensures that the unit has been designed, manufactured, and installed correctly such that all system design objectives have been obtained and that equipment reliability is optimized.

One remaining factor to be proven is the successful operation of the system with the critical equipment unit in operation. During initial start-up, it is recommended that the functional test be re-performed with the unit operating. While this advice may seem dangerous, unless the unit operators ensure that the subject system has the ability to totally protect critical equipment while operating, auxiliary equipment will never be tested while the unit is in operation.

Remember, critical equipment is designed for 30 years or greater life. The components that comprise the auxiliary system are many and have characteristics that will change with time. Therefore, reliability of auxiliary systems can only be maintained if the systems are totally capable of on-line calibration and functional checks. The functional pre-commission procedure should be modified to include an on-line periodic functional checking procedure. Such a procedure is included at the end of this section.

At this point, we can clearly see that the major determination of continued equipment reliability rests with the operation, calibration and maintenance of the equipment. In order to assume maximum continued auxiliary equipment reliability, periodic on-line functional checks and calibrations must be performed. How can this be done? The only way is by convincing plant operations of the safety and reliability of the procedure for on-line checking. This can be reinforced during pre-commissioning by including operators in functional testing checks and on-site training sessions to show the function of the system. A site training course modified for the specific equipment would prove immensely valuable in achieving those results.

Only by involving unit operators in the pre-start check ups can it be hoped to establish a field functional checking procedure that will be utilized and followed through. Remember, a pressure switch less than $400 in cost could cause equipment shutdown that could reduce on-site revenue on the order of $1-2m U.S. dollars a day. The pressure switch selected could be the best, the highest quality in the world, properly

installed and set. If its calibration is not periodically checked, it could cause an unnecessary shutdown of equipment and result in this revenue loss.

Functional lube/seal system test procedure outline

Objective:	To confirm proper functional operation of the entire system prior to equipment start-up
Procedure format:	Detail each test requirement. Specifically note required functions/set points of each component. Record actual functions/set points and *all* modifications made.
Note:	All testing to be performed *without* the unit in operation.

I Preparation

A. Confirm proper oil type and reservoir level
B. Confirm system flush is approved and *all* flushing screens are removed
C. Confirm all system utilities are operational (air, water, steam, electrical)
D. Any required temporary nitrogen supplies should be connected
E. All instrumentation must be calibrated and control valves properly set
F. Entire system must be properly vented

II Test procedure

A. Oil Reservoir
 1. Confirm proper heater operation
 2. Check reservoir level switch and any other components (TIs, vent blowers, etc.)
B. Main pump unit
 1. Acceptable pump and driver vibration
 2. Absence of cavitation
 3. Pump and driver acceptable bearing temperature
 4. Driver governor and safety checks (uncoupled) if driver is a steam turbine
C. Auxiliary pump unit
Same procedure as item B above.
D. Relief valve set point and non-chatter check
E. Operate main pump unit and confirm all pressures, differential pressures, temperatures and flows are as specified on the system schematic and/or Bill of material
F. Confirm proper accumulator pre-charge (if applicable)
G. Confirm proper set point annunciation and/or action of *all* pressure, differential pressure and temperature switches.
H. Switch transfer valves from bank ‘A’ to bank ‘B’ and confirm pressure fluctuation does not actuate any switches.
I. Trip main pump and confirm auxiliary pump starts without actuation of any trip valves or valve instability.
Note: Pressure spike should be a minimum of 30% above any trip settings
J. Repeat step I above but slowly reduce main pump speed (if steam turbine) and confirm proper operation
K. Simulate maximum control oil transient flow requirement (if applicable) and confirm auxiliary pump does not start
L. Start auxiliary pump, with main pump operating and confirm control valve and/or relief valve stability
Note: Some systems are designed to *not* lift relief valves during two pump operation

III Corrective action

A. Failure to meet any requirement in Section II requires corrective action and retest
B. Specifically note corrective action
C. Sign-off procedure as acceptable to operate

Best Practice 11.7

Continuously monitor all dynamic equipment performance (pumps, compressors, steam and gas turbines) to prevent component failures and to optimize component MTBFs, unit safety and reliability.

Approximately 80% of machinery component failures are related to process condition changes.

Failure to monitor, calculate and trend machinery performance (efficiency, flow rate, head and power) will lead to false root cause analysis conclusions and reduced component MTBFs.

Programs are available for all machinery types to download measured performance parameters (pressures, temperatures, flows and power), perform required calculations and trend efficiency, flow, head and power to determine:

- If operating condition changes are possible to optimize machinery performance
- If machinery internal inspection is required and the predicted maintenance requirements
- If machinery maintenance can be extended to the next turnaround

Lessons Learned

Failure to trend machinery operating points and performance indicators reduces machinery safety and reliability

As previously stated 80% of component failure root causes lie in the effects of the process.

Failure to integrate the machinery operating point and internal performance to mechanical effects (vibration, temperature, etc.) impact component MTBF by not identifying the root causes of the reliability issues.

Benchmarks

This best practice has been used since 1984, while I was involved with the start-up of a large petrochemical complex to identify pump operation outside the EROE (see B.P. 2.7) and compressor and steam turbine internal fouling issues for on-line cleaning without the necessity of shutdowns.

B.P. 11.7. Supporting Material

Refer to the following best practices for supporting material concerning performance monitoring and trending:

- Pumps ——————B.P. 2.7
- Compressors ——————B.P. 3.14, 3.17 and 3.27
- Steam turbines ——————B.P. 5.4 and 5.13
- Gas turbines ——————B.P. 6.9 and 6.10

Best Practice 11.8

Use predictive maintenance results (PDM) to reduce PM – preventive maintenance tasks – and expand PM intervals.

Continually review all preventive maintenance (PM) tasks along with PDM results to eliminate PM and/or extend intervals.

Lessons Learned

PM-heavy programs produce lower machinery MTBFs and more breakdown maintenance.

Benchmarks

This best practice has been used since 1990 to optimize plant machinery safety and reliability. Extension of PM intervals has increased low pump MTBFs (less than 12 months) to over 48 months.

B.P. 11.8. Supporting Material

See B.P. 11.1 for supporting material.

Best Practice 11.9

Organize the plant reliability effort to integrate operations and process personnel into the existing maintenance dominated reliability program to:

- Bring 'process awareness' into the program to significantly increase machinery reliability.
- Increase operator and process engineer awareness of machinery reliability key factors.
- Increase recommendation implementation by having process and operations support.
- Minimize 'finger pointing' between maintenance and operations.

The best practice of having operations and process engineering personnel in the reliability group has been implemented by many clients to result in plant machinery safety and reliability of the highest degree.

Lessons Learned

Reliability groups which do not include operations and process engineering input produce lower machinery MTBFs and less implementation of recommendations.

Maintenance-centered plant reliability programs produce lower machinery MTBFs than reliability programs that have integrated maintenance, operations and process engineering functions.

Benchmarks

This best practice has been recommended since the mid-1990s when the company was involved with a number of site machinery audits in refineries and gas plants. Implementation of this best practice led to significant improvement of machinery reliability plant-wide.

B.P. 11.9. Supporting Material

Regardless of the level of a site reliability optimization program, it can be improved. My personal experience is that:

- Bad actors* are defined but not permanently solved.
- The majority of maintenance activities still are reactive – 'firefighting'.
- Preventive maintenance (time based) activities are excessive.
- Predictive maintenance (condition based) activities still are minimal.
- Some action plans for bad actor resolution have been defined but are not implemented.
- Lessons learned, from bad actors, have not become best practices for machinery upgrades and/or new equipment purchases.
- The reliability effort is maintenance based and does not incorporate other site disciplines.

*A bad actor is defined as any machine or stationary item that experiences one or more ESDs per year (unscheduled shutdowns or failures). Note that while this book is devoted to rotating equipment, the principles discussed here are equally applicable to all the equipment in any plant.

Why are these characteristics prevalent in most plants? My opinion is that the plant management has not been convinced of the opportunities available to save considerable operating costs and increase profits by reducing downtime related to reliability issues and excessive maintenance time. Plant management must be shown results and the associated savings before they will endorse any plan for reliability optimization. Often, a 'salesman' that has the ear of management is required. I have found that this person is usually a trusted senior operator or a process engineer. And considering that most reliability efforts are maintenance based, this salesman is nowhere to be found.

Setting up an effective multi-disciplined site reliability initiative

I have worked with many site reliability efforts, which have had different names, different cultures and different levels of experience. In the early days of my experience with these efforts (1970s and 1980s) there was but one constant – all efforts were strictly maintenance based. Oh, yes, there was another constant – opportunities for reliability improvement that were identified were not implemented.

The efforts also usually had a name, some of which were: the reliability group, the vibration monitoring group, the failure analysis unit, the failure analysis and PM group, etc., etc. Regardless of the name, the effort was aimed at improving site reliability of all equipment, not only rotating equipment. They definitely achieved results, but they were usually short term and problems recurred. Over the years, as is natural, the effectiveness of the program in a certain plant varied as personnel entered and left it. Those who left usually did so for higher pay, and those that entered did so for increased experience.

Review of the historical efforts of these groups also showed an interesting similarity. Every group or plant effort usually had identified the root cause of a particular problem, but was unable to gather sufficient, continued management support to implement the identified action plan. As a corporate consultant for a major oil and chemical company and later as an independent consultant, I would be asked to review various site reliability problems and recommend a cost effective action plan. In the majority of the cases, my end result was very close to what had been recommended prior to my study. However the difference was that my action plan was usually accepted and worked (just as the site group's plan would have also been successful!). This is not very encouraging to someone who works long hours, is 'on call' and has to attend to reactive maintenance issues at the usual times – weekends! Incidentally, have you ever noticed that equipment usually decides to 'pack it in' on the weekend everywhere in the world (Thursday, Friday in the Middle East, Saturday, Sunday in the West, etc.)?

I asked myself, why this was the case? After observing the characteristics noted above for a number of years, my opinion is that the efforts lacked an effective sales program. As a result, since the mid-1990s, I have been recommending that all site reliability efforts are integrated and definitely include an operations representative who can be a process engineer or senior operator or both. This is because all site equipment reliability depends upon the process requirements and because, let's face it, the plant is run by operations and if there is an element of operations in the reliability group who agrees with the group's recommendation, then acceptance and implementation have a much better chance. Because … there is now a salesman in the reliability group!

Naturally the approach will be different in each plant and there are as many possible variations as there are plants. Figure 11.9.1 presents some of the possible structures of a site reliability initiative.

- Specific, multi-discipline experienced reliability group
- A site culture change that makes reliability everyone's responsibility
- Operation, process and instrument reliability group members are on a one year rotation assignment

Fig 11.9.1 • Site reliability initiative guidelines

Rotation of the operations, process and perhaps instrumentation members of the reliability group has shown to be a very good idea, since returning members of the group to their own disciplines will naturally spread the word regarding function awareness and the importance of the program. It amounts to automatic function awareness training! Selection of the members of the group on rotation should be made carefully. I have found that these people should be the 'believable people' (experienced personnel who have the respect of their coworkers).

Remember again that regardless of make-up, the initiative must be multi-disciplined. The advantages of this arrangement are many. The major ones are presented in Figure 11.9.2.

- Valuable process information input
- Process and operation members are salesmen
- Significantly greater degree of function awareness of site equipment among all site disciplines
- Higher degree of ownership among all site disciplines
- Less 'finger pointing' when problems occur

Fig 11.9.2 • Multi-disciplined reliability group advantages

It is interesting to note that smaller, remote plants (Arctic region, New Zealand, Southern Chile and platforms) have consistently out-performed larger units in terms of action plan implementation. In these locations, they have to work together! And when they do, since everyone learns something regarding other discipline responsibilities and work scope, the results and acceptance of action plans flow smoothly.

Once the decision is taken to include operations and maintenance groups in the reliability program, consideration should be given to establishing 'ownership' in these groups with regards to the program.

Best Practice 11.10

Conduct operator awareness training for all shifts on a regular basis.

This training should cover the following:

- Increased machinery component condition monitoring outside of the control room
- Provide advance warning of component condition changes
- Initiate mutual cooperation between operations and maintenance to extend maintenance cycles if possible to the turnarounds
- Minimize 'finger pointing' between maintenance and operations

Operators are trained troubleshooters.

Around 80% of machinery failure root causes are the result of process changes.

Increasing operator awareness of the functions of machinery components will increase machinery safety and reliability.

Lessons Learned

Operations and process engineering groups welcome and need increased machinery functional awareness. Failure to provide this information will expose the plant to safety and reliability issues.

Experience in presenting site-specific machinery workshops since 1986 has shown that operators and process engineers lack machinery functional awareness knowledge and welcome it when it is offered. I have constantly been asked by experienced operators, 'Why wasn't this information given to me at the beginning of my career? I could have saved a lot of problems and shutdowns'.

Benchmarks

The best practice of increasing the machinery component awareness and interest in component condition monitoring of operators since the late 1980s has resulted in increased machinery safety and reliability in all plants where site-specific operator training sessions have been presented worldwide.

Best Practice 11.11

Maintain a 'best in class' reliability group by sending members (maintenance, operations and engineering) to yearly key machinery seminars.

Some recommended seminars are:

- Annual Texas A & M Turbomachinery
- Annual Texas A & M Pump
- Annual Gas Turbine Users Association
- Seminars sponsored by the ASME (American Society of Mechanical Engineers)
- Global Reliability Conferences

Continuous acknowledgement of the needs and interests of personnel who are associated with machinery safety and reliability will improve the plant safety and reliability effort.

Attendance at key machinery seminars and training workshops exposes the personnel to practices of other plants and will expand the alternatives for machinery reliability improvement.

Lessons Learned

Plants who do not send personnel to machinery seminars on a regular basis suffer lower machinery MTBFs and do not continuously improve their reliability practices.

Discussion with plant personnel always results in questions concerning available seminars, and requests that we, as machinery consultants, recommend seminar attendance to plant management in the hope that plant personnel will be sent to important industry seminars on a continuous (yearly) basis.

Benchmarks

This best practice has been recommended to clients since 1990. When this best practice has been implemented, site machinery MTBFs have increased as a result of corrective action taken to resolve long-term plant reliability issues. The action plans have come from information obtained during attendance at important industry seminars.

Best Practice 11.12

Conduct yearly machinery vendor improvement meetings on site.

These meetings serve to:

- Alert the vendors of pending reliability issues
- Obtain vendor machinery reliability improvement recommendations
- Establish a 'team' spirit with the vendor
- Establish a working relationship that will last for the life of the plant and not be dependent on personnel traits and relationships

After plant start-up, machinery vendor contacts are few and usually only for:

- Breakdown maintenance activities
- Turnaround activities
- Machinery re-rates (increased capacity modifications)

Establishing a yearly vendor contact and having site meetings will ensure continued vendor support of the highest quality.

Lessons Learned

Plants that do not establish and nurture a plant/vendor relationship are subject to varying support over the life of the plant, dependent on personal contacts and the individuals involved with a specific task.

Discussions with clients over the years has shown that failure to maintain a consistent vendor relationship after plant start-up results in an inconsistent quality of vendor service engineer and home office support.

Benchmarks

This best practice has been recommended since 1990 to all clients. Where this recommendation has been followed, clients have experienced continuous vendor support of the highest quality. In addition, they have been immediately alerted to any required modifications or improvements that could be executed during the next turnaround.

12

Implementation and Communication Best Practices

Introduction

As stated in the preface to this book, the objective was to present machinery best practices to site personnel to enable the highest degree of machinery safety and reliability to be achieved in their plant. As valuable as the best practices are in achieving this objective, results will only be attained with continuous management support.

Therefore this final chapter includes those implementation and communication best practices that have resulted in the highest level of success.

Best Practice 12.1

Obtain and maintain continued management support for machinery reliability issues by:

1. Preparing a brief statement of the specific reliability issue.
2. Clearly stating the impact of the issue on plant safety, reliability and revenue (cost of unavailability).
3. Presenting the reliability improvement plan and its impact on improved safety, reliability and revenue.
4. Defining the total cost of implementing the reliability improvement plan.
5. Showing the cost savings of executing the proposed reliability improvement plan.
6. Providing timely updates detailing the progress of the reliability improvement program.

Lessons Learned

A low implementation rate of reliability improvement recommendations is usually the result of poor presentation to plant management.

A review of past failed machinery reliability recommendations has shown:

- A lack of concise identification of the cost impact of the issue on plant revenue.
- The failure to identify the savings to be gained by implementation of the proposed reliability improvement recommendation.

Benchmarks

This best practice has been recommended to clients since 1990. It has resulted in a significantly increased recommendation implementation rate (above 50%) and increased machinery safety and reliability.

B.P. 12.1. Supporting Material

Reliability optimization is an important part of plant revenue and profit. The objective of this section is to provide information that will enable the reader to optimize reliability by implementing proven methods I have used throughout my career. The major components of reliability improvement are shown in Figure 12.1.1.

Before these objectives can be met and implemented, a number of important concepts and terms need to be reviewed and presented.

To optimize site rotating equipment availability by implementation of practical:

- Site reliability audits
- Assessment methods
- Availability improvement plans
- Condition monitoring techniques
- Preventive and predictive maintenance plans
- Troubleshooting techniques

Fig 12.1.1 • Reliability improvement objectives

The end user's objectives

The objectives of the end user are shown in Figure 12.1.2.

In order to maximize profit, a piece of machinery must have maximum reliability, maximum product throughput and minimum operating costs (maximum efficiency). In order to achieve these objectives, the end user must play a significant part in the project during the specification and design phase, and not only after the installation of the equipment in the field. Effective field maintenance starts with the specification phase of a project. Inadequate specifications in terms of instrumentation and the location of instrumentation will impact equipment reliability.

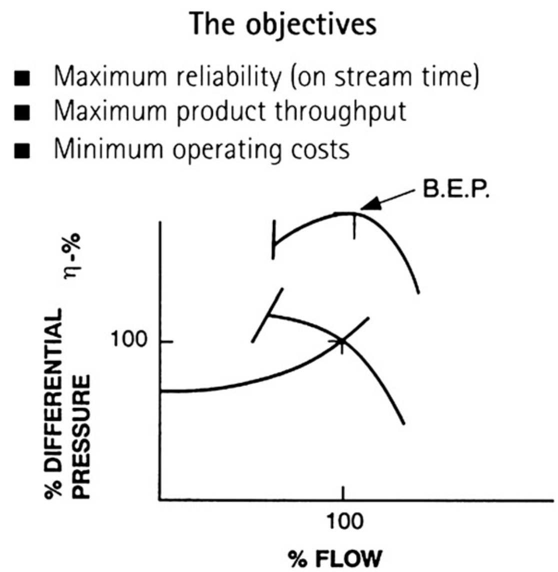


Fig 12.1.2 • The objectives

It is important to understand that the life span of rotating equipment is extremely long compared to the specification, design and installation phase. Refer to Figure 12.1.3.

A typical installation will have a specification, design and installation phase of only approximately 10% of the total life of the process unit. Improper specification, design or installation will significantly impact the maintenance requirements, maintenance cost and availability of a particular piece of machinery. Proper screening of equipment design (pre-bid technical meetings etc.) prior to equipment vendor selection establishes the foundation on which reliability is built. Likewise, enforcing shipment, construction, installation and commissioning specifications optimizes reliability and truly makes it 'cost effective' in terms of the life cycle of the equipment.

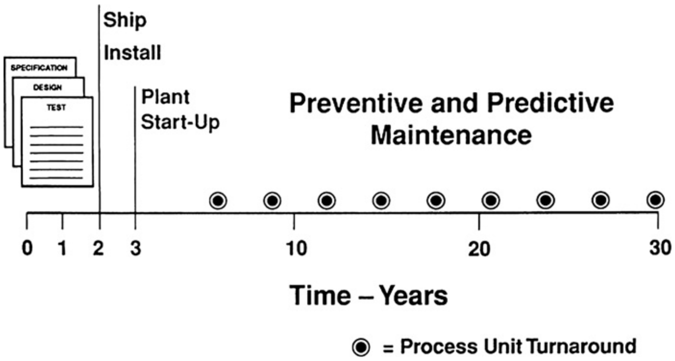


Fig 12.1.3 • The life span of rotating equipment

Reliability terms and definitions

Before we can optimize reliability, certain terms and definitions need to be presented. These terms are shown in Figure 12.1.4.

- Reliability
- Availability
- Maintainability
- Cost of unavailability

Fig 12.1.4 • Reliability terms

Reliability

Reliability is the ability of the equipment unit to perform its stated duty without a forced (unscheduled) outage in a given period of time (see Figure 12.1.5).

The definition of reliability for critical (un-spared) equipment is presented in Figure 12.1.6.

In the case of general purpose equipment (spared), reliability is not usually calculated since a spare unit should be available for operation if required. In the case of unreliable general purpose units, reliability could be defined as shown in Figure 12.1.7.

Note in Figures 12.1.6 and 12.1.7 that reliability does not account for planned downtime for preventive and/or predictive maintenance.

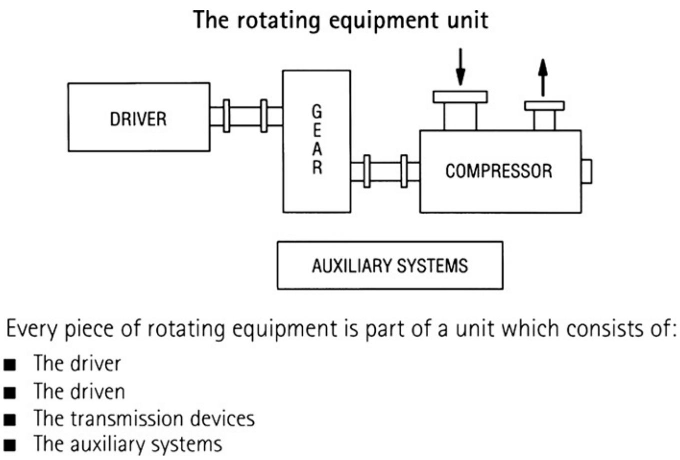


Fig 12.1.5 • The rotating equipment unit

The amount of time equipment operates in one year

$$\text{Reliability } (\% =) \left(\frac{\text{Operating hours per year}}{\text{8760 hours}}\right) 100$$

Fig 12.1.6 • Reliability – critical equipment

Hours per year spared equipment operated as a percentage of the hours it was required to operate

$$\text{Reliability } (\% =) \left(\frac{\text{Yearly hours in operation}}{\text{Yearly main unit forced outage hours}}\right) 100$$

Fig 12.1.7 • Reliability – general purpose (spared) equipment

Availability

Availability considers preventive and predictive maintenance downtime as shown in Figure 12.1.8.

One measure of both reliability and availability is mean time between failure (MTBF). See Figure 12.1.9.

The amount of time equipment operates in one year as a percentage of the available hours per year

$$\text{Reliability } (\% =) \left(\frac{\text{Yearly operating hours}}{\text{8760} - \text{planned downtime (T \& Is or turnarounds)}}\right) 100$$

Fig 12.1.8 • Availability

$$\text{MTBF} = \frac{\text{Total operating hours}}{\text{Number of failures}}$$

Fig 12.1.9 • Mean time between failure

Maintainability

Simply stated, maintainability is the ability to perform all maintenance activities; preventive, predictive and forced outage in a minimum time that requires rotating equipment unit shutdown. It is understood that the total maintenance time required will restore the unit to its original ‘new’ condition.

One parameter that can be used to measure maintainability is mean time to repair – MTTR as shown in Figure 12.1.10. The lower the MTTR, the greater the maintainability.

Cost of unavailability

All terms discussed so far, reliability, availability and maintainability directly affect the product revenue of the plant. Product revenue is the value obtained from one day’s production

$$MTTR = \frac{\text{Total repair hours*}}{\text{Number of repair events}}$$

*includes corrective maintenance time
(actual labor hours as a result of design modifications)

Fig 12.1.10 • Mean time to repair

expressed in local currency. At this point, note the amount of daily revenue from a process unit in your plant in Figure 12.1.11. Note that amounts can vary from $100,000 to over $5,000,000 per day depending on the process and the size of the unit.

If a critical equipment unit suffers a forced outage or is out of service due to poor maintainability (extended repair time), the product revenue shown in Figure 12.1.11 will be lost for each day the critical equipment unit remains out of service. Therefore, the cost of unavailability is the total of the values shown in Figure 12.1.12.

The cost of unavailability can be a powerful tool to use in preparing reliability improvement plans.

Date: ________________
Amount: ________________
Process unit type: ________________

Fig 12.1.11 • Daily product revenue for a process unit

- Lost product revenue × days forced outage
- Maintenance costs
- Replacement part cost
- Labor cost
- Unnecessary turnaround time*

*Assumes process unit start-up is delayed by activity

Fig 12.1.12 • The cost of unavailability critical rotating equipment (per year)

Optimizing reliability

The key to reliability improvement is to build a solid program foundation. Figure 12.1.13 shows the reliability pyramid.

The success or failure of any reliability improvement program directly depends on obtaining and maintaining management support. Figure 12.1.14 presents guidelines for meeting this important objective.

Input data

Once management support is obtained, input data forms the foundation of the program. Figure 12.1.15 presents important guidelines concerning input data.

The environment or surroundings for any piece of rotating equipment play an important part in determining the availability of that particular item (refer to Figure 12.1.16).

This figure shows that the rotating equipment environment is the process unit in which the equipment is installed. If any of these

Fig 12.1.13 • The Reliability Pyramid

1. Clearly stating impact of problem on plant profit (cost of unavailability)
2. Preparing a brief statement of:
   - Problem
   - Impact on plant
   - Reliability improvement plan
3. Being confident!
4. Being professional
5. Being autonomous (do not expect management todoy our job!!)
6. Providing timely updates

Fig 12.1.14 • Obtain and maintain management support by the above

items are not taken into account, the accuracy of the conclusions reached during the assessment phase will be significantly reduced.

In my experience, most failures in predictive maintenance and troubleshooting exercises occur because the entire system in which the component operates is not considered. Every component in every piece of machinery operates in a system. Defining the system and all of the components contained therein is a very important step in successful problem analysis. Refer to Figure 12.1.17.

Experience counts!

Having experienced analysts to determine the root causes of low reliability is the next step in building a strong program.

- Include all the facts (operation, reliability, maintenance failure analysis, etc.)
  - Consider the machinery environment
  - Consider the entire system
- Only use proven data (don't guess!)
- Accuracy is most important – confirm data is correct

Fig 12.1.15 • Reliability input data

- Process condition change
- Piping and foundation change
- 'Unit' (driven, driver, transmission, auxiliaries)
- Ambient conditions

Fig 12.1.16 • The rotating equipment environment

- Think system!
- Every component is a part of a system
- In order to determine root causes, systems and not just components must be considered

Fig 12.1.17 • The concept of a system

Figure 12.1.18 suggests ways to build and develop a practical, strong analyst group.

Utilize practical, correlated assessment techniques whenever possible.

Today, many statistical methods are available to the analyst to determine causes of failure and to predict equipment and component life. The personal computer makes the use of these methods quick and easy.

However, the reader is cautioned to regard all statistical methods as only a part of the process. Whenever possible, actual data concerning failure rates should be used and the correlation of statistical methods should be defined. It should always be remembered that the basis for most statistical methods have evolved from industries where 'production components' are used, i.e. the electronics and automotive industries. However, the rotating equipment unit, regardless of type, always becomes customized by virtue of its environment. That is, each rotating equipment unit has its own signature. Consequently, care must be exercised when applying statistical methods to rotating equipment reliability assessment. Figure 12.1.19 presents this important fact.

- Select experienced rotating equipment personnel
- Ideally, design and field experience are the best combination
- Provide site specific training
- Measure results
- Provide opportunities for networking with other specialists within and outside the company:
  - User groups
  - Industry conferences
  - Regional conferences
- Include analysts in all phases of new projects

Fig 12.1.18 • Analyst development strategy

Since every rotating equipment unit, regardless of size, represents a 'customized system', care must be exercised when assessing the results obtained by statistical methods.

Fig 12.1.19 • Statistical methods and rotating equipment

Best Practice 12.2

Use site PDM results (predictive maintenance program) to continuously improve machinery best practices to maintain a site machinery reliability program of the highest level.

Continuous improvement means increasing the effectiveness of the established site best practices by monitored results.

An effective site PDM program using the principles of component condition monitoring as defined in this book (See B.P. 10.11 and 11.7) will identify improvements that can be made to further increase site machinery safety and reliability.

Use B.P. 12.1 to justify continuous improvement recommendations to management and obtain management approval for their implementation.

Lessons Learned

Failure to continuously improve site machinery reliability best practices will result in reduced reliability.

The following two examples show the results of not increasing the effectiveness of established site best practices:

a. One accumulator was installed in a critical compressor train (unspared) lube oil system to prevent a low oil pressure trip. The unit tripped on low oil pressure while the accumulator was being checked for pre-charge and bladder condition on a quarterly basis. After this incident, the quarterly checks of the accumulator were canceled only to cause an additional shutdown. Each shutdown resulted in revenue losses of over $1 MM. The adopted 'continuous improvement best practice' was to install an additional accumulator at the next turnaround and continue the quarterly accumulator checks (one at a time). See B.P. 7.14.

b. A worn steam seal in a large steam turbine/compressor train allowed steam condensate to enter the oil system requiring an oil conditioner to be installed and several on line oil changes to prevent a shutdown of the unit. This unit was supplied with a gland condenser vacuum gauge that was continuously monitored and indicated vacuum eductor wear. This issue was not corrected since the single eductor could not be shut down for corrective measures (orifice replacement). The adopted 'continuous improvement best practice' was to install an additional gland condenser vacuum eductor at the next turnaround. See B.P. 5.9.

Benchmarks

This best practice has been used since the mid 1990s as 'continuous improvement recommendations' to increase the effectiveness of site best practices. These recommendations are included in all site audit and site specific training workshop reports. Clients have always acknowledged that these 'continuous improvement recommendations' have significantly increased site machinery safety and reliability.

B.P. 12.2. Supporting Material

Refer to the following best practices for effective execution of this best practice:

- B.P: 5.9
- B.P: 7.14
- B.P: 10.11
- B.P: 11.7
- B.P: 12.1

Best Practice 12.3

Identify and execute the best practices in this book with the highest potential for site safety, reliability improvement and increased product revenue to ensure the plant machinery reliability program will be 'best of the best'.

Review all of the over 200 machinery best practices contained in this book and prioritize them according to site safety and reliability enhancement.

Present these best practices to management using the guidelines presented in B.P. 12.1.

Lessons Learned

Plant machinery safety and reliability can always be improved. Failure to convert plant, company and industry lessons learned into site best practices will result in lower plant machinery safety and reliability.

Benchmarks

All of the machinery best practices are recommended to all clients globally. The execution of the best practices in this book has resulted in significant improvements in site machinery safety and reliability.

B.P. 12.3. Supporting Material

Each best practice in this book.

Best Practice 12.4

Establish a continuous site specific, machinery function awareness training program for operations, maintenance and engineering personnel, to ensure implementation of site predictive maintenance practices of the highest level.

Identify opportunities for machinery awareness by reviewing the site bad actor list (more than one failure per year).

Design a continuous site machinery awareness training program prioritizing the subjects on the basis of safety, reliability and lost revenue.

Prepare agendas for each training workshop and use vendor, site specialists, corporate specialists and/or consultants that specialize in site-specific and not generalized machinery training workshops (Forsthoffer Associates or equal).

Lessons Learned

Plants that do not consistently offer site specific machinery workshops experience lower machinery safety and reliability. The cost of site-specific machinery workshops can be justified by the savings in lost critical machinery revenue.

FAI has been presenting site-specific machinery workshops since 1990. We recommend that our website be visited and any of our clients be contacted for their impression of the benefit of our site-specific workshops. Available workshops are:

- Machinery Best Practices
- Machinery Reliability Optimization
- Principles of Rotating Machinery
- Pump
- Compressor
- Steam Turbine
- Gas Turbine
- Gear
- Auxiliary Systems
- Component Condition Monitoring

All workshops use site specific data and include classroom exercises and site visits to ensure implementation of learned principles.

Benchmarks

Since 1990, Forsthoffer Associates has presented site-specific machinery workshops globally, to more than 8,000 attendees. These workshops have increased clients machinery safety and reliability by presenting machinery functional awareness information that has enabled attendees to implement the best practices which apply to their plant.

B.P. 12.4. Supporting Material

Contained in this section are two agendas from the most popular site-specific workshops we have presented since 1990:

- Principles of Rotating Machinery
- Machinery Reliability Optimization

All of our workshops are structured on existing site reliability issues, include site-specific machinery documents (secrecy agreements signed) and site visits to ensure implementation of the learned principles.

A detailed workshop report is issued no later than two weeks after site presentation and contains the following information:

- Attendee evaluation results of the workshop
- Instructor evaluation of the attendees
- Specific instructor evaluation of the understanding of each presented principle (Training Module) and recommendations to ensure implementation
- Recommendations for Continuous Improvement based on issues raised by the attendees and discussed during the workshop

Table 12.4.1 Workshop Agenda
Course A

FORSTHOFFER ASSOCIATES INC.
The Crossings (215) 321-1907
1098 Washington Crossing Road — Suite 4 (215) 321-1908
Washington Crossing, Pennsylvania 18977 FAX (215) 321-1909

SITE ROTATING EQUIPMENT FUNCTION OVERVIEW WORKSHOP AGENDA

Notes:
∗ refers to the Chapter of the book to be used for the Workshop Volume 1 of Forsthoffer's Machinery Handbook
∗∗ refers to the appropriate Tab in the supplementary Workshop Manual

	SESSION	DESCRIPTION	CHAPTER*	TAB** MODULE
DAY 1	1	COURSE INTRODUCTION	1	145.20
	2	ROTATING EQUIPMENT OVERVIEW	1	107
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	7, 8	EFFECT OF THE PROCESS ON POSITIVE DISPLACEMENT AND DYNAMIC EQUIPNMENT AND COMPONENT CONDITION MONITORING	3	136
DAY 2	9, 10	PUMP PERFORMANCE CURVES AND DATA	5	4 214
	11	THE CONCEPT OF PUMP HEAD	5	215
	12, 13	HYDRAULIC DISTURBANCES	6	217
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	26-28	SEAL SYSTEM DESIGN	17	7 712715
	29	RECIPROCATING COMPRESSOR MAJOR COMPONENT FUNCTIONS	18	6 570
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	31-32	TYPES OF STEAM TURBINES ON SITE	2	8 313

(Continued)

Table 12.4.1 Workshop Agenda

	SESSION	DESCRIPTION	CHAPTER*	TAB** MODULE
DAY 5	33, 34	STEAM TURBINE PERFORMANCE CHARACTERISTICS	21	302
	35, 36	STEAM TURBINE MECHANICAL DESIGN	22	303
	37	STEAM TURBINE INLET STEAM REGULATON	23	314
	38, 39	STEAM TURBINE CONTROL AND PROTECTION SYSTEMS	24	315
	40	STEAM TURBINE OPERATION	25	316

Course B
FORSTHOFFER ASSOCIATES INC.
The Crossings (215) 321-1907
1098 Washington Crossing Road — Suite 4 (215) 321-1908
Washington Crossing, Pennsylvania 18977 FAX (215) 321-1909

ROTATING EQUIPMENT RELIABILITY OPTIMIZATION WORKSHOP
Notes: Tab = Tab in Supplementary Manual
Chapter = Chapter in Forsthoffer's Machinery Handbook Volume 5

1.6	SESSION DESCRIPTION	CHAPTER*	TAB** MODULE
1.6.1	DAY 1		
1, 2	WORKSHOP INTRODUCTION & OVERVIEW	1	1 112.01
•	INTRODUCTION		1 12.02
•	WORKSHOP OBJECTIVES		
•	INSTRUCTOR & STUDENT INTROS		
•	WORKSHOP AGENDA		
•	FORMAT & SCHEDULE		
•	BENEFITS & DUBAI PETROLEUM CONTRIBUTIONS		
•	FUNDAMENTALS & RELIABILITY TERMS		
3-8	THE CAUSES OF MACHINERY FAILURES (5 WHYS)	2	2 927
•	“FAILURES ARE NOT RANDOM”		
•	PROCESS CONDITION CHANGES		
•	INSTALLATION ERRORS		
•	OPERATING PROCEDURES		
•	DESIGN PROBLEMS		
•	COMPONENT WEAROUT		
•	CLASSROOM EXERCISE		
1.6.2	DAY 2		
9-14	ROOT CAUSE FAILURE ANALYSIS PROCEDURE	5	3 942
•	INTRODUCTION		
•	PROCEDURE OVERVIEW		
•	SPECIFIC ANALYSIS DETAILS		
•	CLASSROOM EXERCISES 1, 2, 3, & 4		
DAY 2			
15, 16	RCFA EXAMPLE	6	4 943
•	INTRODUCTION TO PROBLEM		
•	PROBLEM — CLASS & INSTRUCTOR		
DAY 3			
17-24	DESIGN SCREENING EXAMPLES (ANALYSIS TECHNIQUES)	7	5 960
•	ROTORS — THE EFFECT OF PROCESS HEAD		136,515
•	JOURNAL BEARINGS & VIBRATION		126
•	THRUST BEARINGS & BALANCE DRUMS		132
•	PUMP MECHANICAL SEALS		220
•	COMPRESSOR SEAL SYSTEMS (LIQUID AND DRY GAS)		712,713,714,715
•	AUXILIARY SYSTEMS		701
DAY 4			
25-27	ROOT CAUSE FAILURE ANALYSIS EXERCISES		6 112.03
	NOTE: CLASS WILL BE DIVIDED INTO 4 GROUPS AND EACH ASSIGNED AN EXERCISE		
•	FACT FINDING — FROM SITE INFORMATION		
•	PACKAGE & COMMUNICATION WITH PLATFORMS		

Table 12.4.1 Workshop Agenda

1.6	SESSION	DESCRIPTION	CHAPTER*	TAB**	MODULE
28-32	CLASSROOM WORK ON ASSIGNED EXERCISES			6	112.03
•	EQUIPMENT KNOWLEDGE				
•	DEFINING ABNORMAL CONDITIONS				
•	LISTING ALL POSSIBLE CAUSES				
•	ELIMINATING NON-RELATED CAUSES				
•	STATING ROOT CAUSE OF THE PROBLEM				
•	DEVELOPING AND IMPLEMENTING FINAL ACTION PLAN				
DAY 5					
33-37	REVIEW OF RCFA EXERCISES			6	
•	GROUP PRESENTATIONS				
•	GROUP ACTION PLANS				

Communication Best Practices

Best Practice 12.5

Follow these email effectiveness guidelines for optimum results:

- Accurately state the subject of the email.
- Explain reasons for all recommendations and benchmark (give references where recommendation has been successfully implemented).
- Attach a formal letter, containing comments and recommendations, whenever necessary.
- Proofread carefully.
- Always include previous emails on the subject in the email response.
- If the previous email requires comments, note your comments in bold and color code if necessary.
- Respond to all persons included in the email, and copy any additional recipients that will be involved in the contents contained in the reply.

Using the above guidelines will ensure accurate communication that will produce immediate responses.

Lesson Learned

Incomplete and inaccurate emails can be the root causes of reliability issues, since they can lead to misinterpretation of recommendations and result in incorrect action plans.

Rapidly issued emails without proper details have caused machinery reliability problems and in some cases component failures.

Benchmarks

The above best practice has been in use since the mid-1990s to ensure correct interpretation of comments and recommendations. This approach has resulted in improved plant safety and reliability from all correspondence.

Best Practice 12.6

Follow these guidelines for email recipients – who to send to, copy and blind copy:

- Reply only to the sender.
- Copy those noted in received email.
- Copy additional personnel who will be involved with details contained in the reply.
- Blind copy anyone who will be involved with this issue in the future.

Lessons Learned

Including email recipients not directly concerned with the issue will delay action due to the additional questions and/or comments involved. Not including all of the proper recipients can result in additional emails being required for corrections and/or for additional action.

Benchmarks

I have followed this best practice since the mid-1990s to ensure that all of the personnel connected with the issue at hand are notified.

Best Practice 12.7

Follow these proven guidelines to produce effective meeting agendas:

- Define the meeting subject to accurately state the contents of the meeting.
- Define the meeting objective in clear concise terms.
- Define the participants required from participating parties (vendor, contractor, 3rd party consultant, etc.).
- Outline detailed contents of the meeting including sections for each separate topic.
- Issue the agenda to all concerned parties no later than two weeks before the scheduled meeting.

Observing the above guidelines will ensure a meeting of optimum effectiveness.

Lessons Learned

Experience shows that meetings conducted without a detailed agenda sent well in advance and without the attendance of all required parties (proper discipline and experience) are ineffective and result in additional meetings.

Noted below are some examples of poorly planned meetings (have you been to one?):

- Agenda not issued.
- Agenda issued a few days before or on the day of the meeting and not received by all parties.
- Meeting objective not stated.
- All the required participants not in attendance.

Benchmarks

I have followed this best practice since the late 1960s, when I was first responsible for setting up technical meetings. This approach has resulted in effective meetings that have produced desired results in the minimum of time and a respect for the management of the meeting shared by all participants.

Best Practice 12.8

Follow these proven guidelines to produce meeting minutes of optimum effectiveness that will result in implementation of all action items:

- Define who will be responsible for recording and issuing the minutes.
- Require that the person recording the minutes will take the minutes simultaneously using a computer.
- Use the meeting agenda to record the minutes – type the minutes from each item noted on the agenda.
- Accurately record the name, title and email address of each participant.
- Define person or company responsible for each action point and the date required for action to be completed.
- Review the meeting minutes, using an LCD projector, prior to the conclusion of the meeting and have all parties agree and sign their agreement.
- Immediately issue, via email, agreed official minutes to all participants.
- Meeting recorder to follow up with all individual assigned action items to ensure execution on or before the required date.

Observation of all of the above items will ensure desired results including implementation of all action items at the desired time.

Lessons Learned

Failure to follow the above guidelines has resulted in ineffective meetings that have not produced the desired results and have led to additional meetings.

Noted below are some examples of poor meetings (have you been to one?):

- Agenda not issued in advance and either handed out or not used.
- Meeting attendee list not taken.
- All action items, person responsible and required date not defined.
- Meeting minutes not recorded during meeting and not reviewed at end of meeting.
- Meeting minutes never issued.
- Action items not followed up.

Benchmarks

I have followed this best practice since the late 1960s (without a computer until the 1990s), when first responsible for conducting technical meetings. This approach has resulted in effective meetings that have produced desired results in the minimum of time and a respect for the management of the meeting shared by all participants.

Best Practice 12.9

Follow these proven guidelines for site audits and/or visits to prepare effective reports that will produce immediate results and rapid implementation of all recommendations:

- Prepare and email daily activity reports containing all observation and recommendations with supporting information and benchmarks for all recommendations.
- Email weekly executive summaries to management if visit extends beyond one week.
- Issue the final report in executive summary format, with recommendations and action items clearly defined.
- Reference all details in each item to daily reports, calculations, specific documents, etc.
- Include all daily reports, referenced documents and calculations in the appendix of the report.
- Prepare an appendix index to identify the location of each appendix item.
- Ensure that the required number of hard copies of the final report are received no later than two weeks after completion of the assignment.

Inclusion of the above items in all reports has resulted in projects of the highest degree of success and clients who continuously use our services.

Lessons Learned

Failure to issue easy to read, detailed technical reports in a timely fashion has resulted in poor recommendation implementation and in some cases machinery failures.

Noted below are some examples of poor reports (have you received one?):

- Detailed reports that do not reach a specific conclusion.
- Reports that do not contain sufficient details concerning each recommendation.
- Recommendations that do have benchmarks to show where the recommendations have been successfully implemented.
- Reports issued long after work is completed.

Benchmarks

I have followed this best practice since the mid 1970s when I was first responsible for issuing reports for the corporate engineering center of a major global oil, gas and chemical company. This approach, since that time, has resulted in high implementation rates (greater than 50%) and significant improvements in plant safety and reliability.

S • E • C • T • I • O • N • 1

MAINTENANCE OF MECHANICAL EQUIPMENT

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CHAPTER 1
PLAIN BEARINGS

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GENERAL

Plain or sleeve bearings are designed to support shafts that rotate, oscillate, or reciprocate. Though seemingly simple, and certainly one of the least expensive of mechanical parts, sleeve bearings are highly engineered components. They range from porous self-lubricated powder metal parts only a fraction of an inch in diameter to stationary power plant bearings, which often exceed 18 in. in diameter.

With few exceptions, sleeve bearing lubrication is hydrodynamic; i.e., during operation, the shaft floats on a thin film of the lubricant. Because of this, friction and wear are minimized. However, it is important to realize that this so-called minimum film thickness is *not* the same as the bearing clearance. While the latter may be up to several thousandths of an inch, the minimum film thickness is typically on the order of one ten-thousandth of an inch. Nevertheless, sleeve bearings can have an almost unlimited life, provided proper maintenance practices are followed. When replacement does become necessary, following proper refurbishing and assembly procedures will assure extended life of the replacement parts.

PREVENTIVE MAINTENANCE

Lubricant Supply. Proper bearing design and material are necessary to achieve long service life but are not by themselves sufficient. The lubricant is the key component of the system which determines bearing life. Reduced to simplest terms, if a sleeve bearing is provided with an adequate flow of the proper *clean* lubricant, long life should be realized.

Lubricant flow to the bearings is a function of the equipment design. Oil pressure at specified speeds should be within the limits given by the equipment builder. Lower values suggest worn bearings. In this case, replacement should be made as soon as is feasible. Excessive pressures indicate a blockage or restriction somewhere in the system. This should be investigated immediately. The oil level also should be checked on a routine basis to avoid pump cavitation and subsequent oil starvation. In nonpressurized lube systems, reservoirs should be checked on a regular schedule to ensure that adequate oil is always present. Wick-fed bearings, such as those in fractional horsepower electric motors, should be lubricated periodically according to the schedule called for by the manufacturer.

Cleanliness. Sleeve bearings simply cannot survive without adequate lubrication. Once this is assured, the next most important consideration is the cleanliness of the lubricant. Since minimum film thickness is so small, the presence of oil-borne debris can greatly accelerate the wear process. If foreign materials such as metal chips and abrasives are large and numerous, bearing failure can occur rapidly. It is therefore of the utmost importance to change the lubricant in accordance with the equipment builder's recommendations. The lubricant filter, if one is used, also must be replaced according to schedule. The air filter, if one is present, should be serviced at recommended intervals, for airborne contamination is a primary source of vitreous abrasives that find their way into the oil. If the equipment is operated for extended periods in dirty or dusty environments, more frequent lubricant and filter changes should be adopted. Usually the equipment builder has established recommended change frequencies under these conditions. If not, a good practice is to make changes at intervals from one-third to one-half of that normally recommended.

Lubricant contamination can occur in storage as well. However, simple good housekeeping, such as covering open containers and reservoirs tightly to exclude dirt and water, and keeping anything which contacts the lubricant (oil can, funnels, etc.) as clean as possible will prevent problems.

Lubricant Type. Ensuring an adequate flow of clean lubricant makes long bearing life possible but does not guarantee it. The oil must be the proper one for the application. From a bearing performance viewpoint, lubricant viscosity is the most important parameter. Lower-viscosity (i.e., thinner) oils reduce oil film thickness. This increases the wear rate and can possibly lead to failure. It is critical that the equipment manufacturer's lubricant recommendations be followed.

In addition, the proper combination of oil additives is necessary to prevent rapid breakdown, thickening, foaming, and sludging. All these effects can lead to bearing failure, as well as to the damage of other components. Failure to use the recommended lubricant can have dire consequences and, in most cases, voids the equipment warranty. Extended drain intervals should not be adopted without a strictly monitored oil analysis program.

BEARING MATERIALS

Requirements

Surface Action. Sometimes referred to as *slipperiness* or *compatibility,* surface action is the ability of a material to resist seizure when contacted by the shaft. Contact takes place every time the equipment is started or stopped and can also occur during momentary overloads.

Embeddability. The ability of a material to absorb foreign particles circulating in the oil stream is referred to as embeddability. Some particles will go unfiltered, so the material must be soft enough to ingest them.

Conformability. The material also must be soft enough to creep or flow slightly to compensate for the minor geometric irregularities which are present in every assembly. These include misalignment, out-of-round, and taper.

Fatigue Strength. This is the ability of a bearing material to withstand the loads to which it is subjected without cracking. Bearings should not fatigue prior to the normally scheduled overhaul.

Temperature Strength. As operating temperatures increase, bearing materials tend to lose strength. This property indicates how well a material carries a load at elevated temperatures, without breaking up or flowing out of shape.

Thermal Conductivity. Shear of the oil film by the shaft generates significant heat, most of which is carried away by the oil. Nevertheless, it is important for the bearing to transfer heat rapidly from its surface through its back to avoid overheating and resultant reduction in life.

Corrosion Resistance. Oils oxidize with use, and the products of this degradation can be corrosive. Blow-by products and fuel or coolant contamination of the oil also promote a corrosive environment. Bearing materials should be resistant to these effects.

Construction

Most hydrodynamic bearings are metallic, primarily for reasons of thermal conductivity. They may consist of one, two, or three layers.

Monometals. Bearings made from a solid bar or tubes of an aluminum or bronze alloy have been available for a number of years. They are generally used where loads are not very high. In order to have the same rigidity as a bearing with a steel back (see below) and to avoid yielding at operating temperature, they are made with a comparatively thick wall. As a result, they require a larger housing bore.

Bimetals. A bimetal bearing has a steel back, to which is bonded a liner of Babbitt, copper-lead, or aluminum. Babbitts are soft alloys of lead or tin, with additives such as copper, antimony, and arsenic. They have outstanding embeddability, conformability, and surface action but relatively low fatigue strength. Copper-leads and aluminums are harder than Babbitts and have much better strengths, but at a sacrifice of the other properties.

Trimetals. In order to achieve the desirable surface properties of a Babbitt bearing and the strengths of harder materials, the trimetal bearing was developed for heavy-duty applications. In this construction, a thin (usually about 0.001-in.) layer of a soft material is either electroplated or cast onto the copper-lead or aluminum layer of a bimetal. The surface layer (overlay) imparts the desired "soft" properties to the bearing; however, because it is so thin, it derives improved fatigue strength from the intermediate layer; i.e., it is much stronger than a thick layer of the same soft alloy.

Table 1.1 shows the compositions of some of the more popular plain bearing materials in use today.

TABLE 1.1 Bearing Alloys

SAE No.	Nominal composition, %							Bearing construction
Babbits	Pb	Sn	Sb	Cu	As			
12		89	7.5	3.5	—			Bimetal
15	83	1	15	—	1			Bimetal
191	90	10	—	—	—			Plated overlay
192	88	10	—	2	—			Plated overlay
Copper base	Cu	Pb	Sn	Zn				
49	75	24	0.5	—				Trimetal
791	88	4	4	4				Monometal
792	80	10	10	—				Bimetal
793	84	8	4	—				Bimetal
794	72	23	3	—				Bimetal
Aluminum base	Al	Sn	Cu	Ni	Si	Cd	Pb	
770	92	6	1	1	—	—	—	Monometal
780	91	6	1	0.5	1.5	—	—	Bimetal/trimetal
781	95	—	—	—	4	1	—	Trimetal
782	95	—	1	1	—	3	—	Trimetal
787	85	1.5	1	—	4	—	8.5	Bimetal

DESIGN

In order to do a better job of maintaining bearings, it is helpful to have an understanding of certain key bearing design factors. Although this discussion centers around half-shell bearings, much of it applies to full round parts (i.e., bushings) as well.

An assortment of bushings, flanged and straight-shell bearings, and thrust washers is shown in Fig. 1.1. Standard nomenclature for the various aspects of bearing design can be seen in Figs. 1.2 to 1.4.

There are two types of insert bearings: precision and resizable. The former is made to precise tolerances and can be installed without modification. A resizable part is manufactured with an extra-thick layer of bearing material on the inside diameter (I.D.). This permits machining to any desired size.

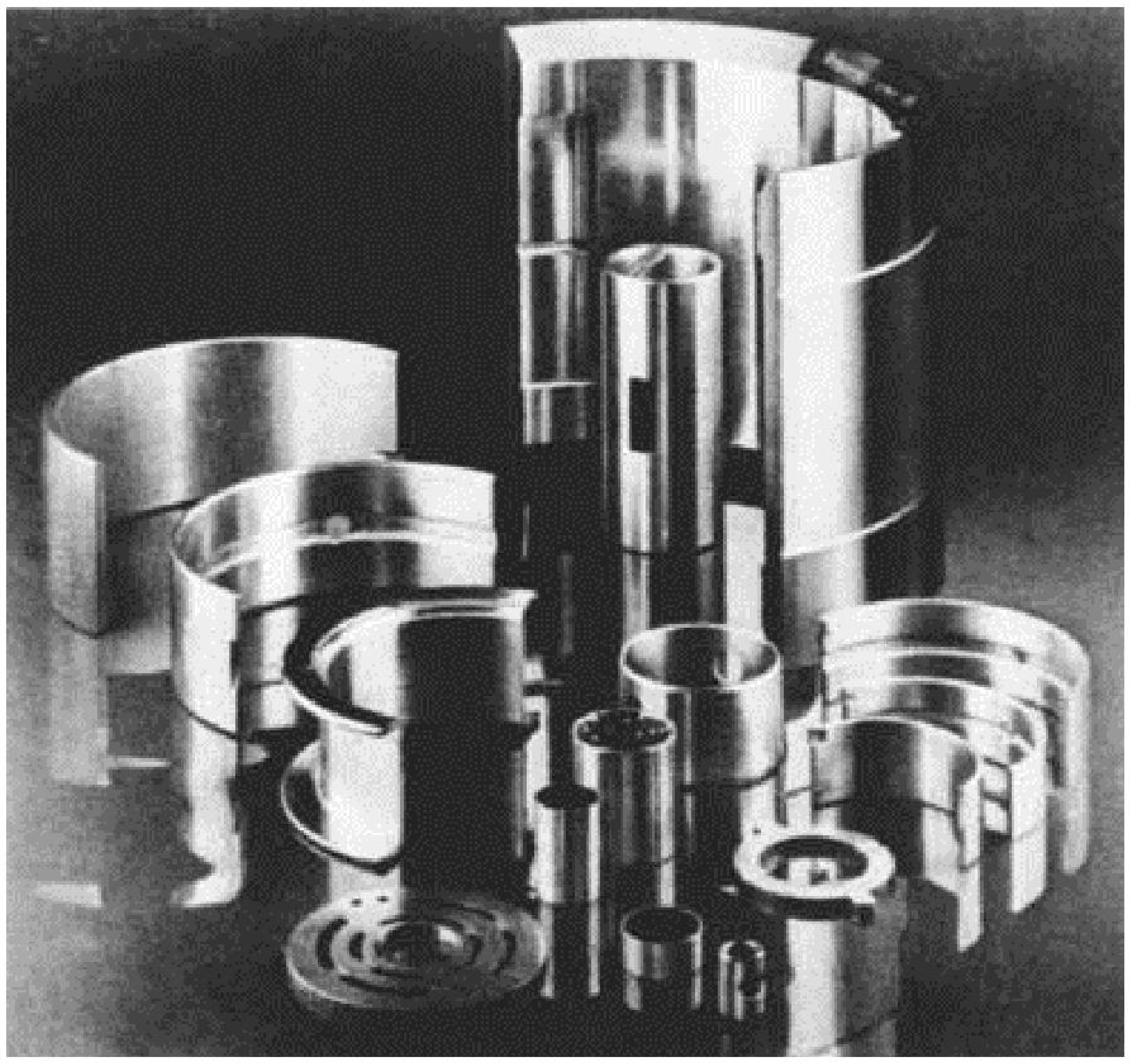


FIGURE 1.1 A sampling of bearings, bushings, and thrust washers.

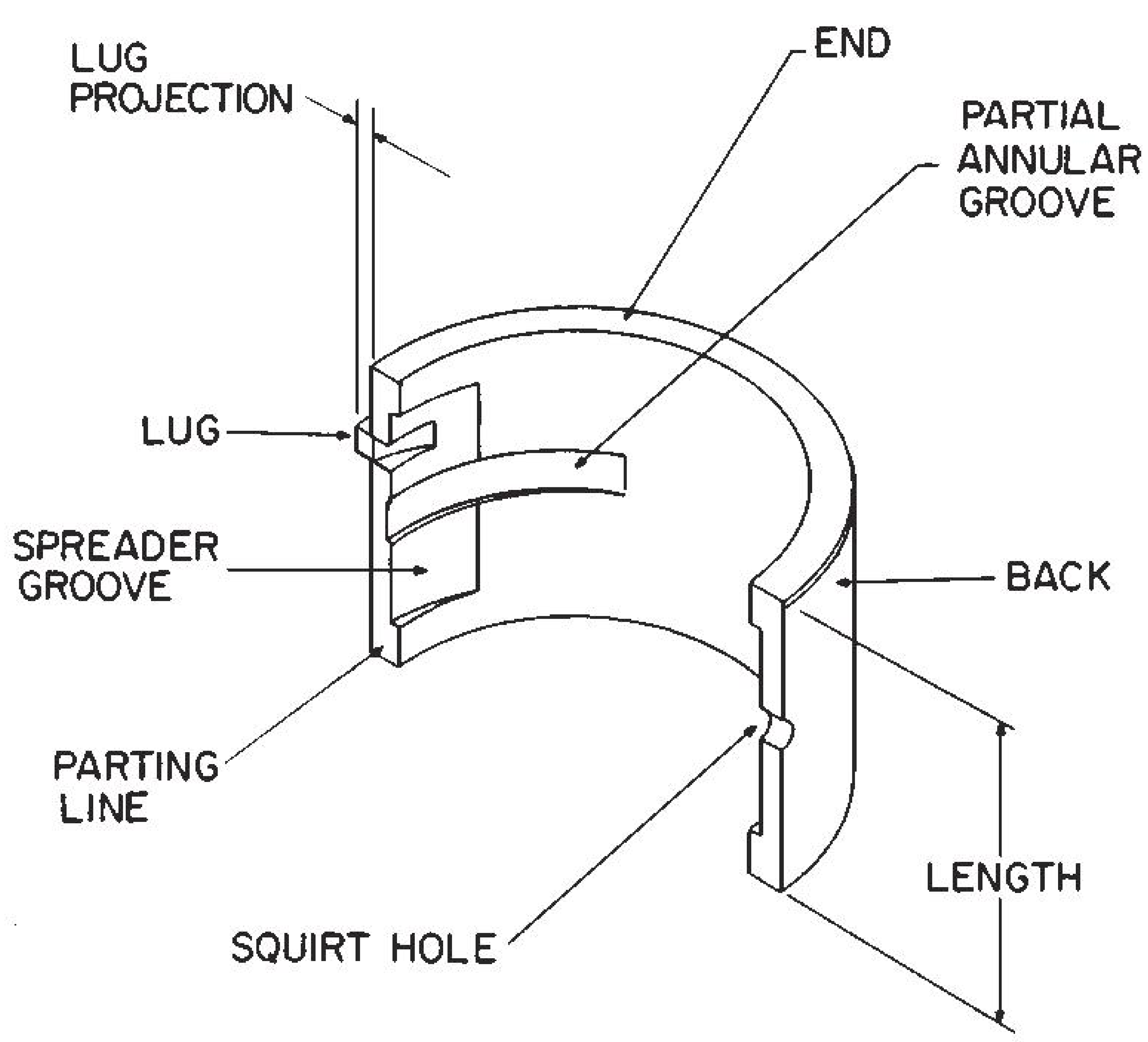


FIGURE 1.2 Bearing nomenclature.

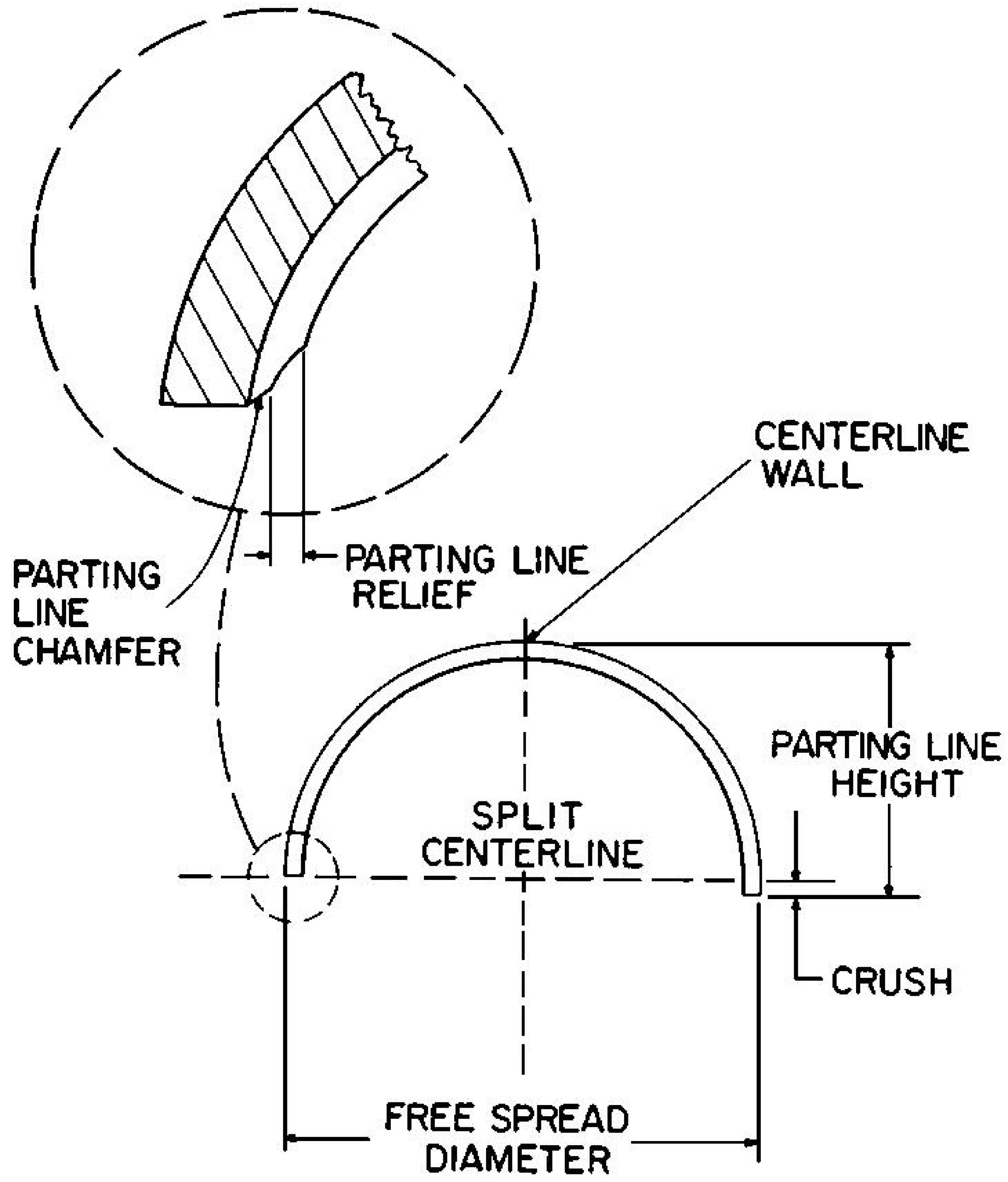


FIGURE 1.3 Bearing nomenclature.

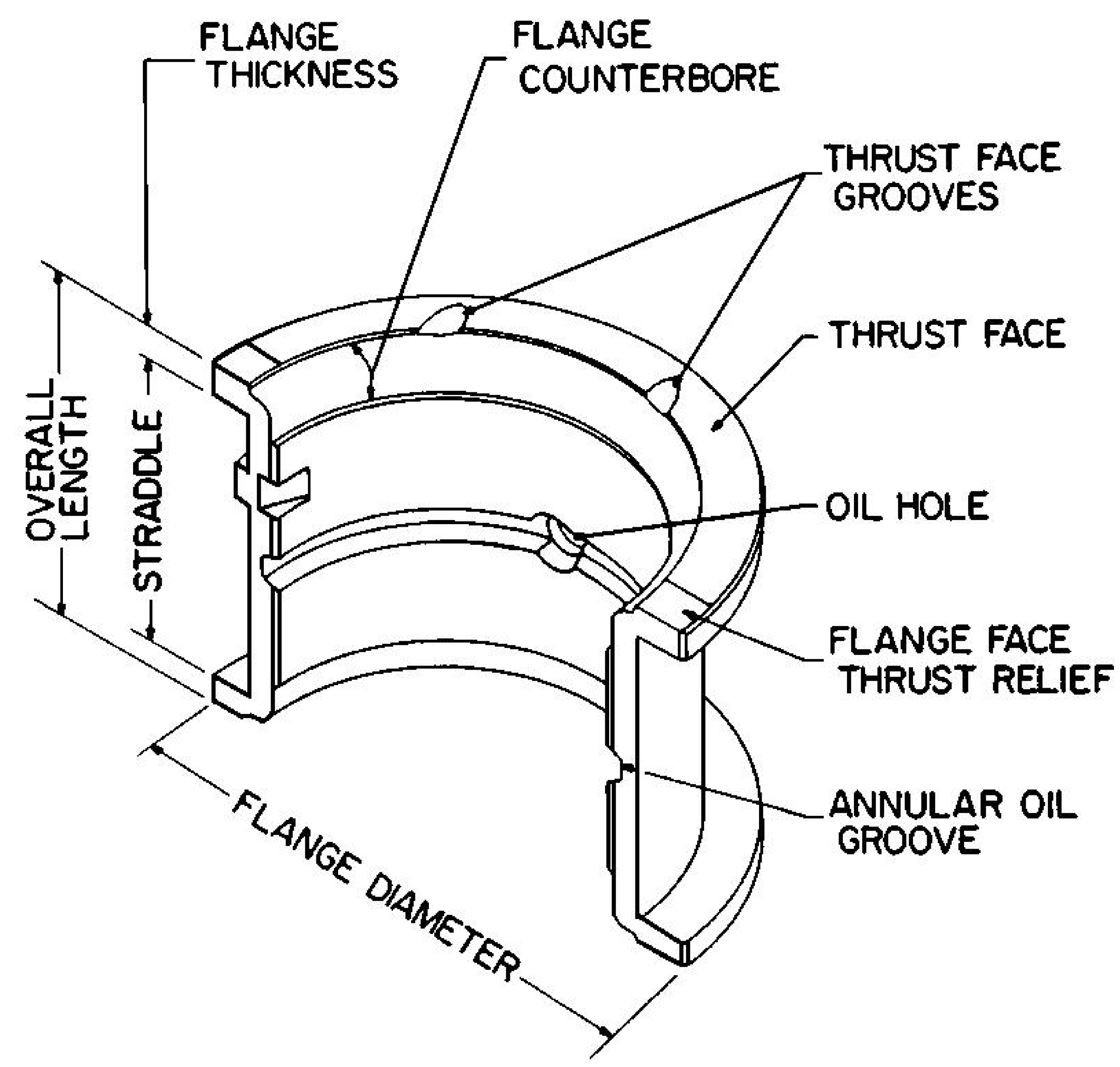


FIGURE 1.4 Flange bearing nomenclature.

Location and Retention. In order to keep a bearing from shifting sideways during installation and to ensure that its axial position is correct (i.e., so that it doesn't interfere with shaft fillets), a locating lug is formed at the parting line (Fig. 1.2). Another means used less frequently is a dowel in the housing, which protrudes partially into a mating hole in the bearing.

The sole purpose of the design feature, referred to as *free spread* (Fig. 1.5), is to aid in bearing installation. Bearings are manufactured with the distance across the outside parting edges slightly greater than the housing bore diameter. To install the bearing, a light force must be used to snap it into place. Once installed, the bearing will stay in place because of the pressure of the free spread against the housing bore.

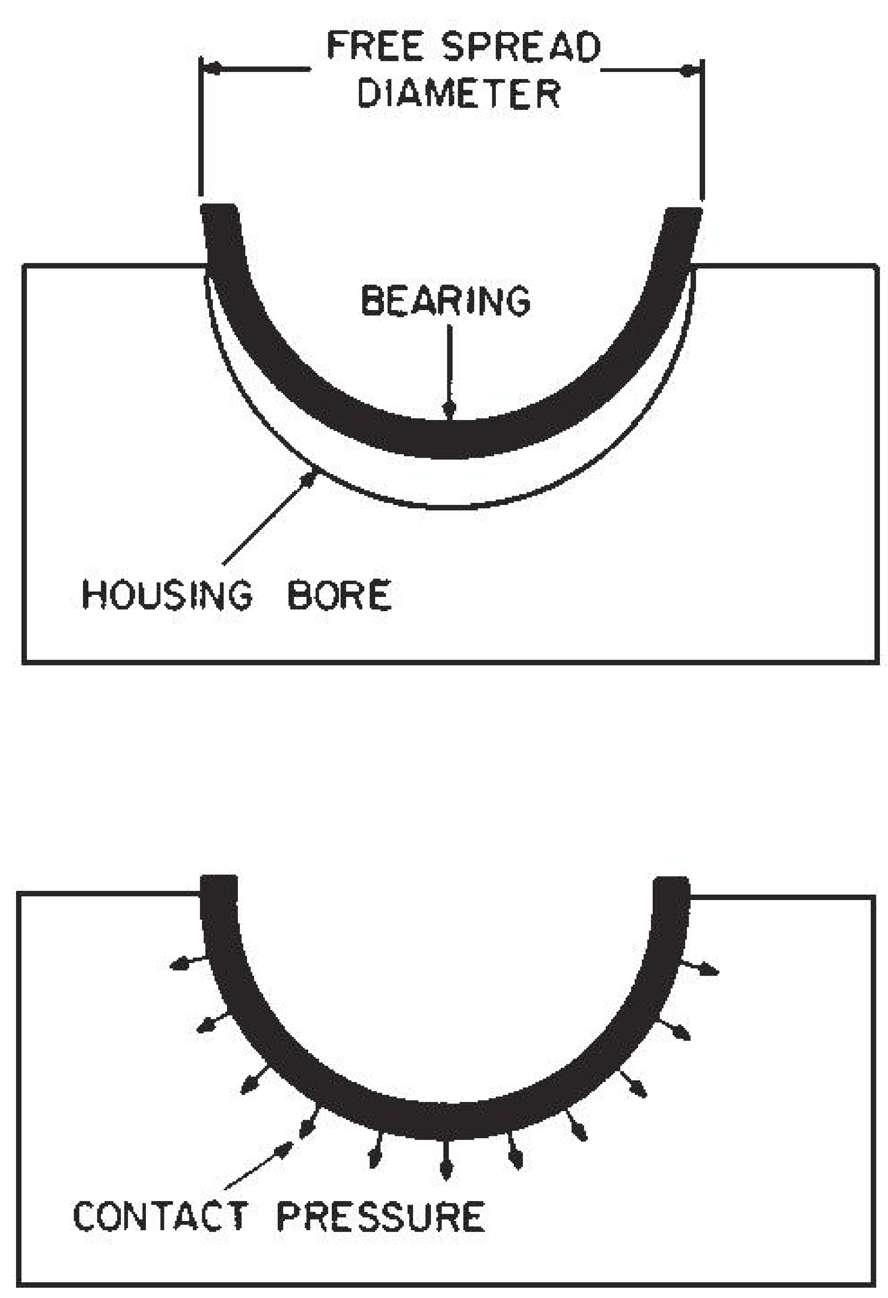


FIGURE 1.5 Free spread ensures bearing retention during installation.

It must be emphasized that neither the lug nor free spread will keep a bearing from spinning during operation. Such an action can cause a catastrophic failure and must be prevented. The means by which rotation is prevented is known as *crush* (Figs. 1.3 and 1.6). Bearings are manufactured so that they are slightly longer circumferentially than their mating housings. Upon installation, this excess length is elastically deformed ("crushed"), which sets up a high radial contact pressure between the bearing and housing. This ensures good back contact for heat conduction and, in combination with the bore-to-bearing friction, prevents spinning. *Under no circumstances* are the bearing parting lines filed or otherwise altered to remove the crush.

Lubrication. As indicated earlier, without adequate lubrication, particularly in the loaded area, a sleeve bearing will not work. Many bearings receive their oil supply through holes drilled in the housing. This necessitates a mating hole in the bearing to introduce oil to the clearance space. Sometimes holes are used to increase oil flow to nearby parts. Since the size and location of oil holes are critical to proper lubrication, replacement bearings must always be examined to ensure that oil holes match original equipment specifications.

Often a hole cannot adequately distribute oil to the loaded area of the bearing because the clearance space offers too much resistance to flow. In these cases, grooves are provided. Examples of typical grooves can be seen in Figs. 1.2 and 1.4. Grooves are used to aid flow both circumferentially (for distribution) and axially (for distribution, flushing debris, or lubricating adjacent parts).

Half-shell bearings are frequently manufactured with a circumferential taper of the wall (shown greatly exaggerated in Fig. 1.7). This *eccentricity* is typically less than 0.001 in. and serves two purposes. First, it increases the average clearance in the assembly without raising the noise level while the bearing is in operation. Tight vertical clearances can be run for noise control, while operating temperatures are kept at reasonable levels through the increased horizontal clearance. Second, additional clearance is often needed at the split line because of housing deflections that occur during engine operation; e.g., four-cycle engine connecting rod bores elongate in the vertical direction due to the tensile load on the exhaust stroke. Without eccentricity, oil flow could be pinched off at the split line.

Another feature used to assure the formation of a good oil film is parting-line relief (see Fig. 1.3). Because of manufacturing tolerances, it is almost impossible to guarantee that the upper and lower bearings will have the same wall thickness at the split line. The thicker bearing can act as a wiper, removing oil from the shaft and hindering formation of a good oil film. By employing parting-line relief this potential problem is avoided.

Parting-time chamfers (Fig. 1.3) are also used to avoid abrupt steps in the bearing I.D. surface which can disrupt the oil film. They are also used in conjunction with spreader grooves (see Fig. 1.2)—large dirt particles tend to become trapped in the grooves and are then flushed out the bearing ends through the chamfers.

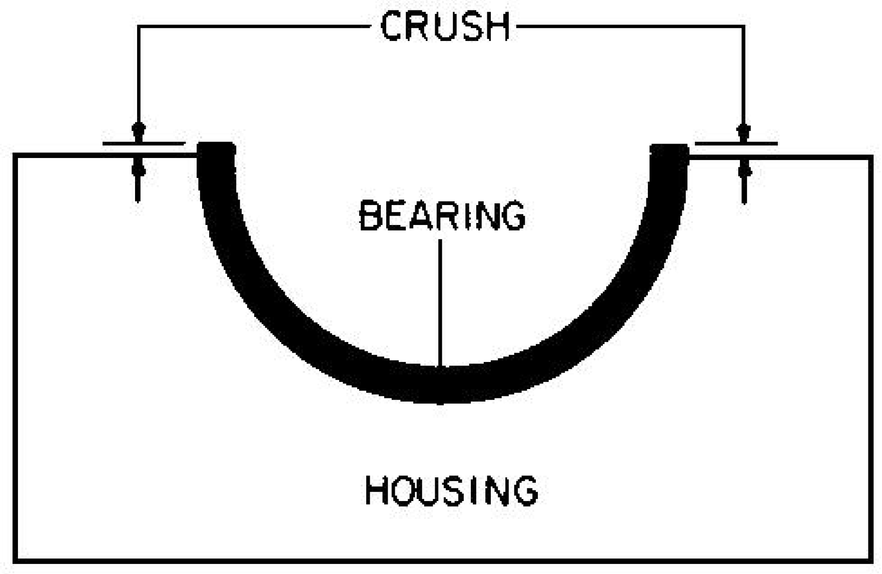


FIGURE 1.6 Crush prevents bearing rotation during operation.

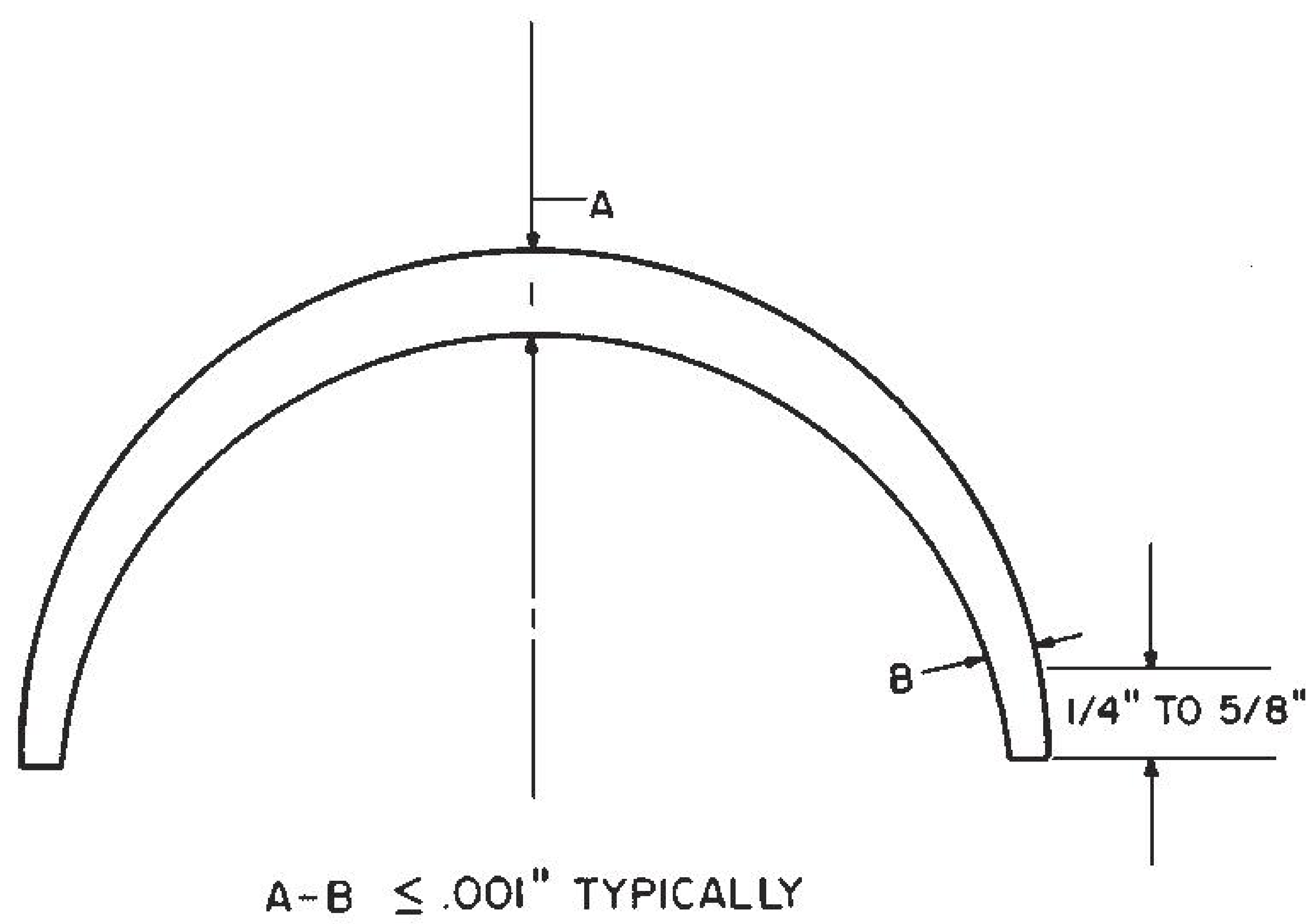


FIGURE 1.7 Bearing eccentricity.

INSPECTION AND RECONDITIONING

Bearings sometimes need to be replaced. Frequently, this is not because they simply wear out, but rather because they have been damaged in use and can no longer perform their intended function. If a damaged part is simply replaced without determining the cause of the problem, it is very likely that the replacement part will become damaged by the same cause. Although the discussion that follows is oriented toward engines, the principles apply equally well to any equipment having sleeve bearings.

Preliminaries

When disassembling an engine to examine the bearings, two things are of utmost importance. First, be sure that the work is being done in a clean area. Dirt is a mortal enemy of an engine's interior, particularly the bearings. The replacement job can be nullified if dirt enters the engine and remains there after the rebuild. Second, lay out all dismantled parts in an orderly fashion. Mark or identify each so that it can be reinstalled in its original position. Lay out the bearings as they were in the engine, i.e., with the main bearings in consecutive order, and with the connecting rod bearings placed between the same mains which flanked them in the engine. The following discussion should be helpful in describing various bearing conditions, their possible causes, and corrective actions to be taken to avoid a recurrence.

Analysis of Used Bearings

Normal Appearance and Wear. Most wear occurs during break-in, when minor geometric deviations are being accommodated. Thereafter, in a properly maintained engine, only those dirt particles too small to be filtered will be present to abrade the bearing surface. Two features usually mark normal wear. First, if the bearings are of trimetal construction, some of the overlay will have been removed, exposing the thin barrier layer between the overlay and the intermediate layer and possibly some of the latter as well. If of a bimetal construction, the surface will be noticeably burnished. There also may be minor surface scratches. These are generally not serious unless the intermediate layer has been deeply penetrated.

Dirt. Dirt is responsible for more bearing failures than any other mechanism. When dirt particles are large or numerous, they embed in the bearing lining, deforming the structure beneath and displacing the surrounding metal upward. The resulting high spot may be large enough to contact the journal. (A heavily embedded bearing will have numerous halos from this action.) Rubbing then creates heat which can, in conjunction with the stressed structure beneath, cause a rupture and removal of the bearing lining. If the particles embed only partially, the protruding portions will wear the shaft by a grinding wheel action. Figure 1.8 is an example of severe dirt embedment. Figure 1.9 shows a bearing which was badly scored by circulating particles which were too large to embed.

FIGURE 1.8 Bearing with severe dirt embedment.

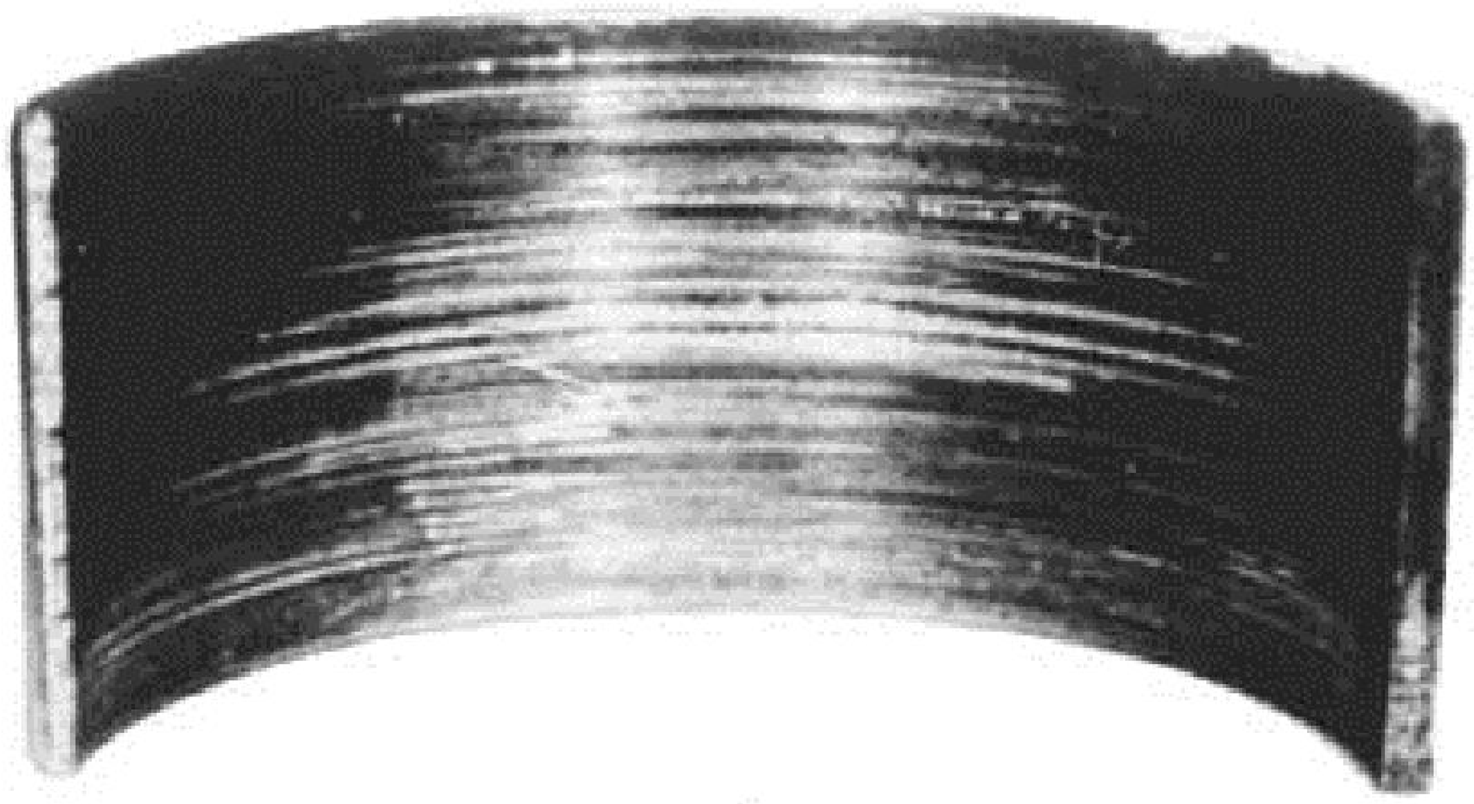


FIGURE 1.9 Bearing damaged by scoring.

FIGURE 1.10 Fatigue of bearing lining.

Causes of dirt contamination are improper cleaning of the engine and parts prior to assembly, road dirt and sand entering through the air-intake manifold, or wear (including failure) of other engine parts, causing small fragments to enter the oil supply. Poor maintenance practices are generally the root cause of the problem.

Corrective actions: (1) grind and polish the journal surfaces if necessary, (2) install new bearings, paying particular attention to cleaning procedures, and (3) change oil and filters at the intervals recommended by the engine manufacturer.

Fatigue. Generally speaking, bearing fatigue results when either the load or time in service exceed the alloy's capability. There are several possible causes: load concentrations due to dirt, poor shaft or bore geometry, misassembly of the bearing, material weakness caused by high-temperature operation or corrosion, or simply exceeding the bearing's normally expected life span.

Fatigue cracks initiate at the bearing surface and propagate perpendicular to it. Before reaching the steel, the cracks turn, run parallel to the steel, and join. The material can then flake out.

The most common type of fatigue is that of the overlay on trimetal bearings. But since the primary overlay functions are to absorb small dirt particles and provide a slippery surface for starting and stopping conditions, slight overlay fatigue is not regarded as a bearing failure. The load-carrying strength of a bearing is in its intermediate layer. A true fatigue failure involves the intermediate material rather than the sacrificial overlay.

Few bearings escape some degree of fatigue during normal operation. Premature failures occur in stages, beginning with normal "hen track" patterns. As fatigue progresses into the second stage, it takes on the classical "wormhole" appearance, shown in Fig. 1.10 for a lead-base Babbitt bearing.

Figures 1.11 and 1.12 show fatigue caused by misshapen shafts, while Fig. 1.13 illustrates what happens when the bearing cap shifts. Regrinding the crankshaft will correct the former. Cap shift can be avoided by: (1) alternate torquing from side to side to ensure proper cap seating, (2) using new bolts to ensure against excessive play in bolt holes, (3) making sure the cap isn't reversed when installed, and (4) using the correct size socket to tighten the bolts to avoid interference with the cap.

Excessive Wear. Some of the same factors that produce fatigue also can cause excessive wear. Generally, what determines the phenomenon that prevails is the load level and the severity of the irregularity which causes the problem. Geometric defects not only concentrate loads but also cause oil films to be thinner than normal. This results in more frequent metal-to-metal contact and wears the lining much faster than normal. Figure 1.14 shows excessive wear caused by a barrel-shaped

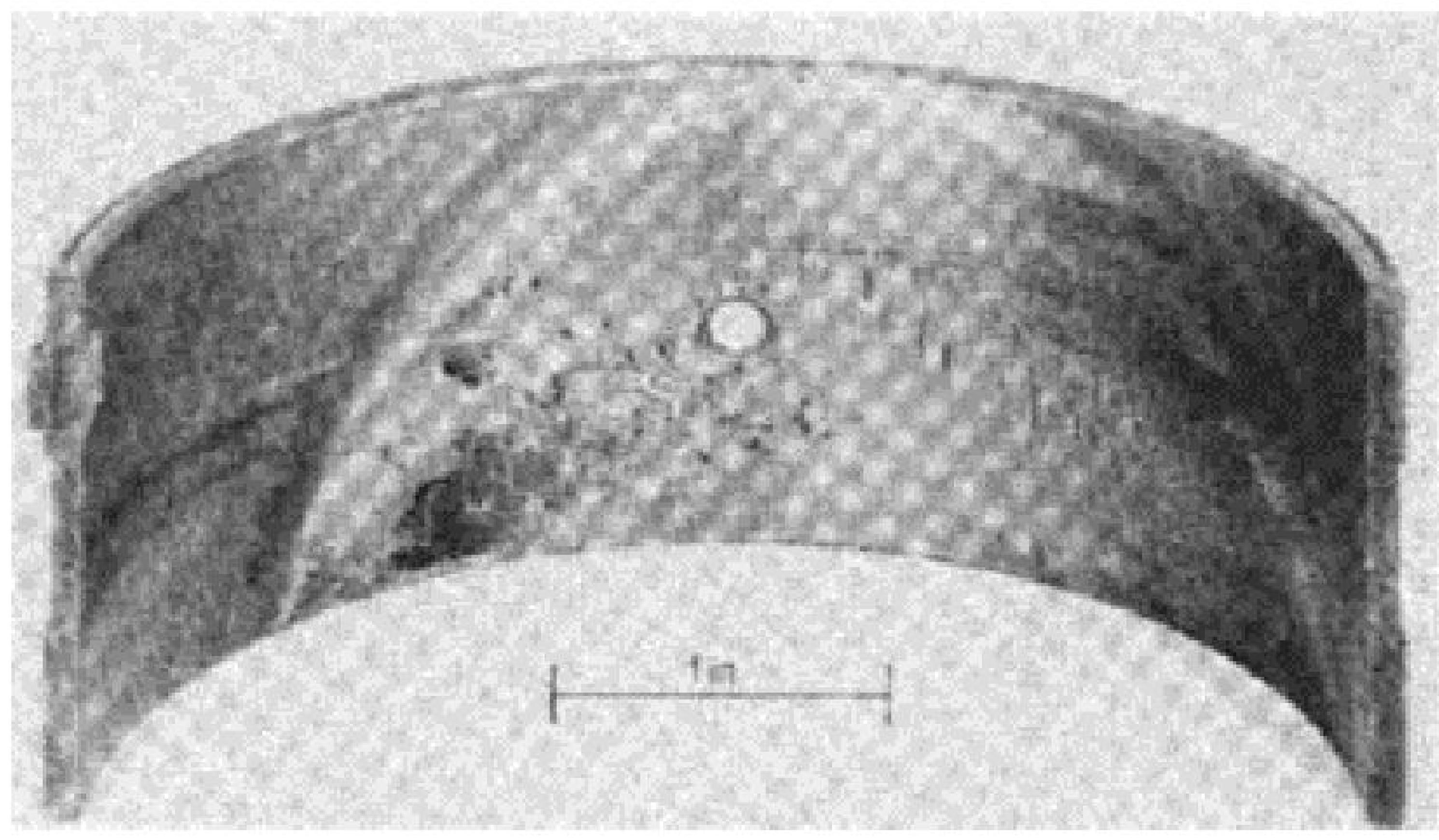


FIGURE 1.11 Fatigue caused by a tapered shaft.

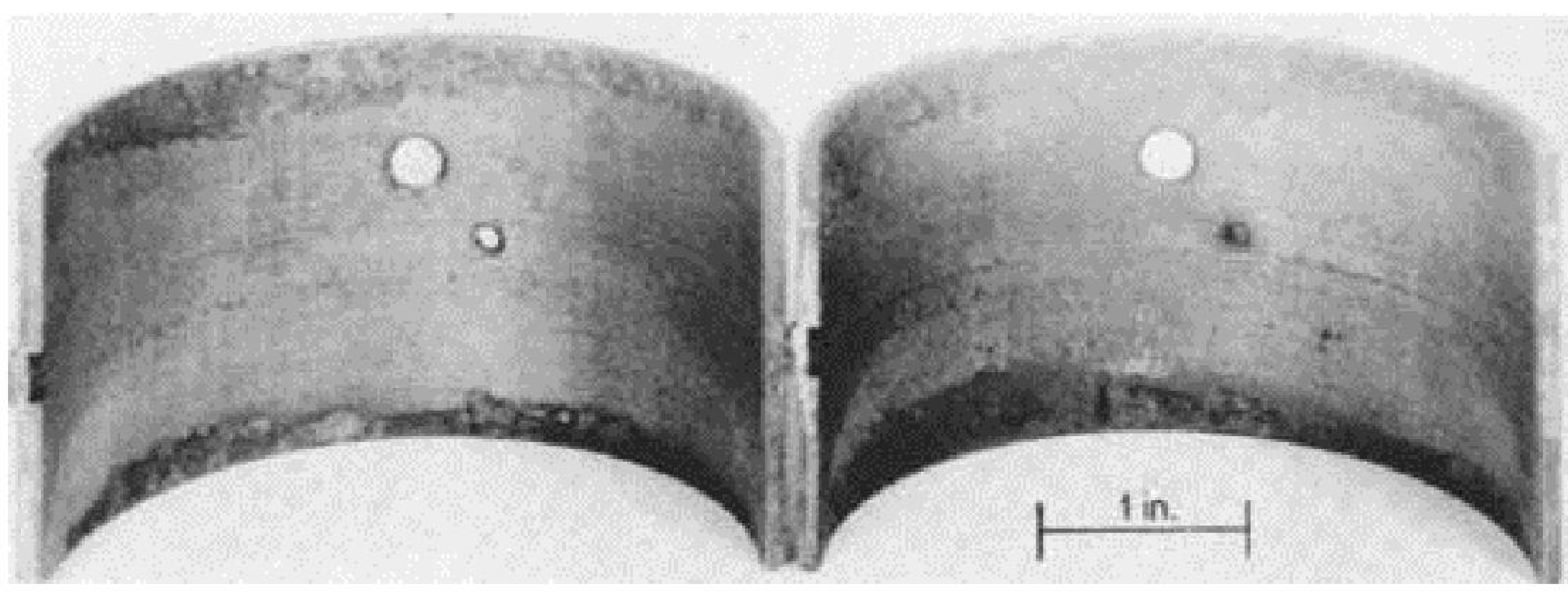


FIGURE 1.12 Fatigue caused by an hourglass-shaped shaft.

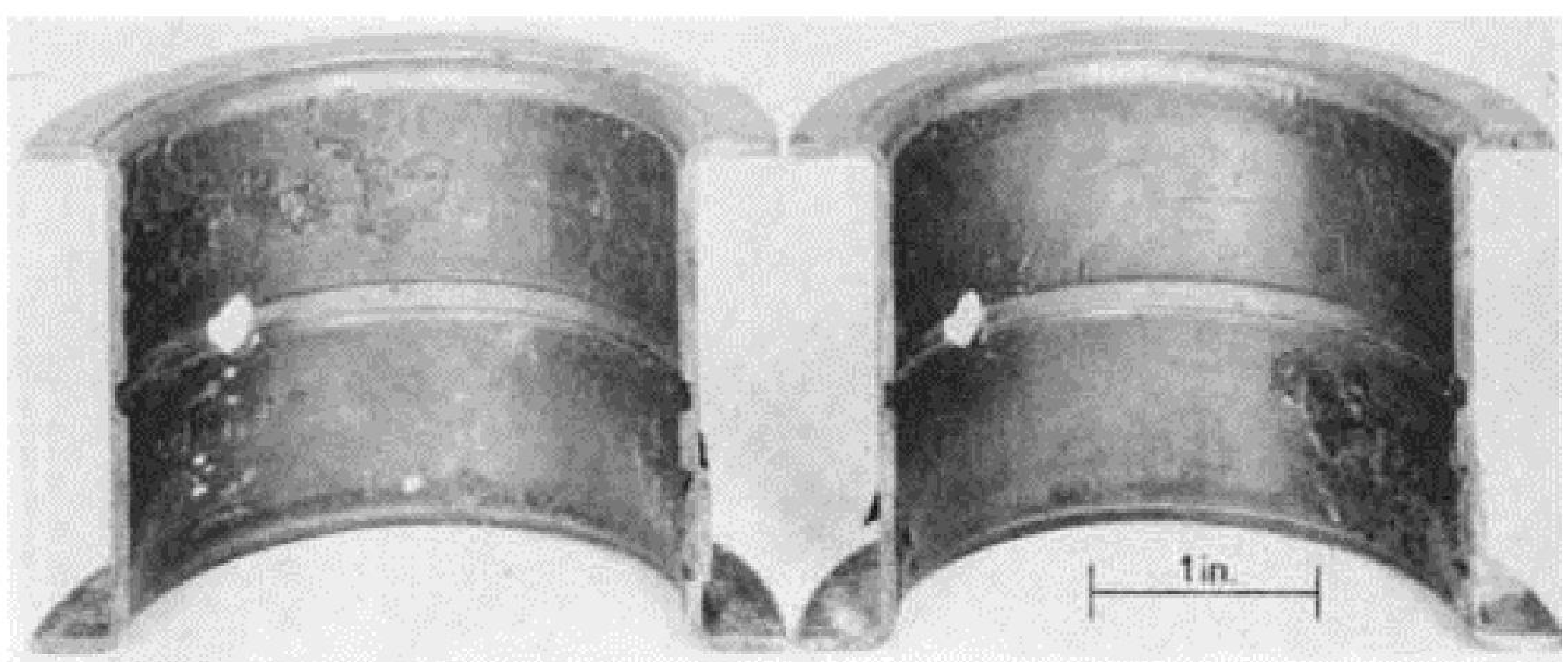


FIGURE 1.13 Fatigue caused by a shifted bearing cap.

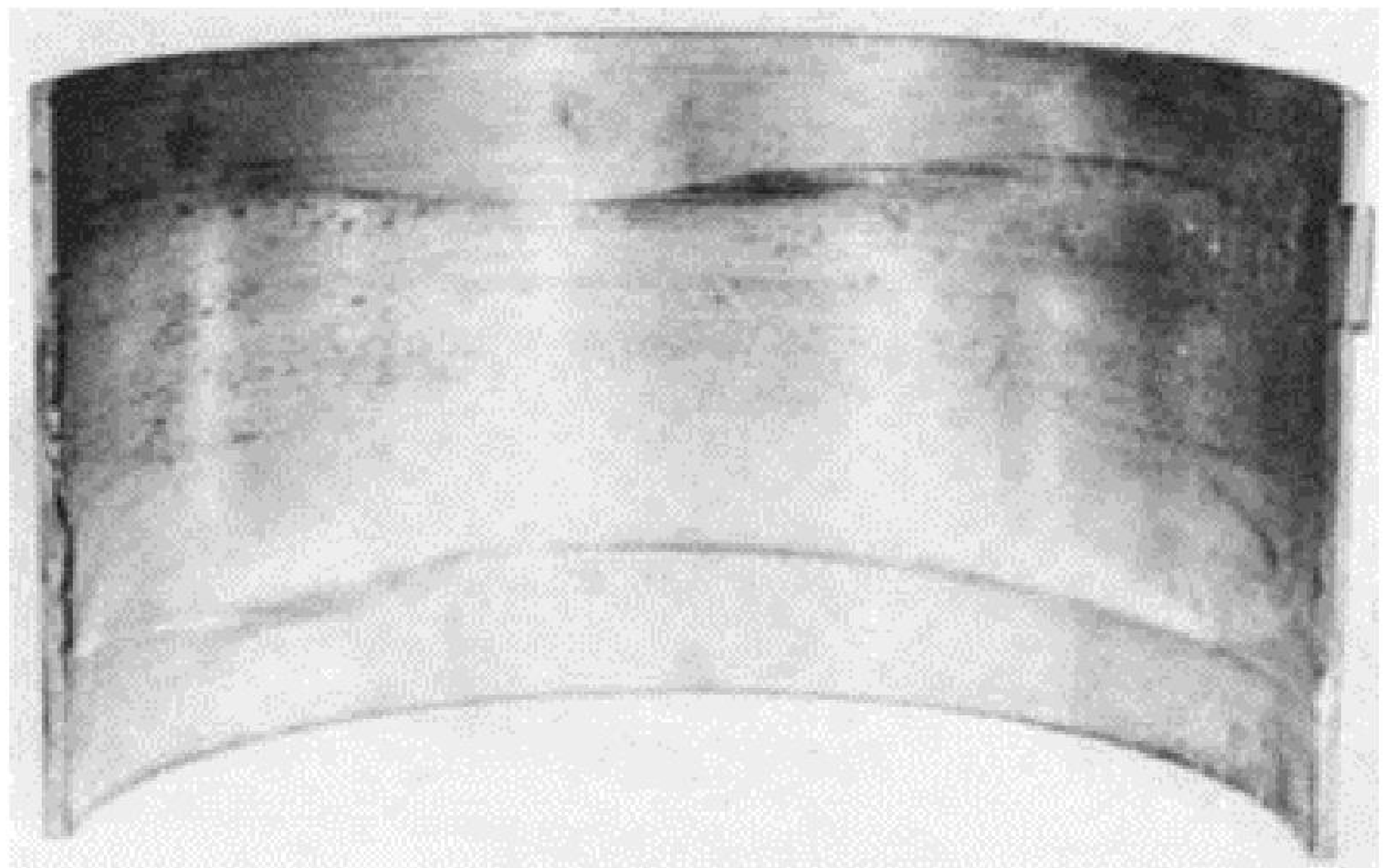


FIGURE 1.14 Bearing worn in the center by a barrel-shaped shaft.

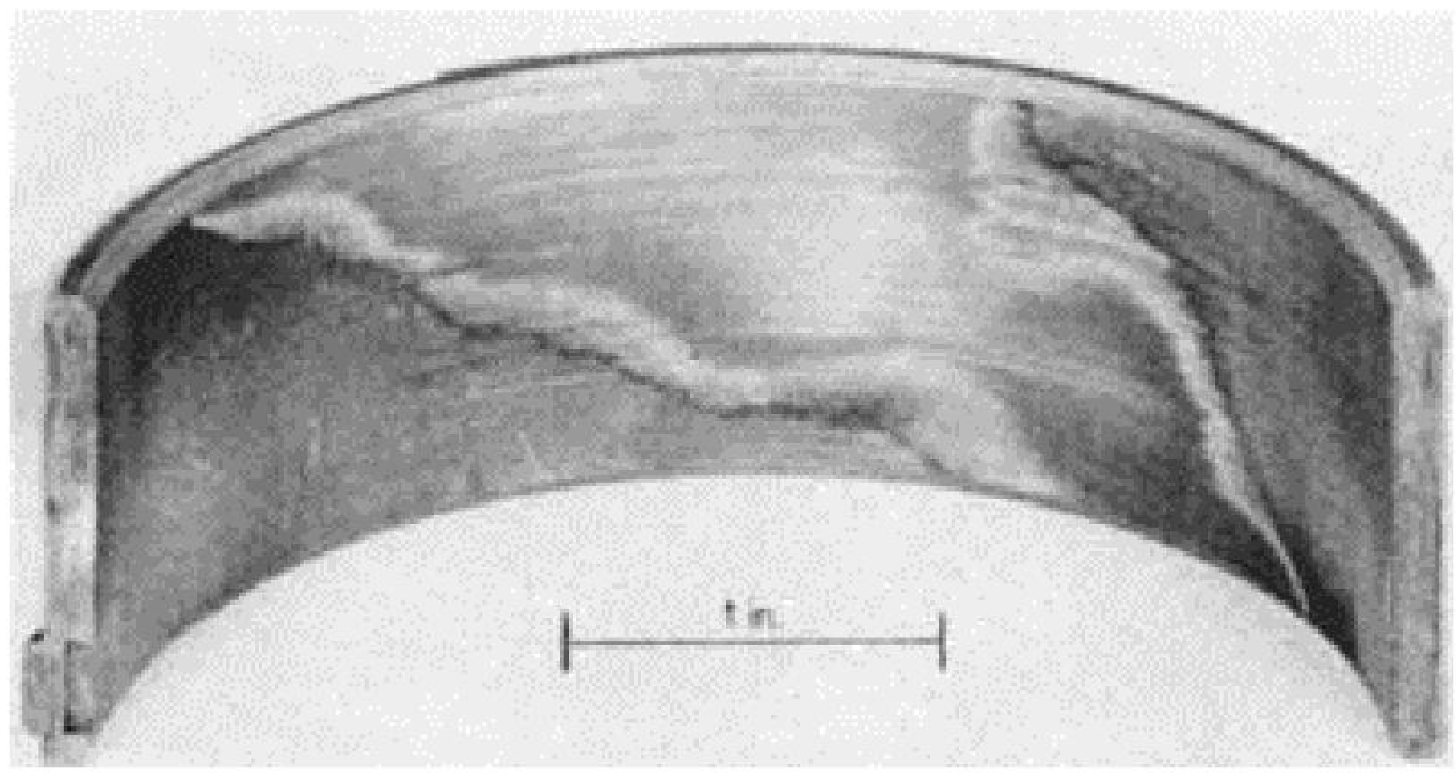


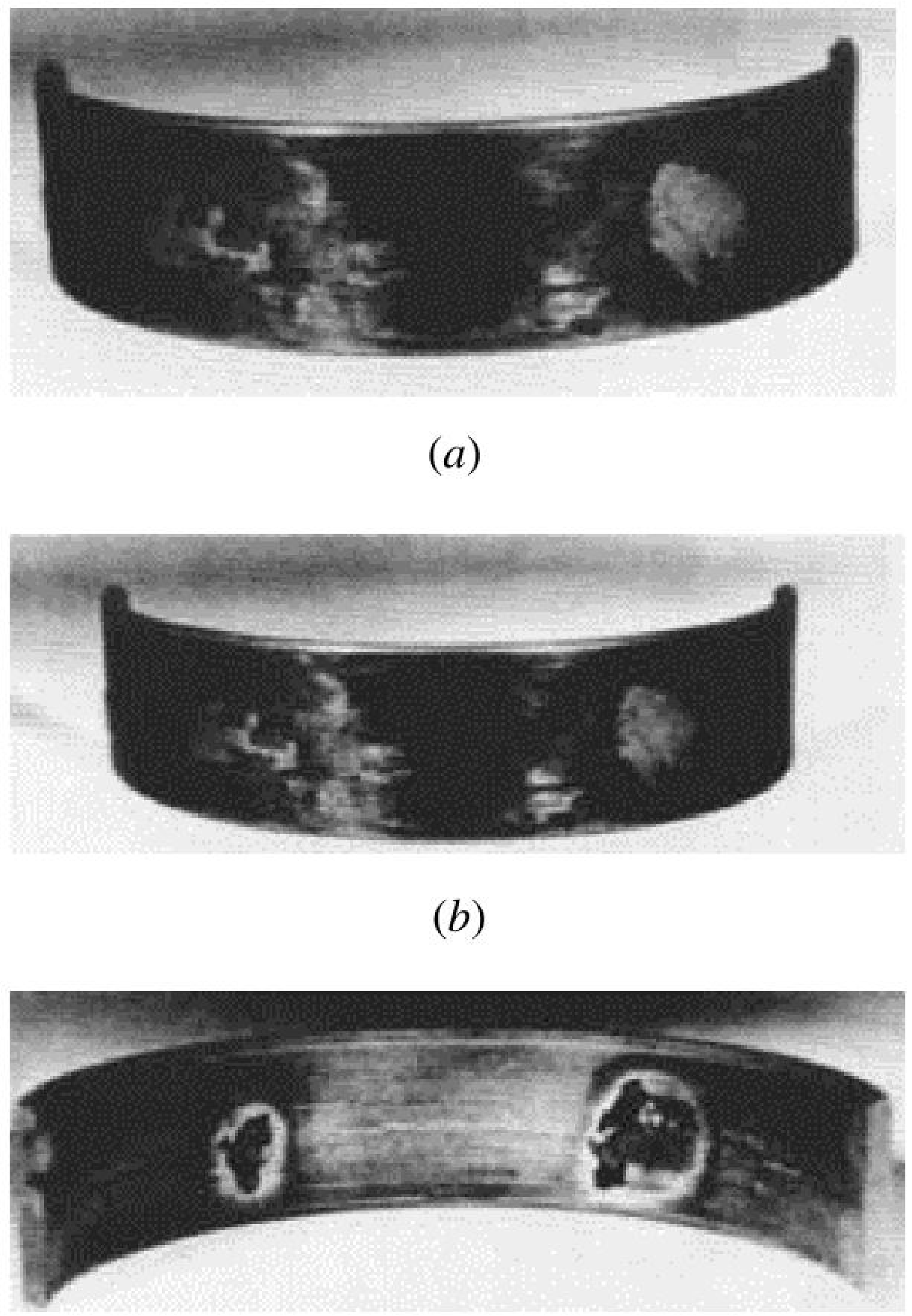
FIGURE 1.15 Skewed wear pattern caused by a bent connecting rod.

journal, while Fig. 1.15 shows the skewed-wear pattern from a twisted connecting rod. Regrinding the shaft will correct the former, while replacing the rod is the best course of action in the latter case. Related upper cylinder parts also should be checked and replaced as needed when a bent or twisted rod has been discovered.

Foreign Material on Bearing Back. Dirt on the bearing back causes high spots on the I.D. It also prevents good heat transfer in these areas, which leads to localized overheating. The end result may be either severe local wear or, as in Fig. 1.16, fatigue. Clearly, this type of problem can be prevented through proper cleaning and burr removal prior to assembly.

Hot-Short Phenomenon. A bearing suffering this type of failure (see Fig. 1.17) is unmistakable in appearance: Large areas of the lining have been cleanly removed from the steel back. The damage occurs when the bearing temperature exceeds the melting point of its lowest-melting-point metal, usually lead or tin. This heat, in conjunction with shear due to shaft-to-bearing contact, leads to the "hot-short" (brittle when hot) condition. Causes of the failure can be insufficient oil flow, excessive dirt in the oil, a rough shaft, or severe misalignment. Historically, dirt has been the most frequent cause of hot-short failures. Correction involves thorough cleaning, regrinding the shaft to fix the damaged journal, checking for blockage of oil passages, the oil-suction screen, and oil filter, and making sure the oil pump and pressure relief valves are operating properly.

Crush Problems. If crush is insufficient, relative movement occurs between a bearing and its bore. So-called fretted areas (see Fig. 1.18) will be visible on the bearing back and sometimes on the parting lines. These will appear to be highly polished and/or pitted. Corresponding damage also may be present in the housing bore. Fretted areas are points of stress concentration. If allowed to operate



(*a*)

(*b*)

(*c*)

FIGURE 1.16 Foreign particles on bearing O.D. (*top*) can result in I.D. fatigue (*bottom*).

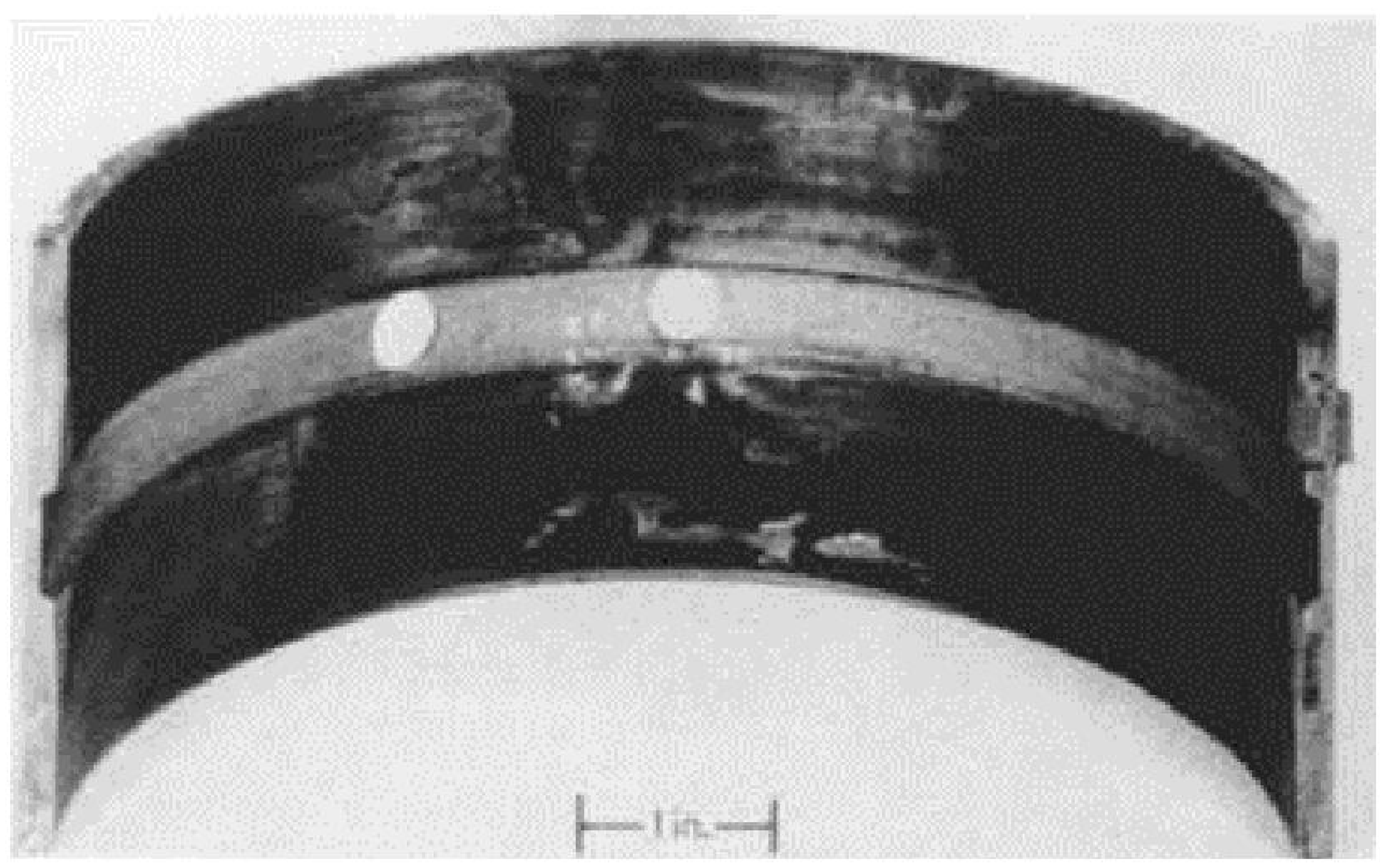


FIGURE 1.17 Hot-short condition (see text for details).

long enough, a fretted bearing may fracture through its steel back, as may the housing. Causes of insufficient crush are (1) filing of the parting lines, (2) dirt or burrs on the contact surfaces of the cap and housing, (3) insufficient bolt torque, and (4) oversize housing bore. Correction of the problem simply involves following good installation practices: Bearings should never be altered, mating faces of the assembly should be clean and burr-free, the bore size should be verified as correct, and proper torque should be applied.

Excessive crush can arise if bearing *caps* are filed down in an attempt to reduce oil clearance or if the cap bolts are overtorqued. When too much crush is present, the bearings bulge inward at the parting lines. This can lead to fatigue or metal-to-metal contact and potentially to a hot-short failure if the oil flow becomes pinched off. Correction involves ensuring that proper torque is applied and, if the cap has been filed down, replacing the connecting rod or reworking the bearing bore if a main bearing is involved.

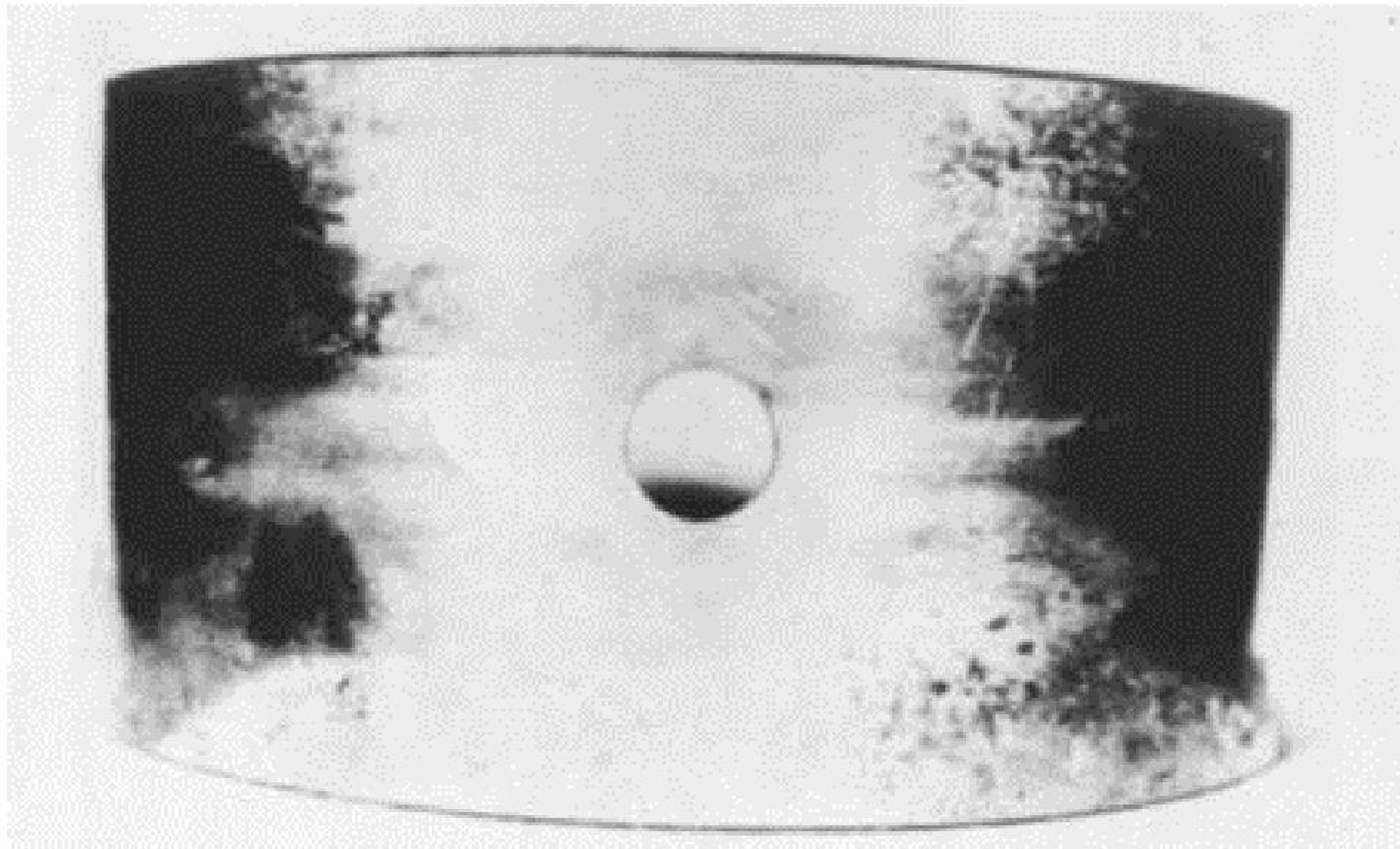


FIGURE 1.18 O.D. fretting.

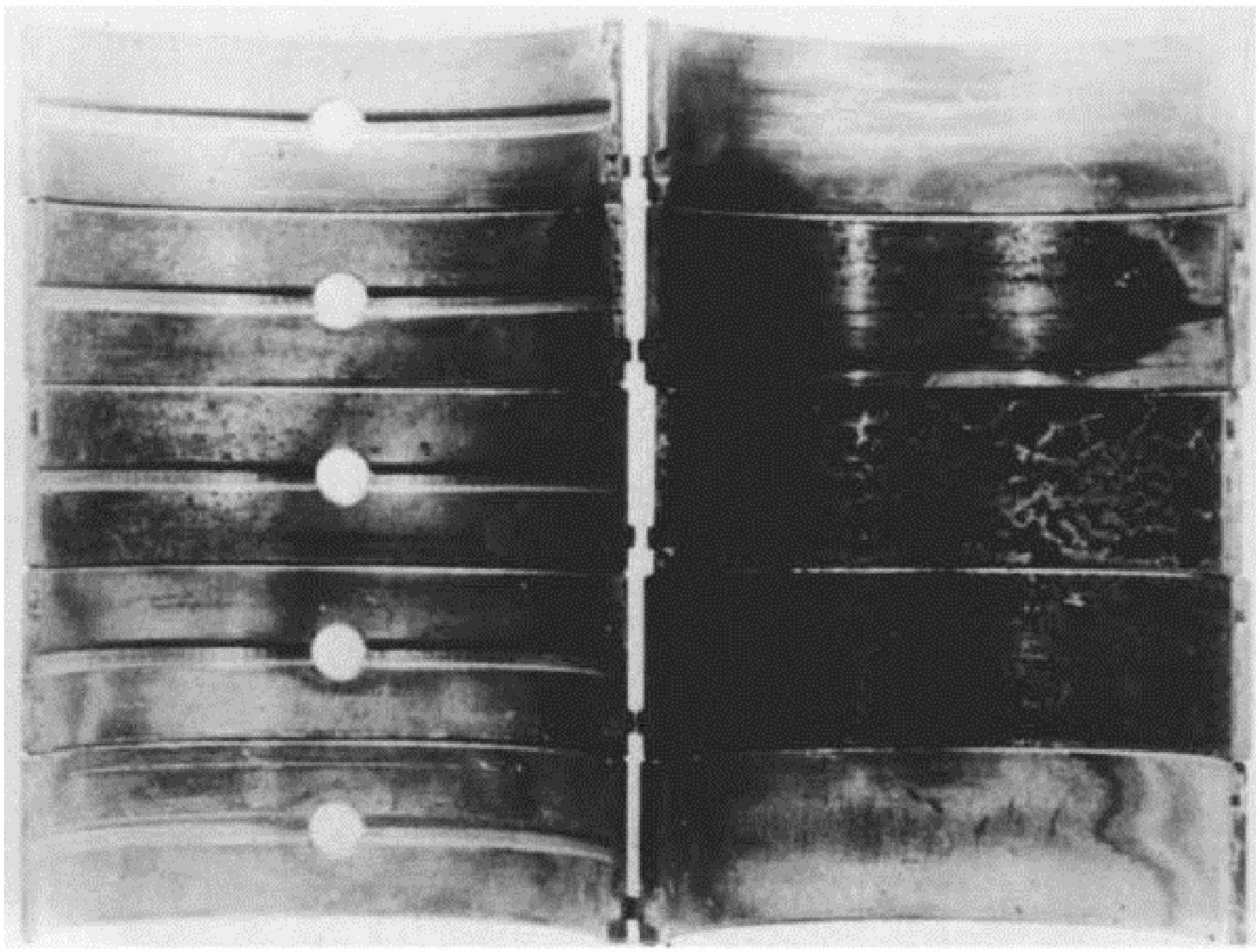


FIGURE 1.19 Fatigue of main bearing set due to a distorted crankcase.

Crankcase and Crankshaft Distortion. These structural defects primarily affect the main bearings. Distortion causes increased loads and lower oil films, with conditions being worst at the point of maximum distortion. The damage varies from bearing to bearing, with the center main usually showing the greatest amount.

If the crankcase is distorted, a wear or fatigue pattern will be present on either the upper or lower bearings (see Fig. 1.19). Alternating periods of engine heating and cooling are the primary cause, though extreme operating conditions and improper torquing of cylinder head bolts also can be responsible. To correct the problem, the crankcase must be line bored.

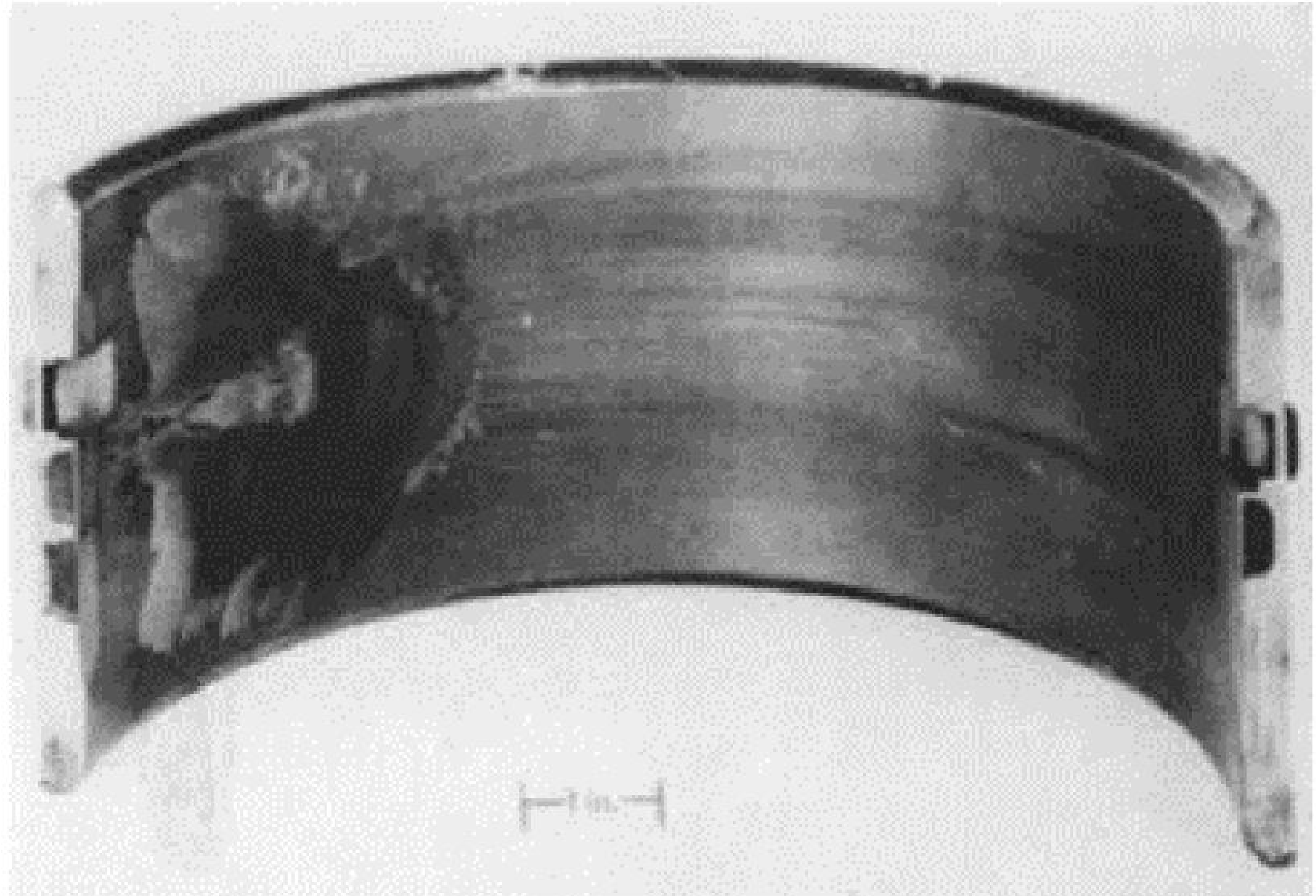


FIGURE 1.20 Cavitation.

A bent crankshaft causes a similar failure, but one which involves both the upper and lower bearings. The primary cause is extreme operating conditions. The corrective action is to install a new or reconditioned crankshaft.

Cavitation. This failure is induced by rapid fluctuations in oil film pressure. When the pressure in one area of the film drops below the oil's vapor pressure, a vapor-filled cavity forms. When the pressure increases again, the cavity collapses. This causes the surrounding oil to impinge on the adjacent bearing metal, eventually eroding the surface (see Fig. 1.20). Cavitation appears to be inherent to some applications. Ensuring that there is no air or water entrainment in the oil may help the problem. If possible, changing to higher-viscosity oil and increasing the oil pressure also may help.

Reconditioning

The preceding section discussed the causes of common bearing failures. It should be obvious that the vast majority of them can be prevented through proper maintenance. When bearings need to be replaced, however, it must again be emphasized that the job is not simply one of removing the old parts and installing new ones. It is mandatory that associated components be inspected and, if required, reconditioned. Every application is unique, and service manuals for the specific equipment should be consulted for detailed instructions. The discussion which follows concerns engines; however, the basic principles apply regardless of equipment type.

Shaft. After cleaning the crankshaft, measure all the main and crankpin journals at several points to determine the out-of-round and taper. If any journal measures 0.001 in. less than the manufacturer's specified diameter, has more than 0.001 in. taper, or is more than 0.001 in. out of round, the crankshaft should be reground before rebuilding the engine. Specifications on tolerances and permissible values for geometric irregularities are given in Table 1.2. Note that the latter are more stringent than the criteria for whether or not to regrind the shaft. This is because most manufacturers permit looser tolerances on rebuilds than on new or remanufactured parts. However, it must be recognized that a penalty is paid for doing this. Namely, one cannot expect to realize the same bearing life from a crankshaft that has not been reground as from one that has been.

In regrinding the shaft, a specific procedure is to be followed. If significant material must be removed from the journals, first turn the shaft (i.e., in a lathe) in the direction of the crankshaft rotation in the engine. Then, with the shaft rotating in the opposite direction, grind toward the high limit. Finally, polish the journals in the direction of the crankshaft rotation in the engine to remove the grinding fuzz. A maximum of 240 grit paper should be used, with a maximum stock removal of

TABLE 1.2 Shaft Tolerances

	Automotive	Heavy duty
Diameter tolerance:		
Up to 1½-in journal	0.0005 in.	0.0005 in.
1- up to 10-in. journal	0.001 in.	0.001 in.
Over 10-in journal	0.002 in.	0.002 in.
Diametral-taper tolerance:		
Up to 1 in. of length	0.0002 in.	0.0001 in.
1 up to 2 in. of length	0.0004 in.	0.0002 in.
Over 2 in. of length	0.0005 in.	0.0003 in.
Out-of-round condition:		
Up to 3 in. diameter	0.0005 in.	0.0002 in.
3 to 5 in. diameter	0.0005 in.	0.0003 in.
Over 5 in. diameter	0.001 in.	0.0004 in.
Maximum misalignment:		
Adjacent main journals	0.001 in.	0.0005 in.
Crankpin parallel with main journals	0.001 in.	0.0005 in.
End clearances:		
Shaft diameter		
2–2¾ in.	0.003–0.007 in.	0.003–0.007 in.
2¾–3½ in.	0.005–0.009 in.	0.005–0.009 in.
3½–5 in.	0.007–0.011 in.	0.007–0.011 in.
Over 5 in.	0.009–0.013 in.	0.009–0.013 in.
Shaft hardness:		
Brinell	200 min	300 min
Shaft-journal finish (all applications):		
Microinches 15 max		
Waviness 0.0001 in. max		
Lobing 0.0001 in. max		
Chatter 0.00005 in. max		

0.0001 in. Oil holes should be blended well into the journal surfaces, with the maximum diameter at the runout of the blend not exceeding twice the hole diameter. The reconditioned shaft and its oil passages should be thoroughly washed in solvent, followed by hot soapy water and a hot water rinse. Use a round, nonmetallic bristle brush to clean the oil passages. Following the final rinse, blow the shaft and oil passages dry with compressed air, and immediately coat all journal surfaces and oil passages with a light film of the oil normally used in the engine.

Connecting Rods. Measure the empty rod bores on several diameters. If any bore is more than 0.001 in. out of round, replace the rod or recondition it. Rods should then be checked for parallelism and twist (Fig. 1.21). A slight bend or twist can be straightened using caution and the proper equipment. Severely bent or twisted rods should be replaced. Table 1.3 shows tolerances and permissible values for geometric deviations for rod bores.

Crankcase. After a thorough cleaning, including all oil holes and passages, measure each empty bore in the same manner as was done for the connecting rods. The standards of Table 1.3 apply to mains as well as rods.

Crankcase alignment must then be checked. The best way of doing this is as follows: Use an arbor ground to 0.001 in. less than the low limit of the empty bore specification. The arbor should be slightly longer than the crankcase. Place the arbor into the main bearing saddles (without bearings),

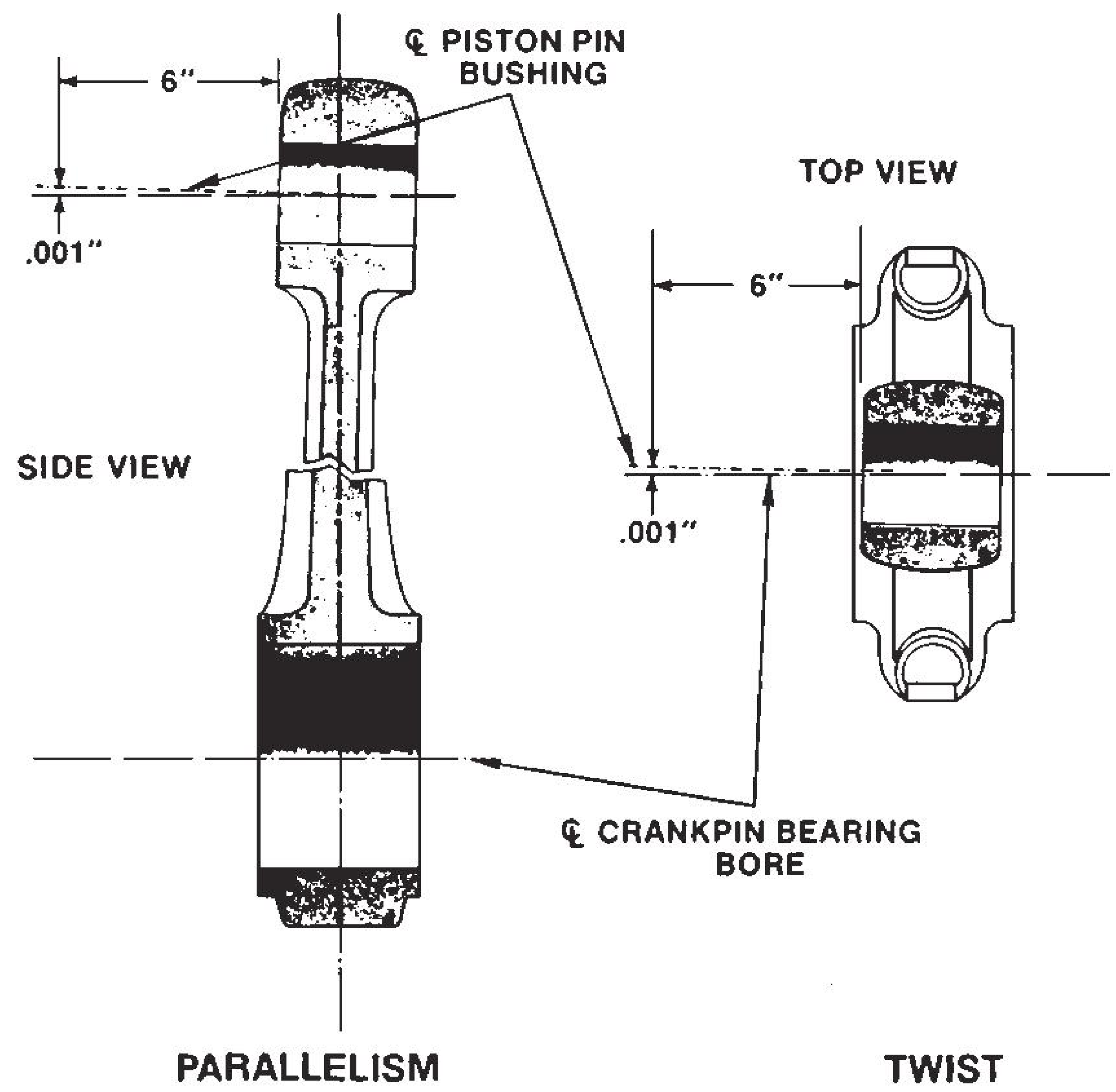


FIGURE 1.21 Connecting-rod parallelism and twist should be checked prior to reassembly with new bearings.

TABLE 1.3 Empty Bore Tolerances

	Automotive	Heavy duty
Diameter:		
Up to 3¼ in. diameter	0.0005 in.	0.0005 in.
3¼–10 in. diameter	0.001 in.	0.001 in.
Over 10 in. diameter	0.002 in.	0.002 in.
Taper, hourglass, or barrel shape:		
1-in. length	0.0002 in.	0.0001 in.
1- to 2-in. length	0.0004 in.	0.0002 in.
Over 2-in. length	0.0005 in.	0.0003 in.
Out-of-round:		
0.001-in. area if bore is larger horizontally than vertically.		
Parallelism and twist between rod bore and wrist-pin bore when measured 6 in. from the end of wrist-pin bushing 0.001 in. max.		

position the main caps in place, and torque the bolts to blueprint specification. Rotate the arbor. If it will not turn and earlier measurements did not indicate an out-of-round condition, the crankcase is probably warped and must be line bored.

Reassembly. Prior to reinstallation of the crankshaft, the main bearings should be given a thin coating of oil and assembled clearances verified with a material such as Plastigage. As soon as all the main caps are in place and torqued, the shaft should be rotated. If binding occurs, proceed no further until the cause is found and corrected. The connecting rod bearings also should be prelubricated prior to attaching the rod assemblies to the crankshaft. After doing so, verify clearances and again check crankshaft rotation. Find and correct any binding that occurs. After reassembling the other parts to the engine, it is advisable to use an engine prelubricator to fill all parts of the lubrication system. This prevents the possibility of a dry start.

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CHAPTER 2
ROLLING-ELEMENT BEARINGS

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GENERAL

Reliable bearing performance is a key factor in reducing maintenance costs and in improving machine availability. When bearings fail, they can bring equipment to an unscheduled halt. Every hour of downtime due to premature bearing failure can result in costly lost product, especially in capital-intensive equipment. To keep machinery in peak operating condition, the bearings should be properly aligned and protected from extreme temperatures, moisture, and contaminants. Knowledge of the proper installation techniques and tools is required to ensure that bearings are not prematurely damaged. Following lubrication and maintenance schedules according to the original equipment builder's operating and maintenance recommendations and monitoring bearing operating conditions are also important in maximizing bearing service life. For special cases, most bearing manufacturers and equipment builders maintain service departments to render technical assistance. As with any precision mechanical component, bearings should be handled with care and common sense to prevent damage from mechanical abuse and contamination.

WHY BEARINGS FAIL

Only a fraction of all bearings in service actually fail. The vast majority outlive the machinery or equipment in which they are installed. A bearing failure may occur for many reasons, i.e., heavier loading than had been originally anticipated, ineffective seals or mountings which are too tight resulting in too little bearing internal clearance, etc. Each of these factors produces its own particular type of damage and leaves its own imprint on the bearing. Consequently, by examining a damaged bearing, it is possible in a majority of cases to determine the cause and take corrective action to prevent a recurrence.

Of the small fraction of bearings that actually do fail, approximately one-third die from old age (e.g., fatigue of the bearing surfaces), another third fail because of poor lubrication, and the rest fail as a result of liquid or solid contaminants entering the bearing or because of handling damage or faulty mounting. The pattern of failure can vary according to the specific industry where the bearing is used. For example, in the pulp and paper industry, poor lubrication and contamination are the major causes of failure, not fatigue.

Bearings are selected based on very specific operating conditions. Too often changes involving a different lubricant, higher machine speeds, higher loads, changes in lubrication systems, etc., are made without anticipating the possible negative effects on bearing performance. Therefore, when replacing a bearing, no other changes should be made which would adversely affect bearing operation.

BEARING DESIGNS AND NOMENCLATURE

Rolling bearings include radial and thrust bearings for radial and axial loads, respectively, and some bearing types which are designed for combined radial and axial loads. Generally speaking, ball bearings are recommended for light to moderate loads; roller bearings are recommended for heavy loads. Nine basic types of rolling bearings are shown in Fig. 2.1*a* to *d.* Some of these basic types are available in many variations; for instance, cylindrical roller bearings may be obtained with one, two, or four rows of rollers, as shown in Fig. 2.1*b.* Single-row deep-groove ball bearings are generally available in nine different external configurations, as shown in Fig. 2.2. Taper roller bearings can come in more than 20 different configurations, some of which are shown in Fig. 2.3. The other basic types do not come in large numbers of configurations, but it should be noted that all types of rolling bearings are available in many design variants and thus may vary greatly in internal design, depending on the manufacturer. It is not within the scope of this handbook to describe all the various designs of rolling bearings used in machinery but rather to alert maintenance personnel to their existence. Details are given in manufacturers' catalogs or by contacting the manufacturers directly.

BOUNDARY DIMENSIONS

In general, most ball, spherical roller, and cylindrical roller bearings made to metric boundary dimensions have standardized boundary plans, dimensions, and tolerances according to the International Standards Organization (ISO). Therefore, bearings from all subscribing manufacturers throughout the world are dimensionally interchangeable. Most taper roller bearings are made to inch dimensions and have standardized boundary dimensions and tolerances according to the

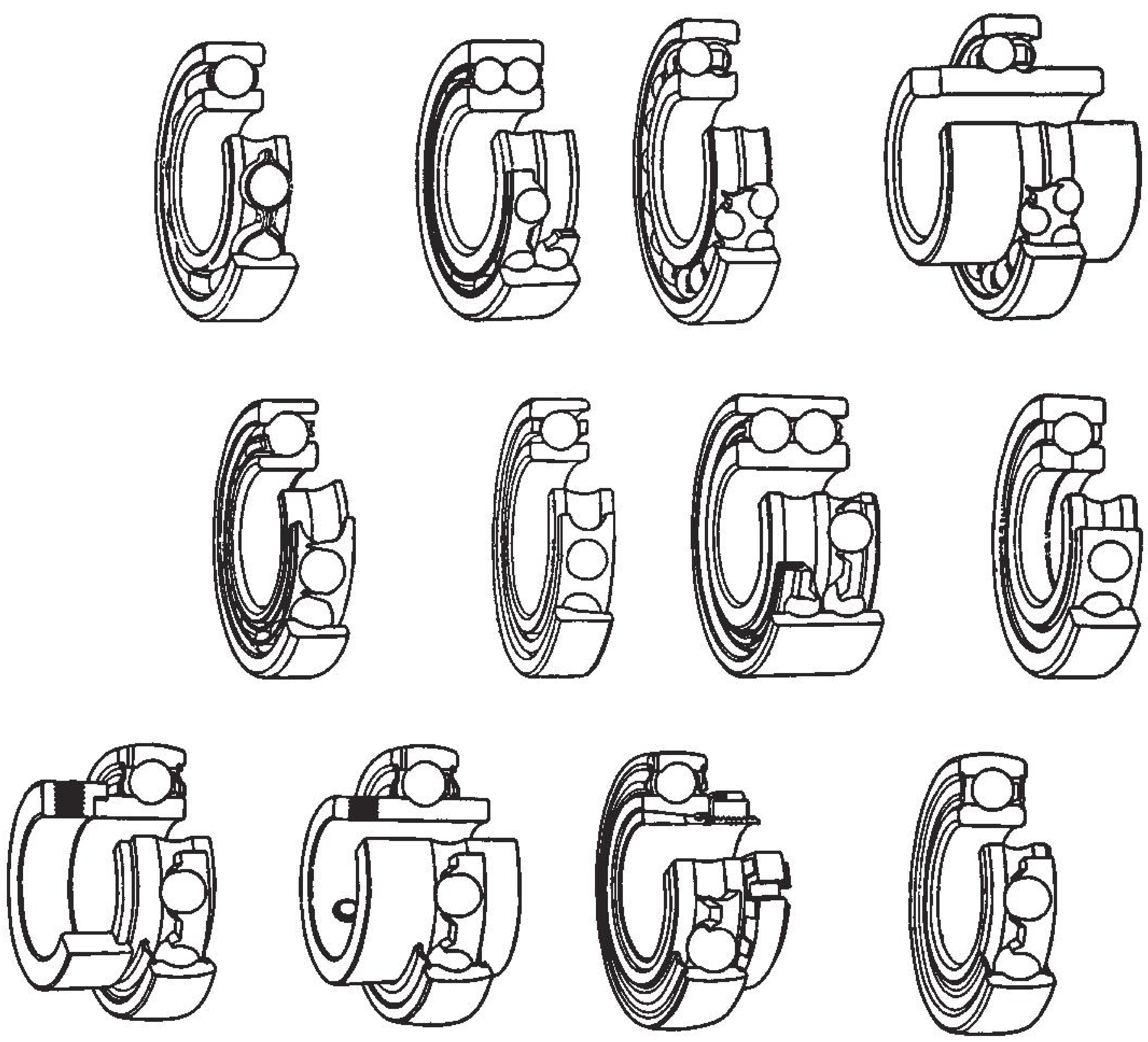


FIGURE 2.1 (*a*) Radial ball bearing types.

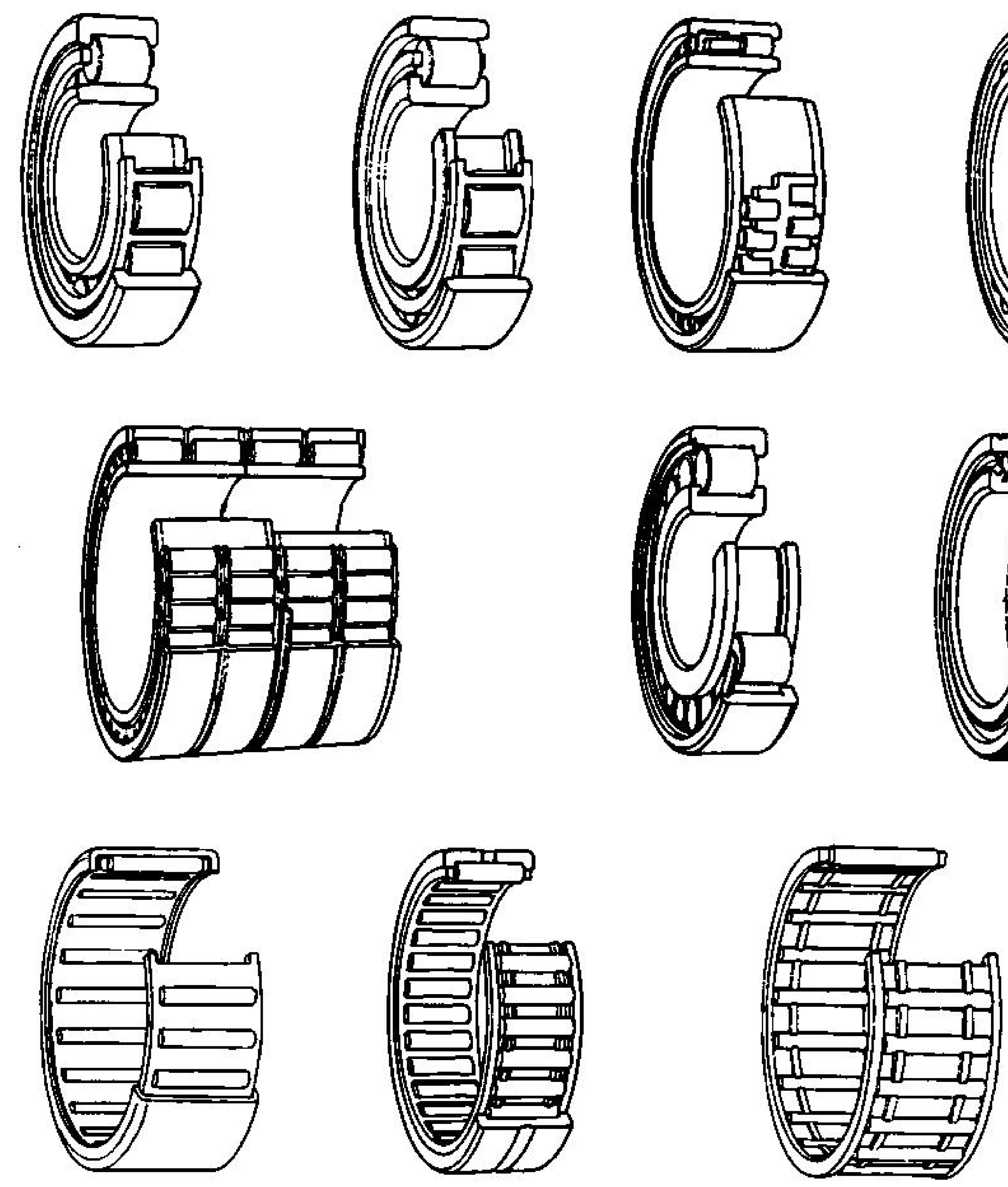


FIGURE 2.1 (*Continued*) (*b*) Radial roller bearing types.

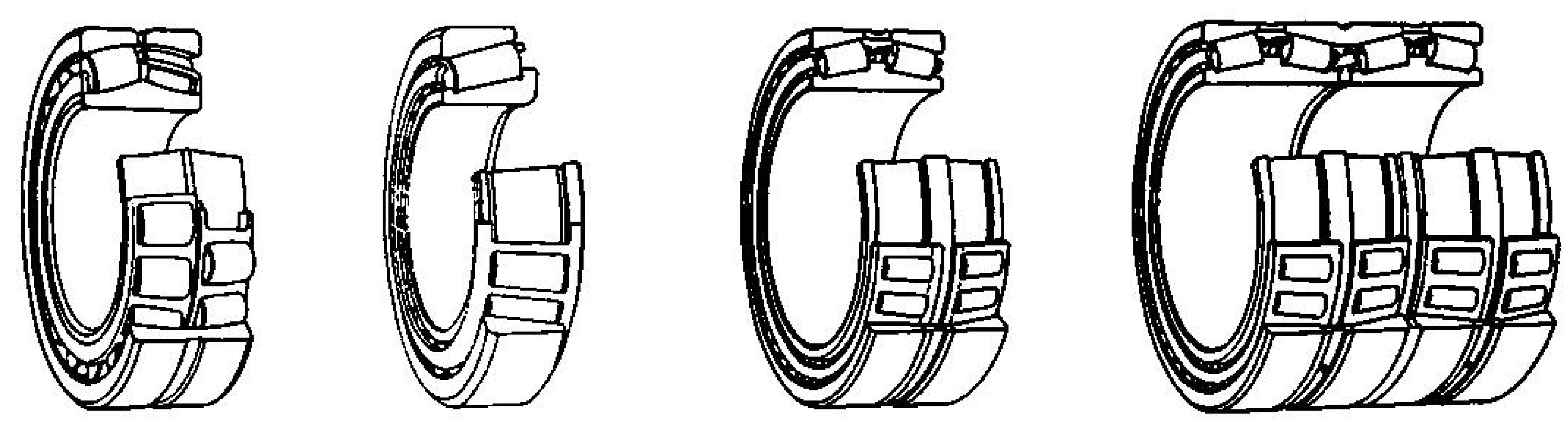


FIGURE 2.1 (*Continued*) (*c*) Roller bearing types for radial and axial loads combined.

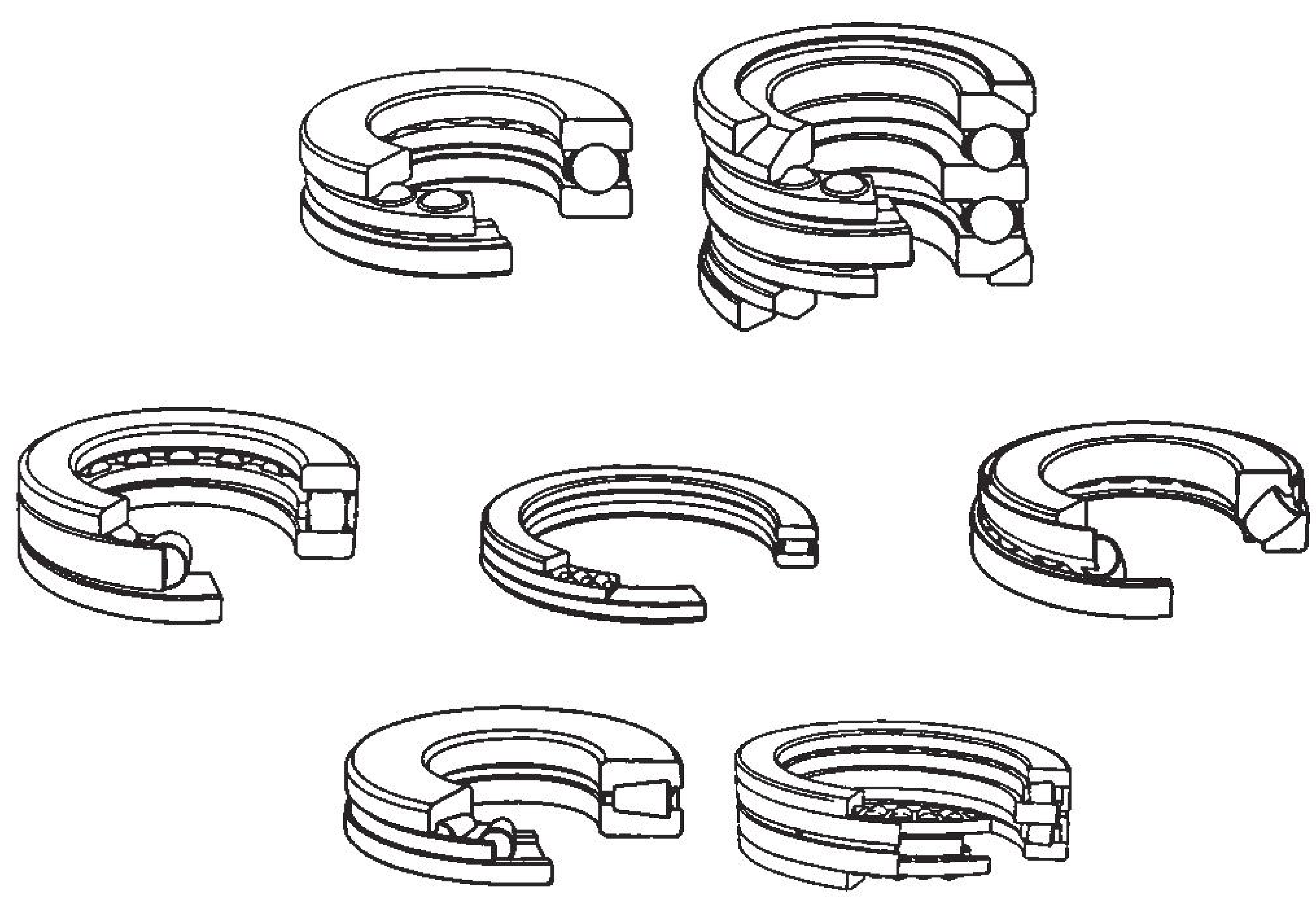


FIGURE 2.1 (*Continued*) (*d*) Thrust bearings.

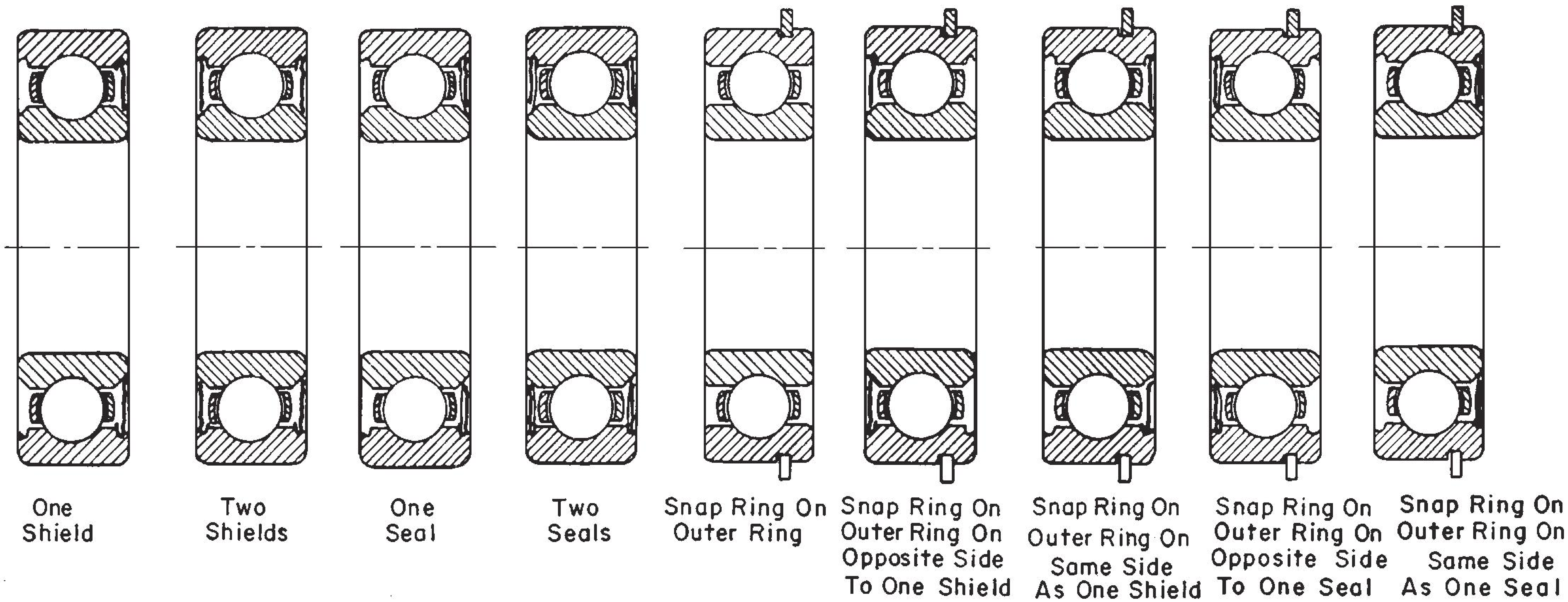


FIGURE 2.2 Single-row deep-groove ball-bearing shields, seals, and snap rings.

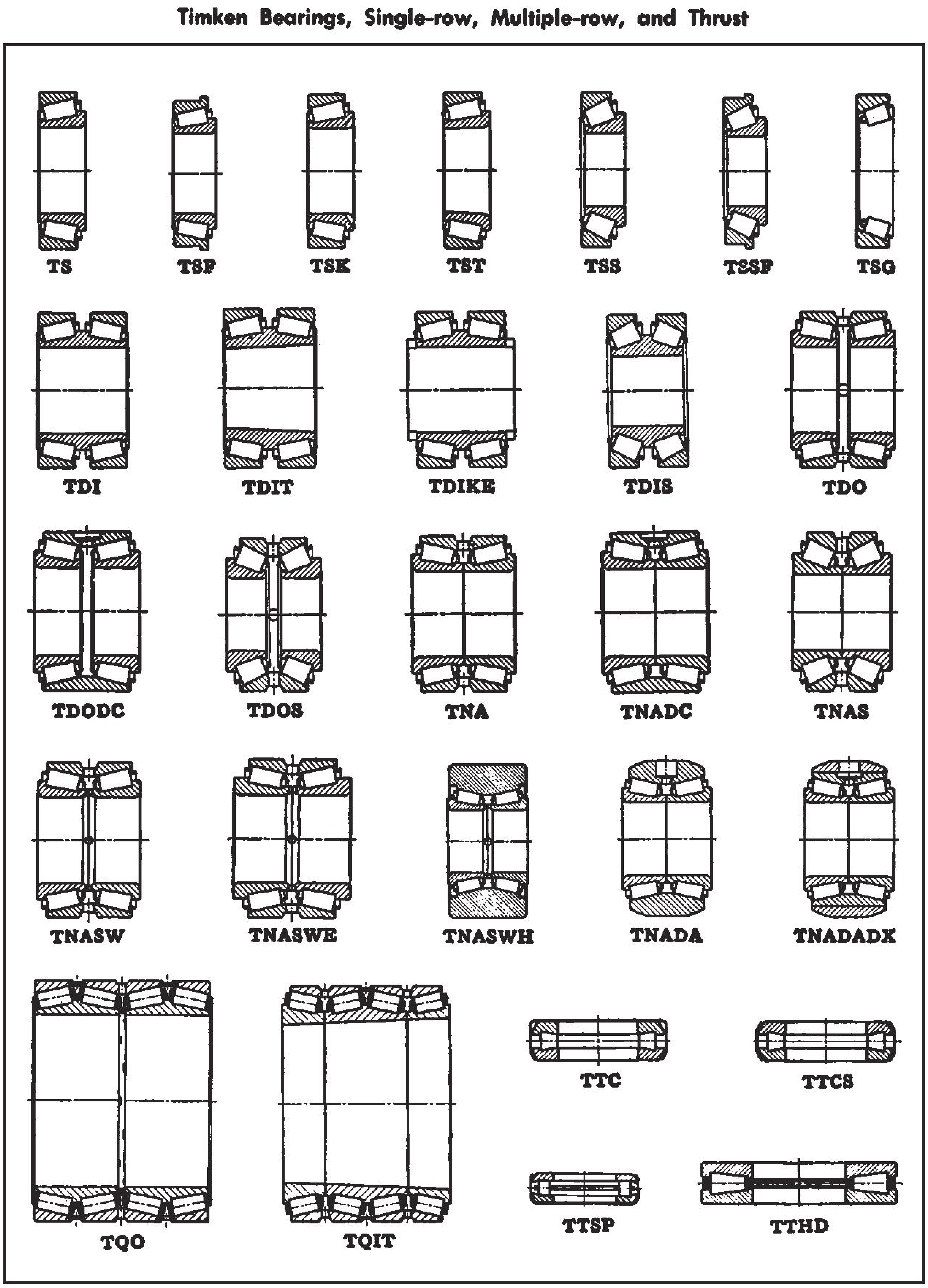


FIGURE 2.3 Tapered roller bearings, single-row, multiple-row, and thrust.

Anti-Friction Bearing Manufacturers Association (AFBMA), a U.S. standards organization. Metric taper roller bearings utilizing ISO boundary plans are also made. Dimensionally interchangeable taper roller bearing components are thus available from several manufacturers. In most cases, identical basic part numbers are used.

Inch-dimensioned cylindrical roller bearings are not manufactured according to any standard and will vary depending on the manufacturer.

BEARING SERIES

For any given bore size, all types of metric rolling bearings are manufactured in several series each for different severity of service. For instance, most ball bearings are made in three series: light, medium, and heavy duty. These are designated as the 2, 3, and 4 diameter series according to the boundary plan shown in Fig. 2.4. Spherical roller bearings are normally available in eight different series, as shown in Fig. 2.5. Taper roller bearings, both inch- and metric-dimensioned, have a larger number of series or duty classifications, but all series are not necessarily available for every bore size (see Fig. 2.6).

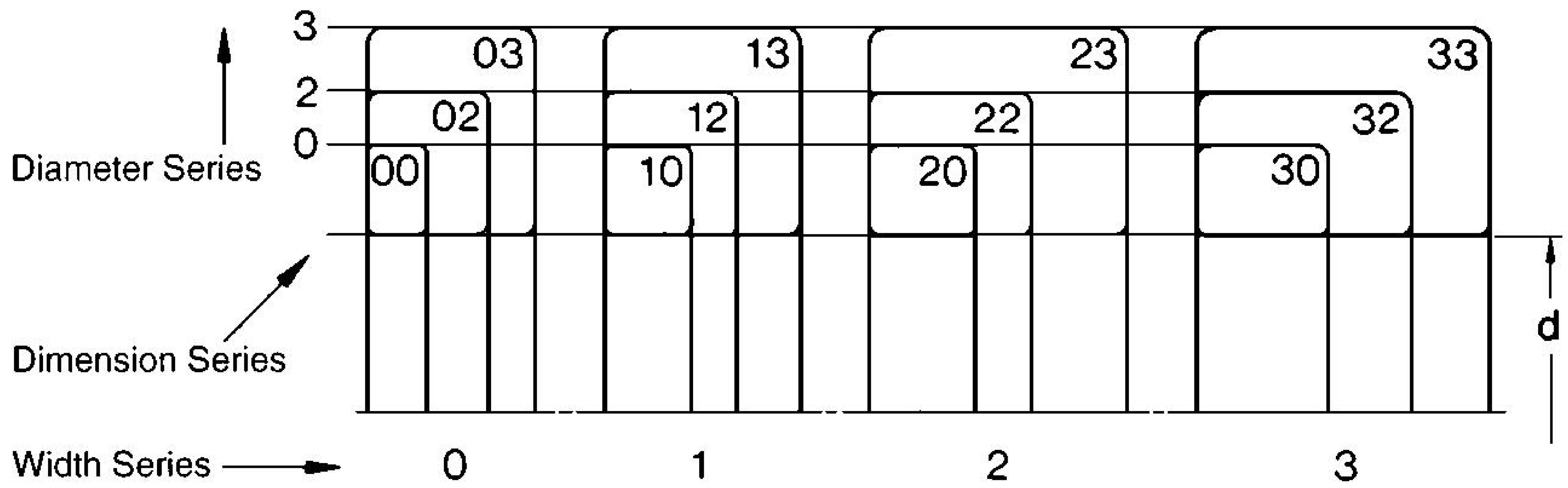


FIGURE 2.4 Metric rolling-bearing boundary dimension plan.

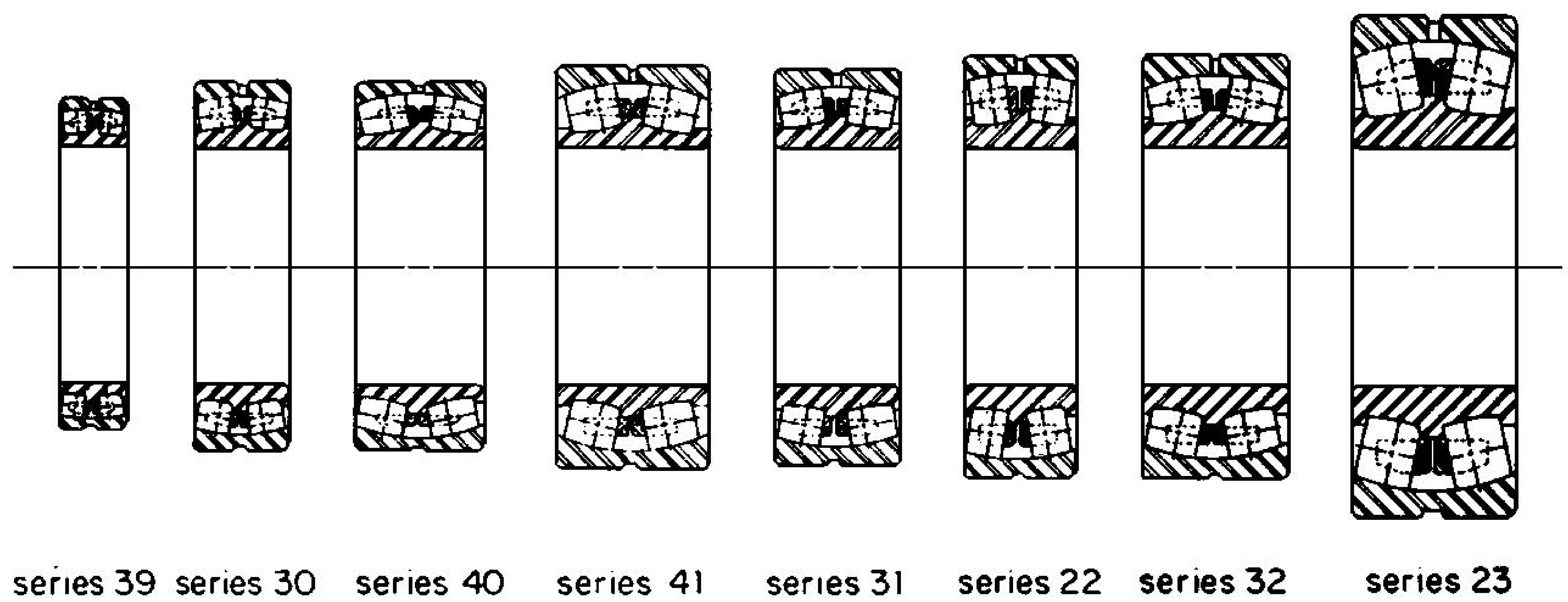


FIGURE 2.5 Spherical roller bearings of different diameter series with common bore size.

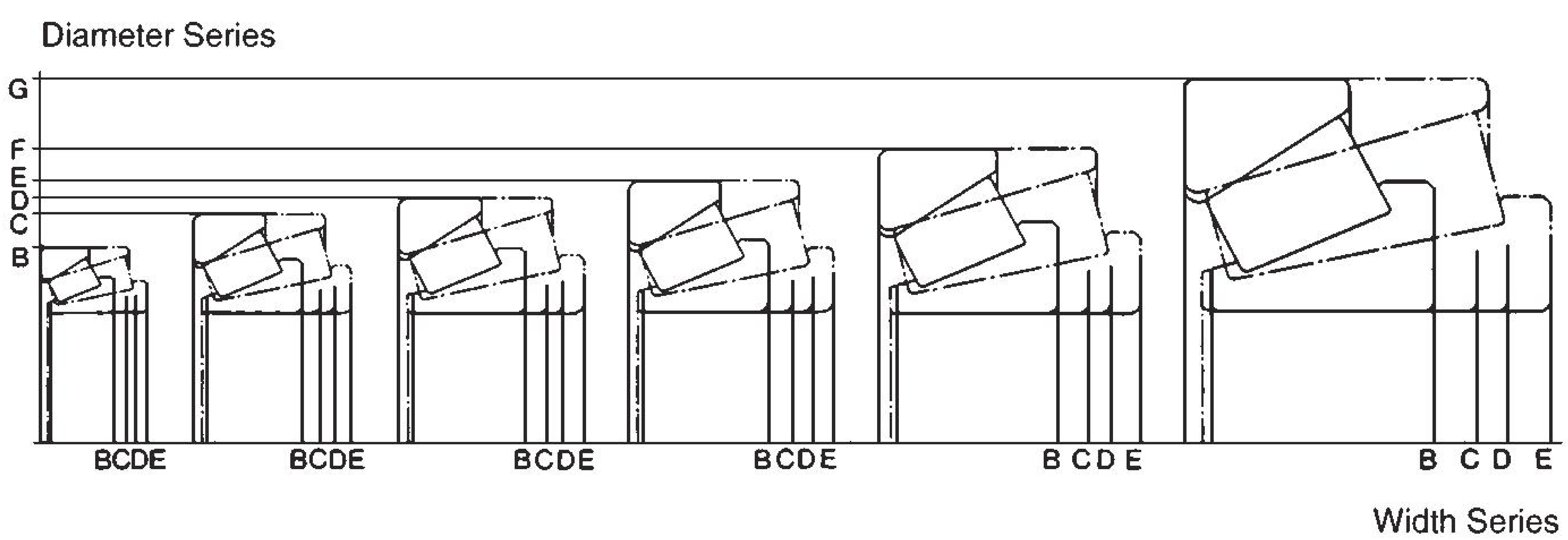


FIGURE 2.6 Metric tapered roller-bearing diameter series with common bore size.

LOAD RATINGS

All manufacturers of rolling bearings establish a dynamic and static load rating for each bearing produced. An ISO method for calculating this method exists, but not all manufacturers adhere to the method. The unfortunate situation therefore exists that two almost identical bearings produced by different manufacturers can have different published load ratings. Ratings are expressed as a load which will provide a basic rating life of a defined number of revolutions. *Basic rating life* is the number of revolutions (or the number of operating hours at a given constant speed) which the bearing is capable of enduring before the first sign of fatigue occurs in one of its rings or rolling elements. The basic rating life in millions of revolutions is the life 90 percent of a sufficiently large group of apparently identical bearings can be expected to obtain or exceed under identical operating conditions. In other words, this is a reliability or statistical rating, the only mechanical component so rated. The ISO definition of the basic rating life is the most common and is at 1 million revolutions. Some taper roller bearings are rated on the basis of 90 million revolutions, or 500 rev/min (rpm) for 3000 hours.

Hence it can be easily seen that comparing manufacturers' ratings as published in catalogs can be misleading if appropriate adjustments are not made to published values.

There are several other "bearing lives," including service life and design or specification life. *Service life* is the actual life achieved by a specific bearing before it becomes unserviceable. Failure is not generally due to fatigue, but to wear, corrosion, contamination, seal failure, etc.

The service life of a bearing depends to a large extent on operating conditions, but the procedures used to mount and maintain it are equally important. Despite all recommended precautions, a bearing can still experience premature failure. In this case it is vital that the bearing be examined carefully to determine a reason for failure so that preventive action can be taken. The service life can either be longer or shorter than the basic rating life.

Specification life is the required life specified by the equipment builder and is based on the hypothetical load and speed data supplied by the builder and to which the bearing was selected. Many times this required life is based on previous field or historical experiences.

SHAFT AND HOUSING FITS

It is a basic rule of design that one ring of a rolling-element bearing must be mounted on its mating shaft or in its housing with an interference fit, since it is virtually impossible to prevent rotation by clamping the ring axially. Generally, it is the rotating ring that is tight, but more correctly stated, it is the ring that rotates relative to the load. In some special cases this is not the rotating ring; for instance, in a vibrating unit where vibration is produced by eccentric weights, the load rotates with the rotating ring, and it is best to have the stationary ring have the tight fit.

Except for special cases as mentioned above, the stationary ring normally can be assembled with the mating shaft or housing with a slip or loose fit.

The magnitude of interference fit will vary with the severity of duty, type of bearing, and different shaft and housing materials. Ball bearings under normal load conditions will have approximately 0.00025 in. interference per inch of shaft when the inner ring is the tight fit. Roller bearings will have fits of approximately 0.0005 in. per inch of shaft. Fits will be increased for heavy-duty service and decreased for light duty. In general, when the outer ring is the tight fit, the interference is less than a corresponding shaft fit.

All bearing manufacturers show recommended fitting practices for their bearings in their general catalogs. With the exception of inch-taper roller bearings, the recommendations are normally expressed in ISO standards. ISO standards define the fit tolerance between the bearing outside diameter and the housing and utilize a designation system using a capital letter and a number such as H7, J6, P6, etc. Fit tolerances between the shaft and bore of the bearing are designated by a lowercase letter and number such as g6, m5, r7, etc. In the ISO system, the letter indicates

the class or type of fit, and the number indicates the tolerance range. The diagram in Fig. 2.7 shows the relationship between the nominal diameters and the tolerance grades. The cross-hatched areas indicate the bearing bore diameter variation and the outside diameter variation, respectively. The blackened rectangles show the range of tolerances for shafts (lower half) and housings (upper half).

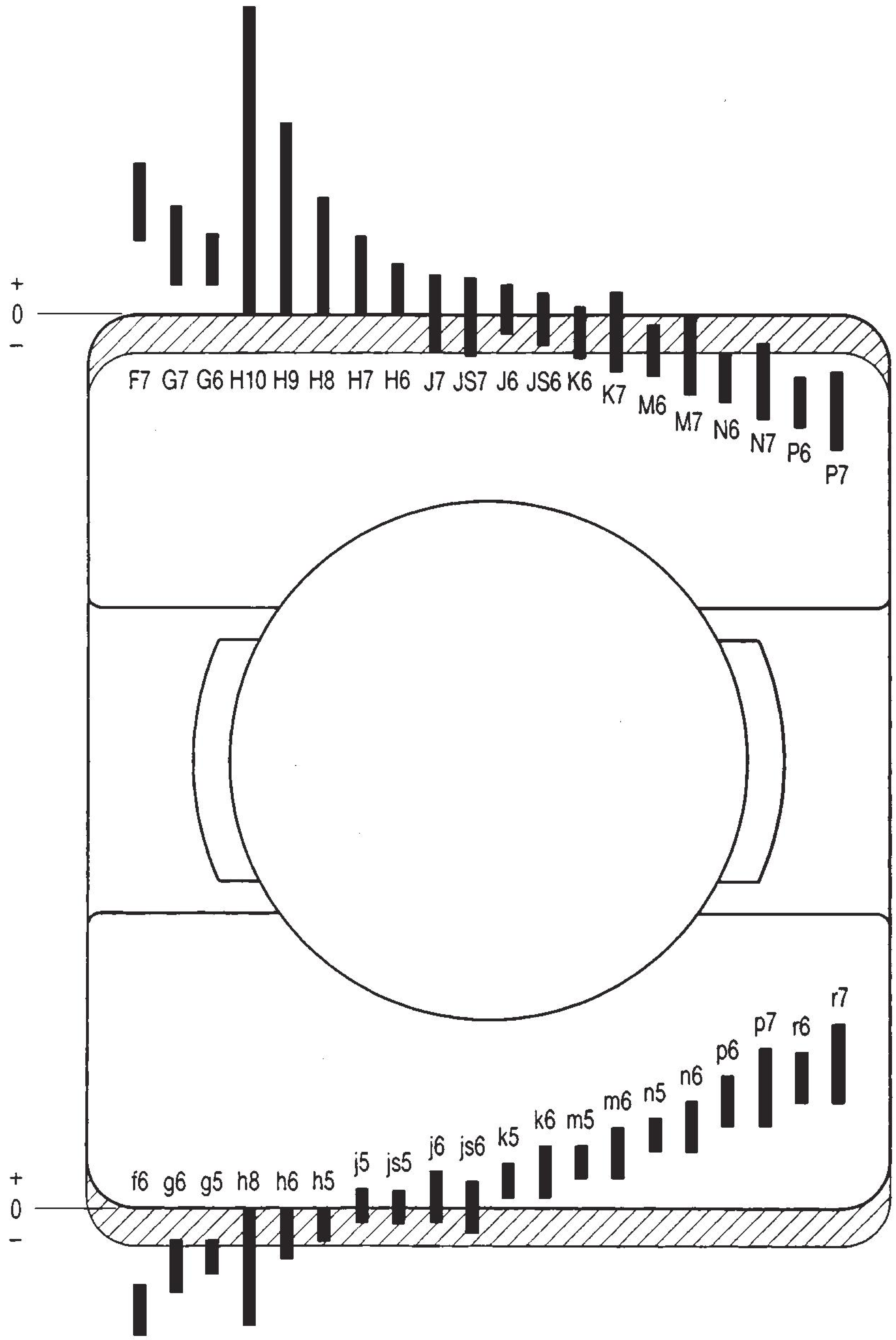


FIGURE 2.7 ISO fit tolerances. Uppercase letters refer to housings; lowercase letters refer to shafts.

BEARING MOUNTINGS

When rolling bearings are mounted on a shaft, some provision must be made for thermal expansion and/or contraction of the shaft. Also, the shaft must be located and held axially so that all machine parts remain in the proper relationship dimensionally. This is normally done by clamping one of the bearings on the shaft. When the inner ring has the tight fit, it is usually locked axially relative to the shaft by locating it between a shaft shoulder and some type of removable locking device. A specially designed nut as shown in Fig. 2.8 is normal for a through shaft. A clamp plate as shown in Fig. 2.9 is normally used when the bearing is mounted on the end of the shaft. For the locating or held bearing of the shaft, the outer ring is clamped axially, usually between housing shoulders or end-cap pilots. This type of mounting restricts axial movement of the shaft to the end movement resulting from the internal clearance of the bearing. If required, this can be zero if the appropriate bearing type is used. The outer rings on all other bearings on the shaft should not be secured axially, and enough clearance should be provided between the side face of the stationary ring and the nearest housing shoulders to allow for anticipated expansion or contraction. Typical mountings are shown in Fig. 2.10.

General types of cylindrical roller bearings are capable of absorbing shaft expansion internally simply by allowing one ring to move relative to the other, as shown in Fig. 2.10*a* and *c* for the nonlocating positions. The advantage to this type of mounting is that both inner and outer rings may have a tight fit. This may be desirable or even mandatory if significant vibration and/or unbalance exists in addition to the applied load.

Where bearing center distances are short and minimum thermal expansion is expected, an opposed mounting as shown in Fig. 2.11 may be used. In addition to its simplicity, this mounting has the advantage that thrust in one direction will be taken on one bearing and thrust in the other direction on the other bearing. Obviously, the clearance between the side face of the bearing and the housing shoulder must be controlled carefully or the shaft will shift excessively in an axial direction with the load direction changes.

Single-row tapered roller bearings and angular-contact ball bearings require special consideration. For example, if a radial load is applied to a single-row tapered roller bearing, an axial component of the load is generated by the angle of the roller set. This tends to separate the inner ring from the outer ring unless the "induced thrust" is resisted by another bearing properly mounted to resist the movement. The other bearing is normally another single-row tapered roller bearing. A mounting

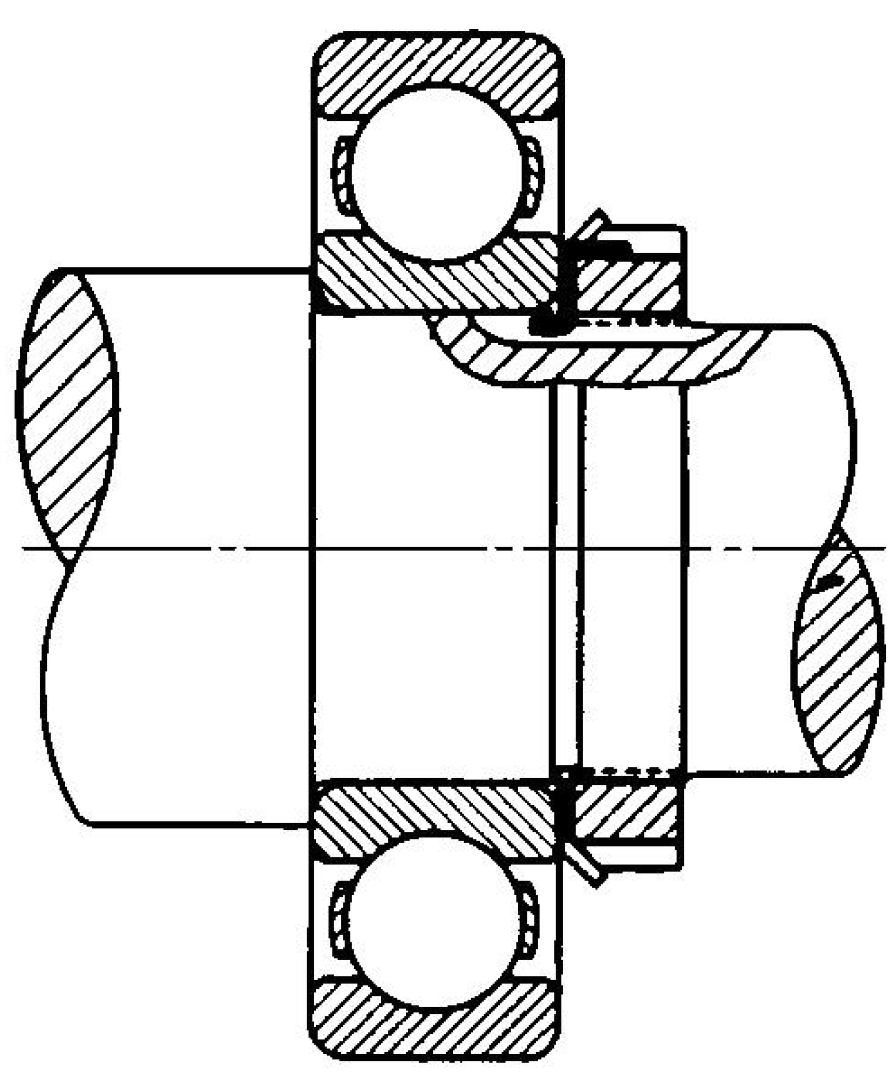


FIGURE 2.8 Bearing mounting with a special nut for a through shaft.

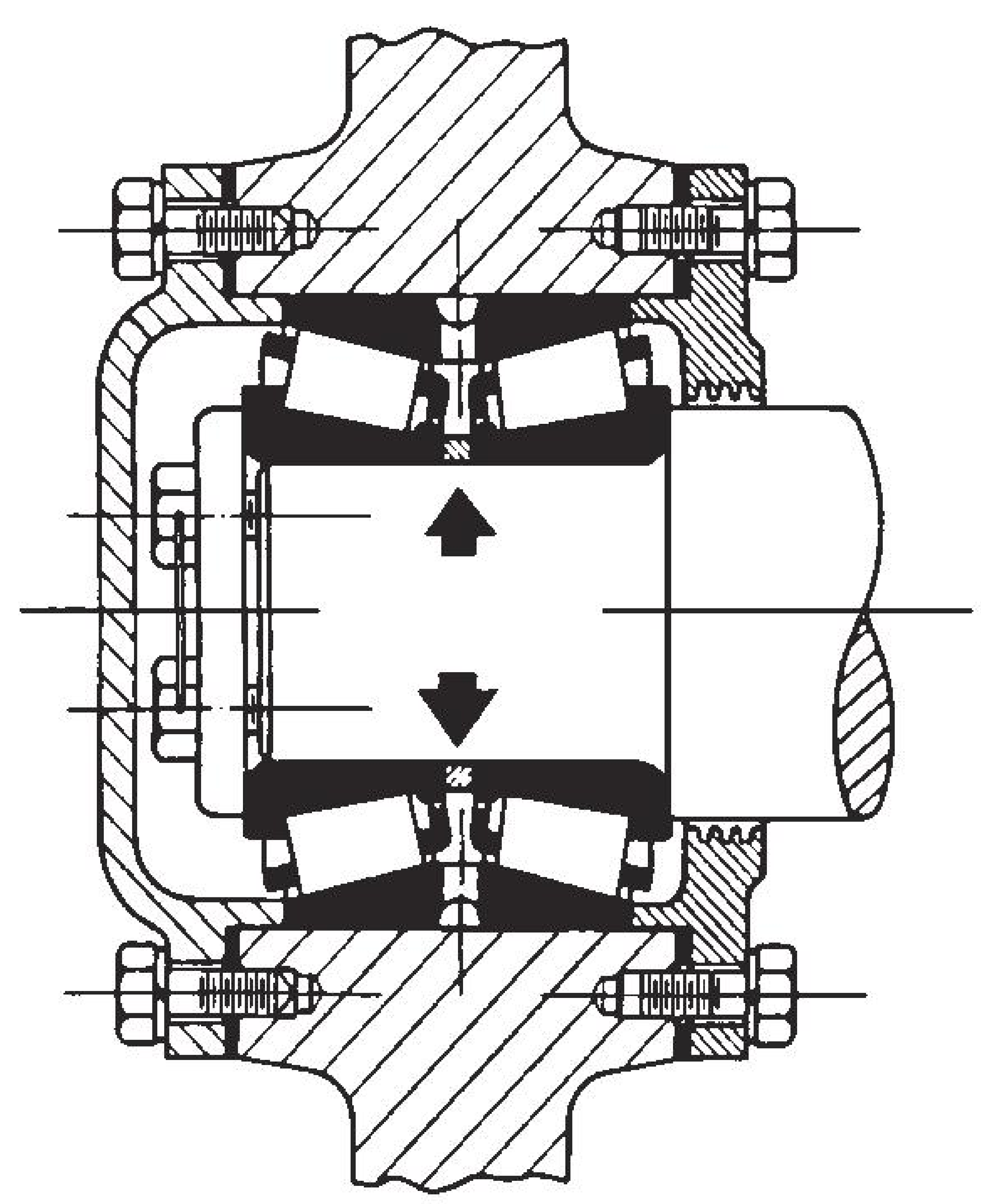


FIGURE 2.9 Bearing mounting with a clamp plate locking device using an inner ring adjustment spacer.

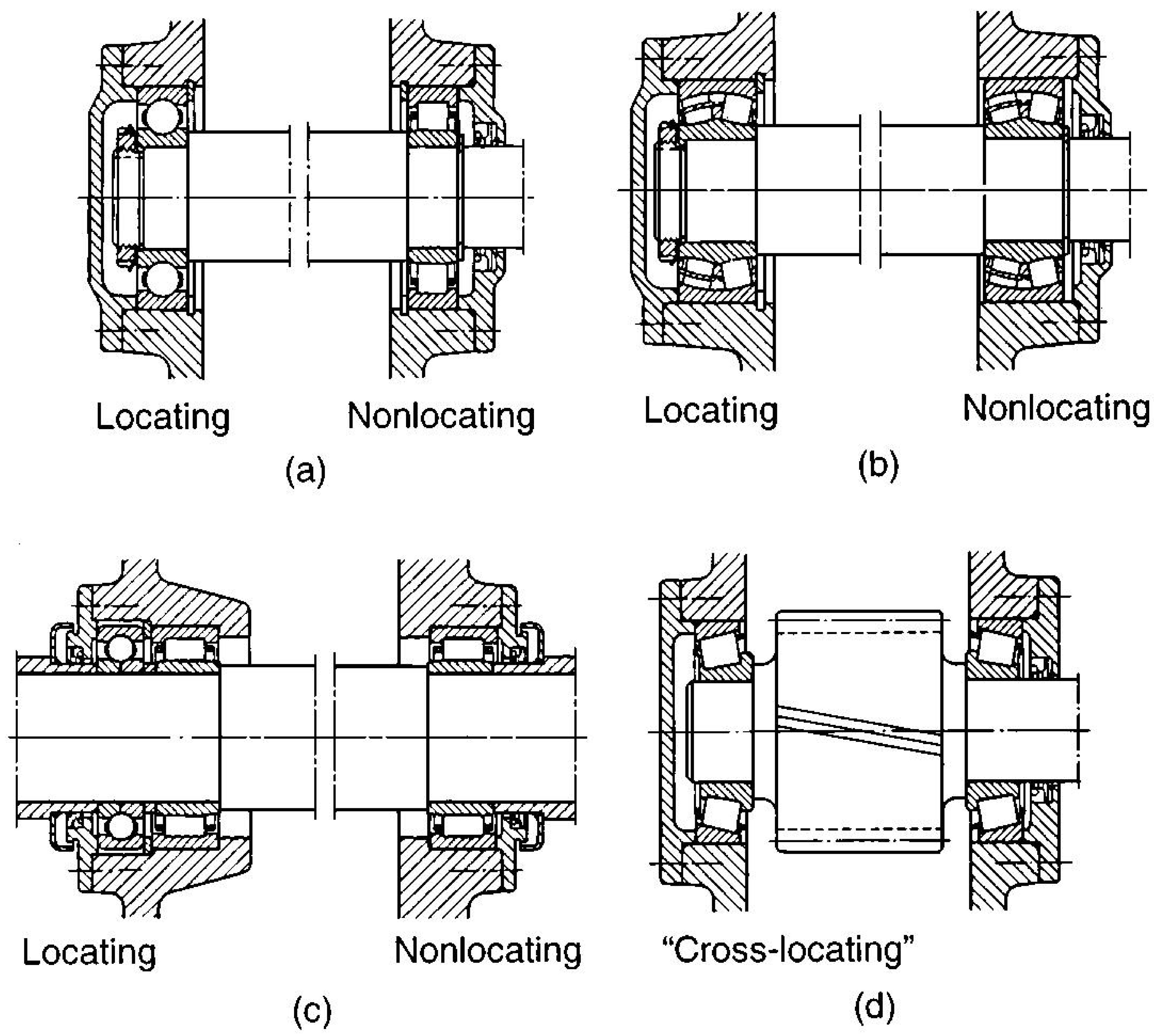


FIGURE 2.10 Typical bearing mountings.

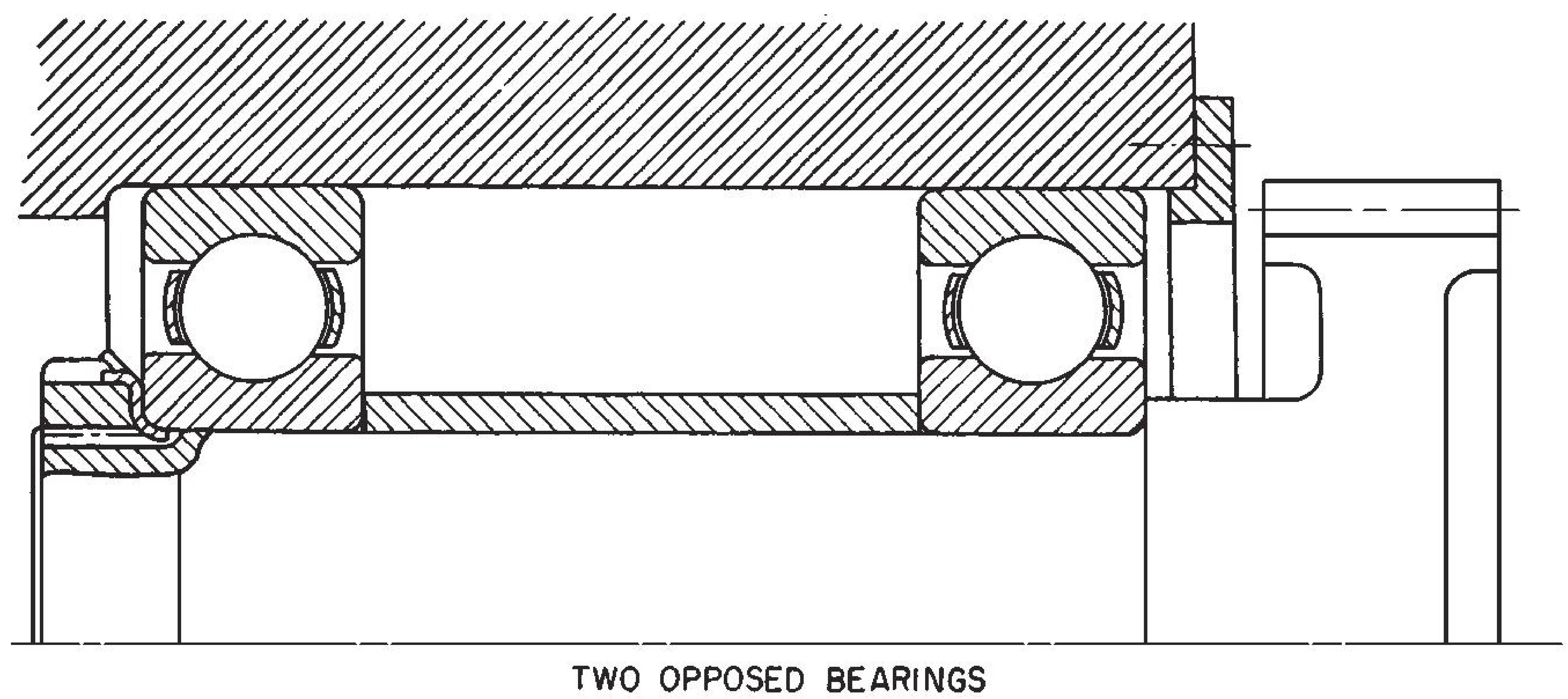


FIGURE 2.11 Example of assembly with opposed mounting of ball bearings.

of this type may have the bearings arranged in one of two ways, as shown in Fig. 2.12. Figure 2.12*a* shows the included angles of the bearing or the tracks opening away from each other. This is known as an *indirect mounting.* Figure 2.12*b* shows the included angles of the bearings opening toward each other. This is known as *direct mounting.* It should be noted before progressing further with a description of this type of mounting that the point of reaction of the load on the centerline of the shaft, or the *effective center,* is not at the geometric center of a single-row bearing but at some point 0 as determined by the angle of the rolling element centerline relative to the centerline of the shaft.

Therefore, if the bearings of two different mountings are physically located the same distance apart with an indirect mounting, the effective centers of the bearings are farther apart than with the direct mounting, a more desirable arrangement when an overturning load exists.

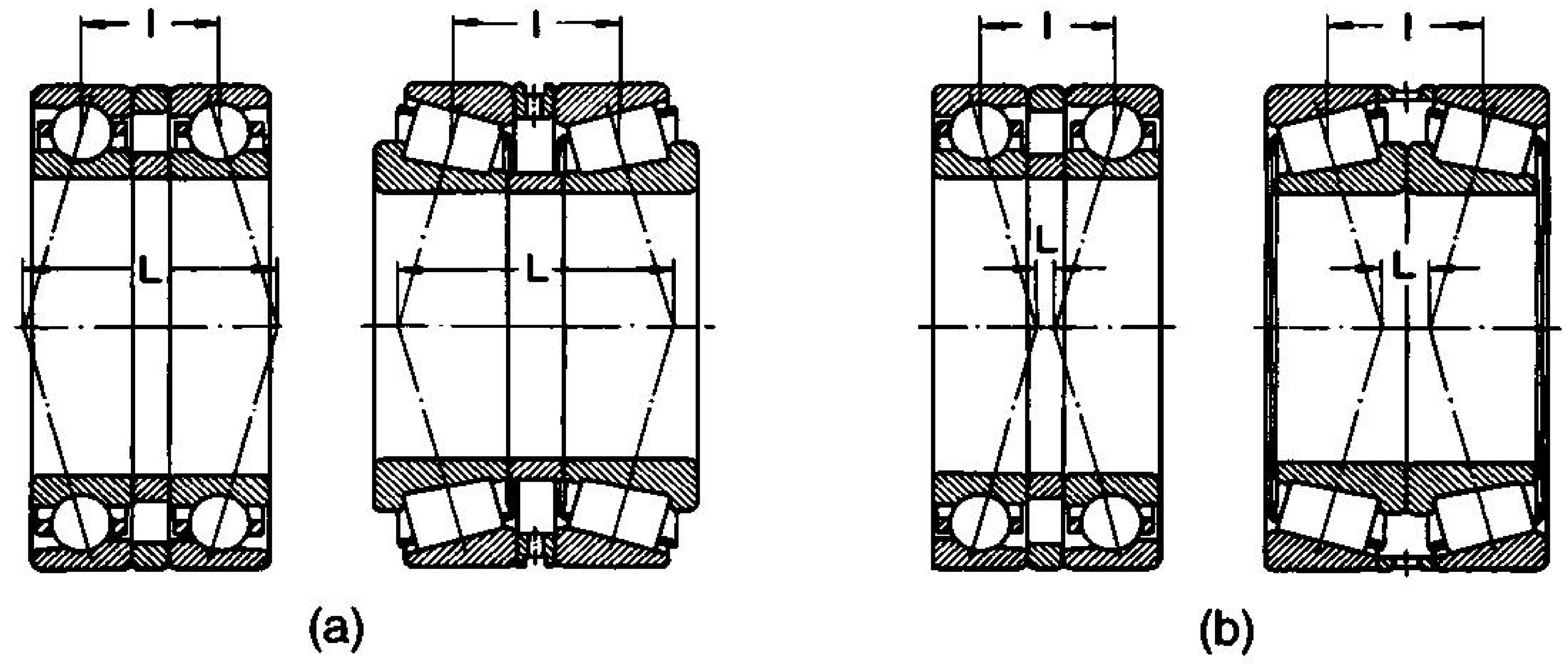


FIGURE 2.12 Direct (*a*) and indirect (*b*) arrangements.

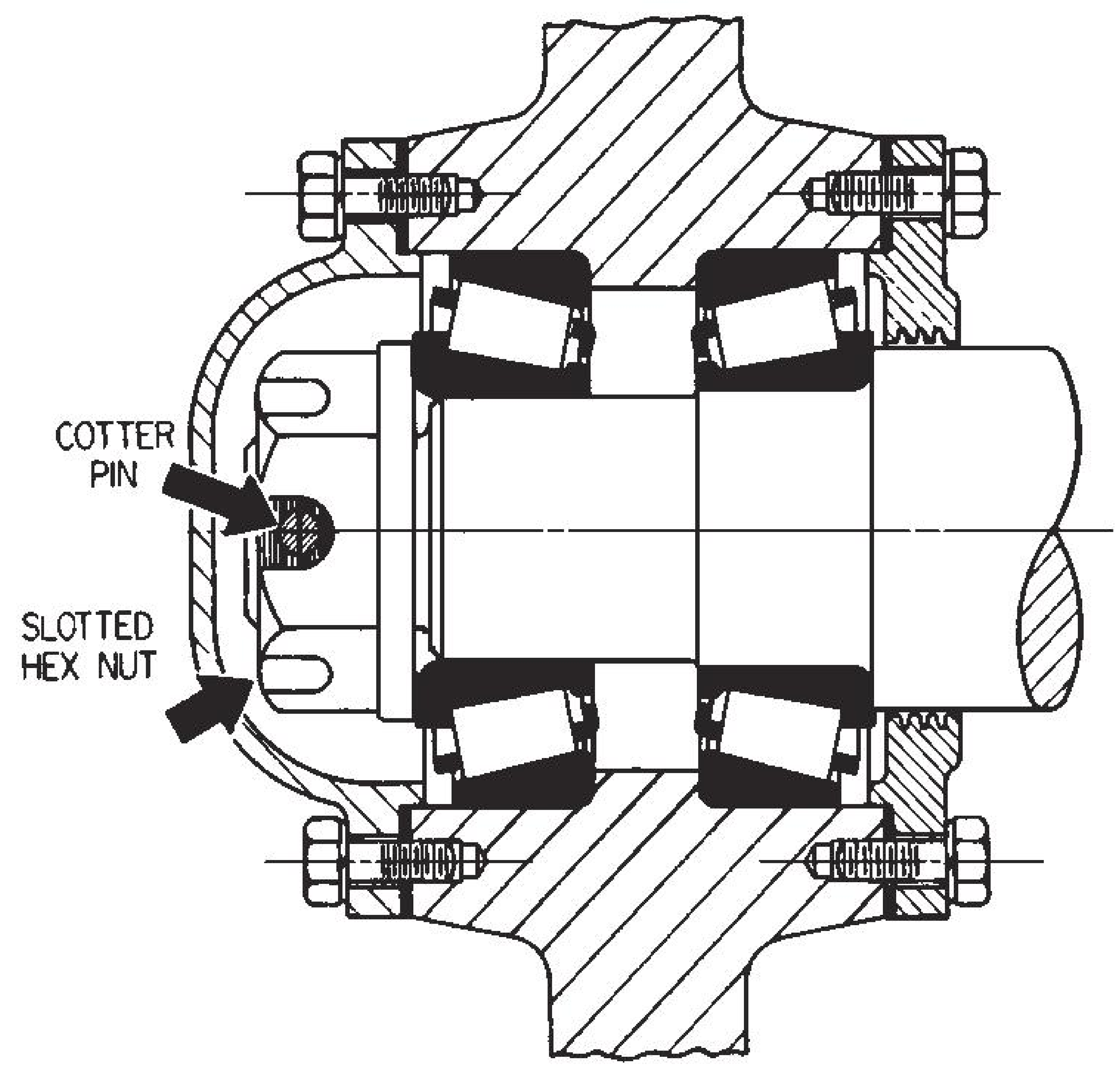


FIGURE 2.13 Slotted-nut adjusting device.

With either a direct or an indirect single-row tapered roller bearing mounting, it is necessary to set the running clearance of the bearings when they are assembled. This is done by adjusting the cones in an indirect mounting and the cups for a direct mounting. Figures 2.13 and 2.14 show two ways of adjusting cones by nuts, and Fig. 2.15 shows a method of shimming for cone adjustment. Figures 2.16 to 2.18 show three ways of shimming cups in a direct mounting. Proper running clearance is controlled by measuring the end movement, or *end lateral,* of the shaft. The machine builder's recommendation for proper end lateral should be strictly followed. It will usually be indicated on the drawing of the particular part or given in the maintenance manual for the equipment.

Obviously, the only provision for thermal expansion in either of these mountings is the end lateral of the assembly. For this reason, they should be used only where bearing centers are relatively short or where little temperature variation is anticipated.

Two-row tapered roller bearings are mounted the same as other types of bearings. Proper end lateral is preadjusted in the factory.

Angular-contact ball bearings are rarely used singly. However, if they are, they must be mounted in a similar manner to single-row tapered roller bearings. The much smaller running clearances used in ball bearings make a mounting of single angular-contact ball bearings very difficult to adjust properly. Angular-contact bearings could be substituted for the tapered roller bearings of Fig. 2.13, and the same comments and nomenclature would apply for single-bearing mountings.

However, angular-contact ball bearings are normally used in pairs. The side faces of these bearings are specially manufactured to permit mounting side by side, as shown in Fig. 2.19. *Tandem*

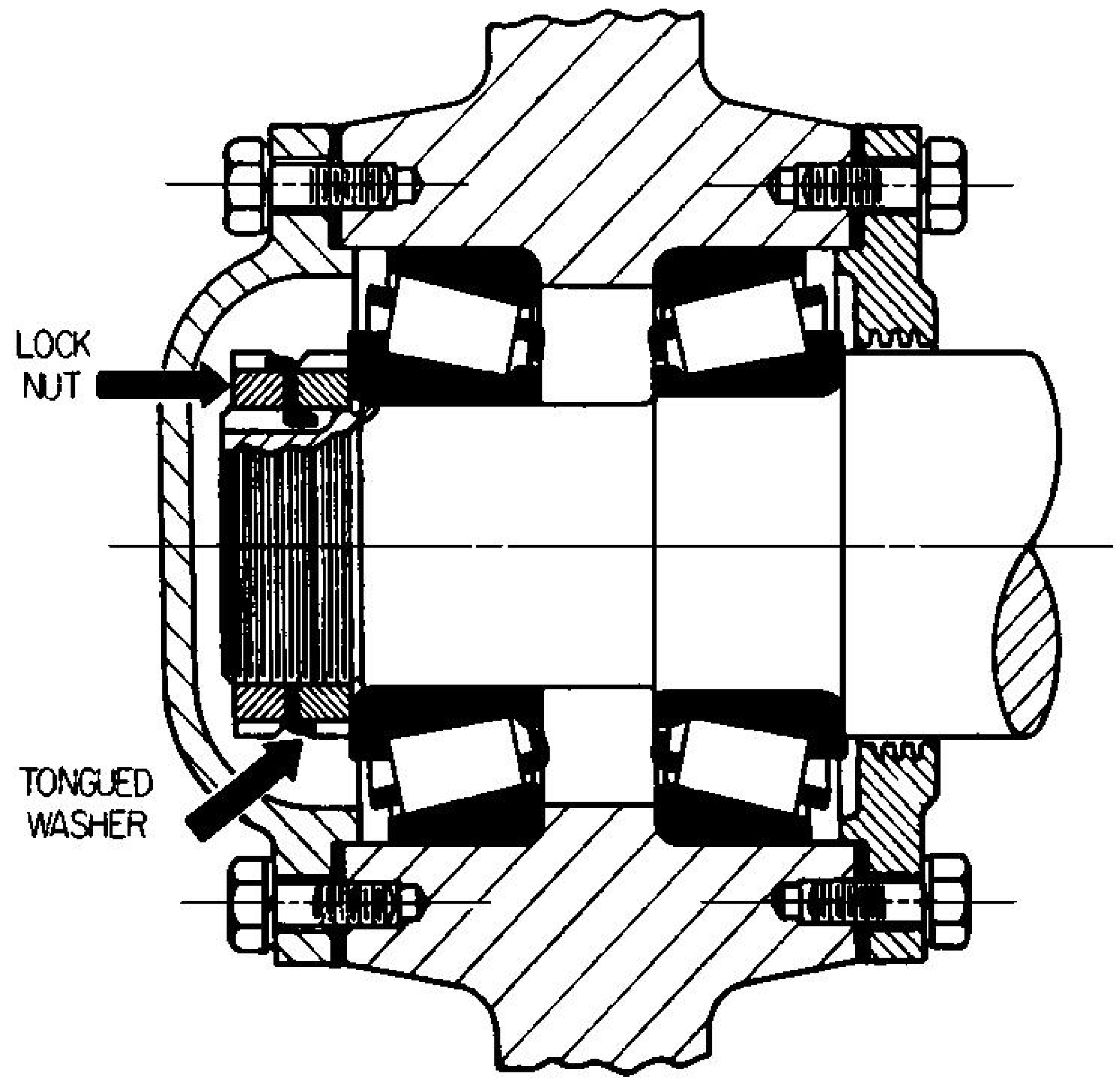


FIGURE 2.14 Double-nut and lock-washer adjusting device.

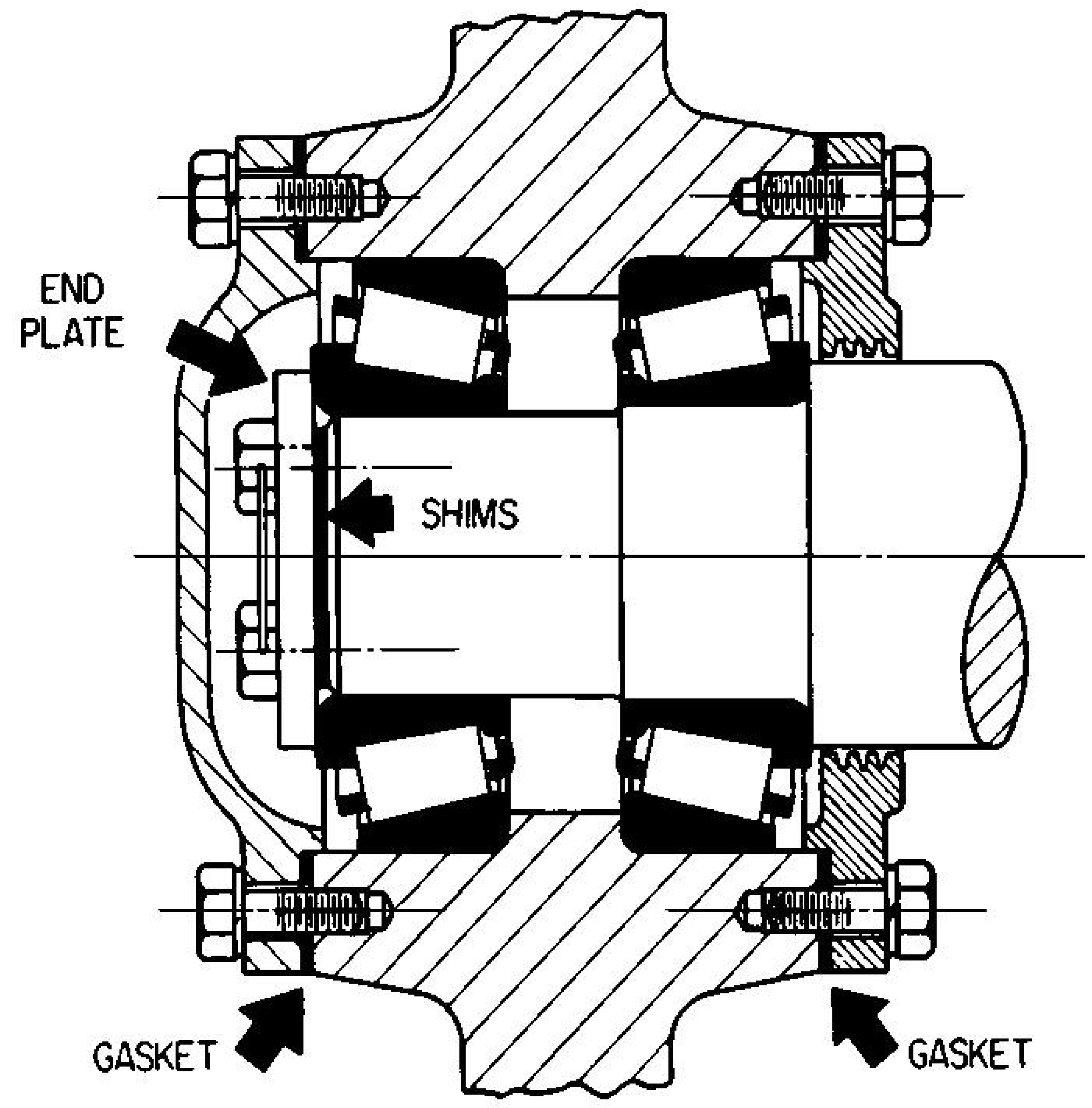


FIGURE 2.15 End-plate and shims adjusting device.

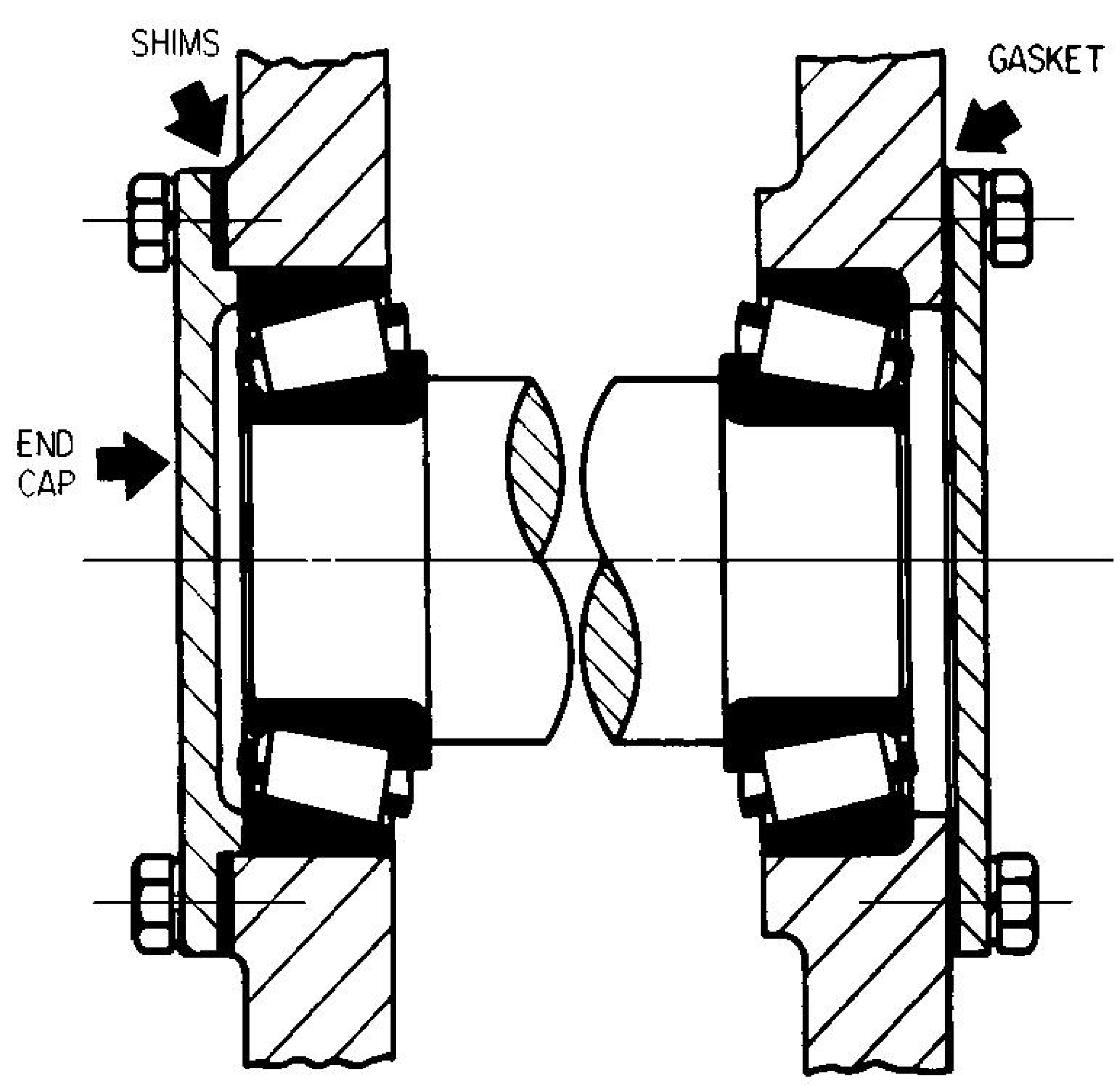


FIGURE 2.16 End-cap and shims adjusting method.

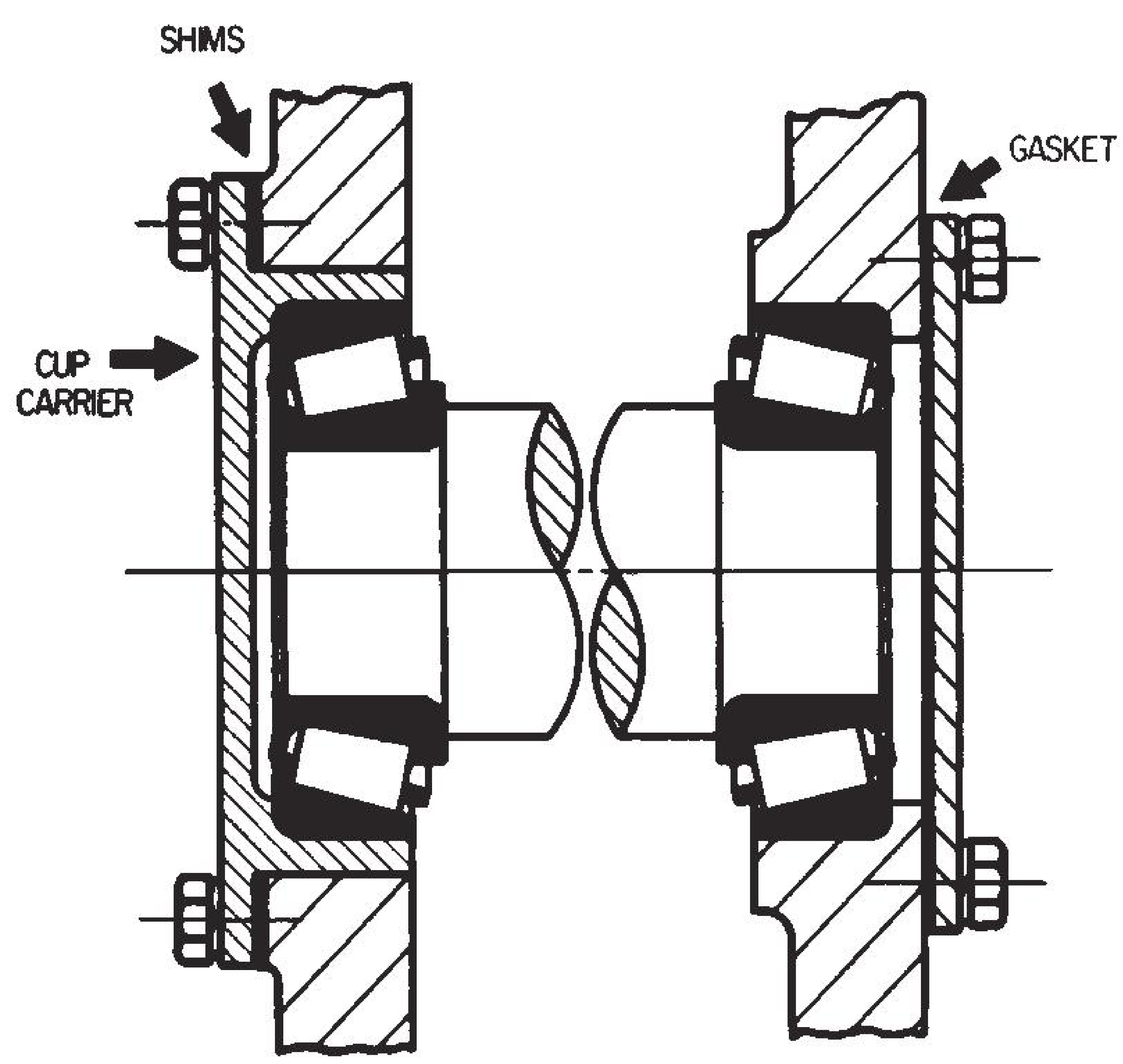


FIGURE 2.17 Cup-carrier and shims adjusting method.

(Fig. 2.19*a*), *back-to-back* (Fig. 2.19*b*), and *face-to-face* (Fig. 2.19*c*) are the common terms for these mountings. When two or more bearings are "stacked" in tandem for high-thrust loads, usually another bearing in the assembly is mounted face-to-face or back-to-back with the tandem "stack." When mounted in any of these arrangements, they may be considered as one multiple-row bearing. Because methods of face modifications may differ from one manufacturer to another, it is advisable not to mix manufacturers in a pair of tandem bearings. The bearing assembly number should indicate in some way that the bearings have been properly manufactured for mounting in pairs. Bearings for single mounting are available and should not be used as part of a pair.

A large percentage of spherical roller bearings are made with tapered bores. Some ball, tapered roller, and cylindrical roller bearings are also available with tapered bores. The bearings may be mounted directly on the shaft, as shown in Fig. 2.20. However, many tapered bore bearings are

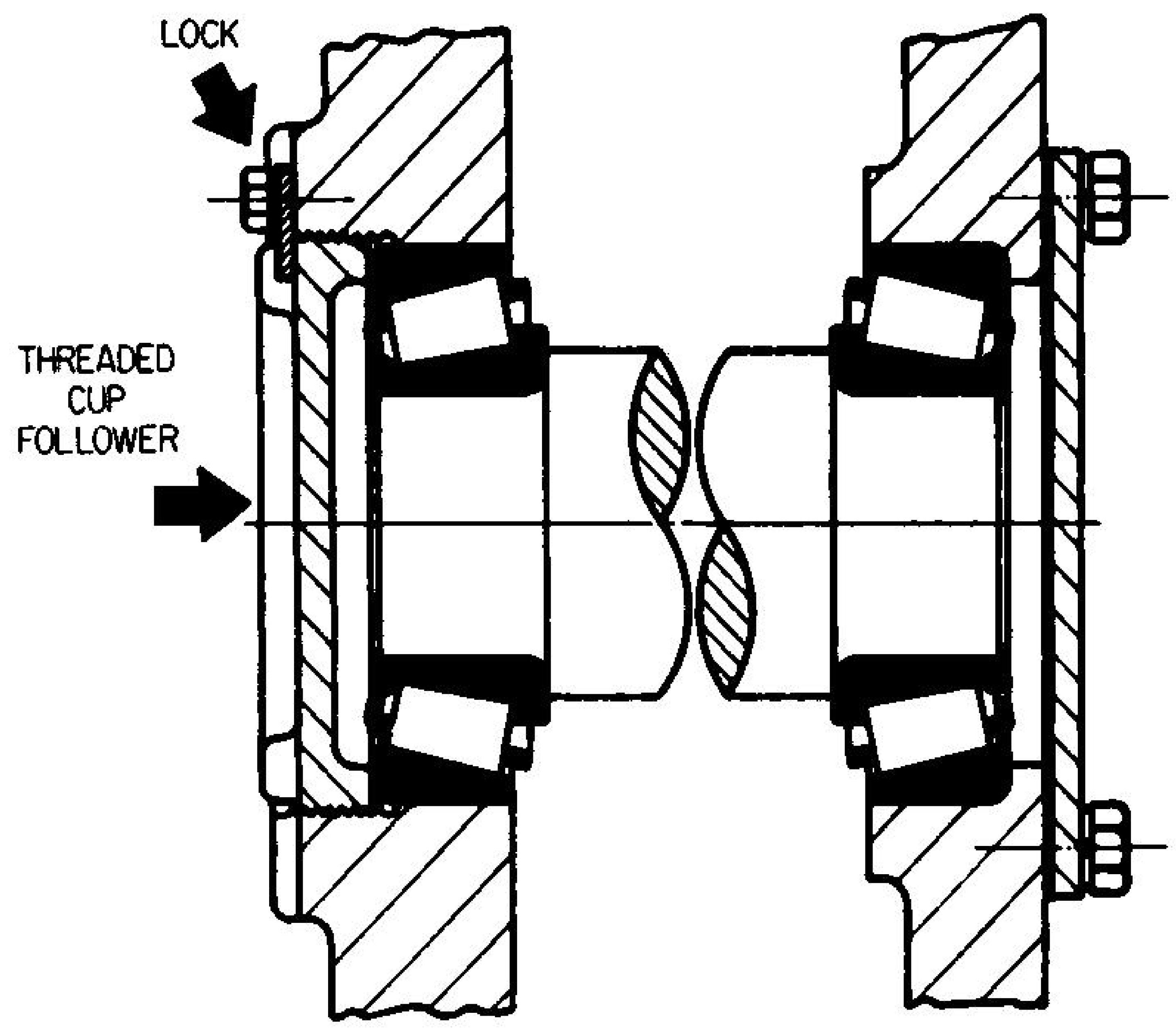


FIGURE 2.18 Threaded-cup-follower adjusting method.

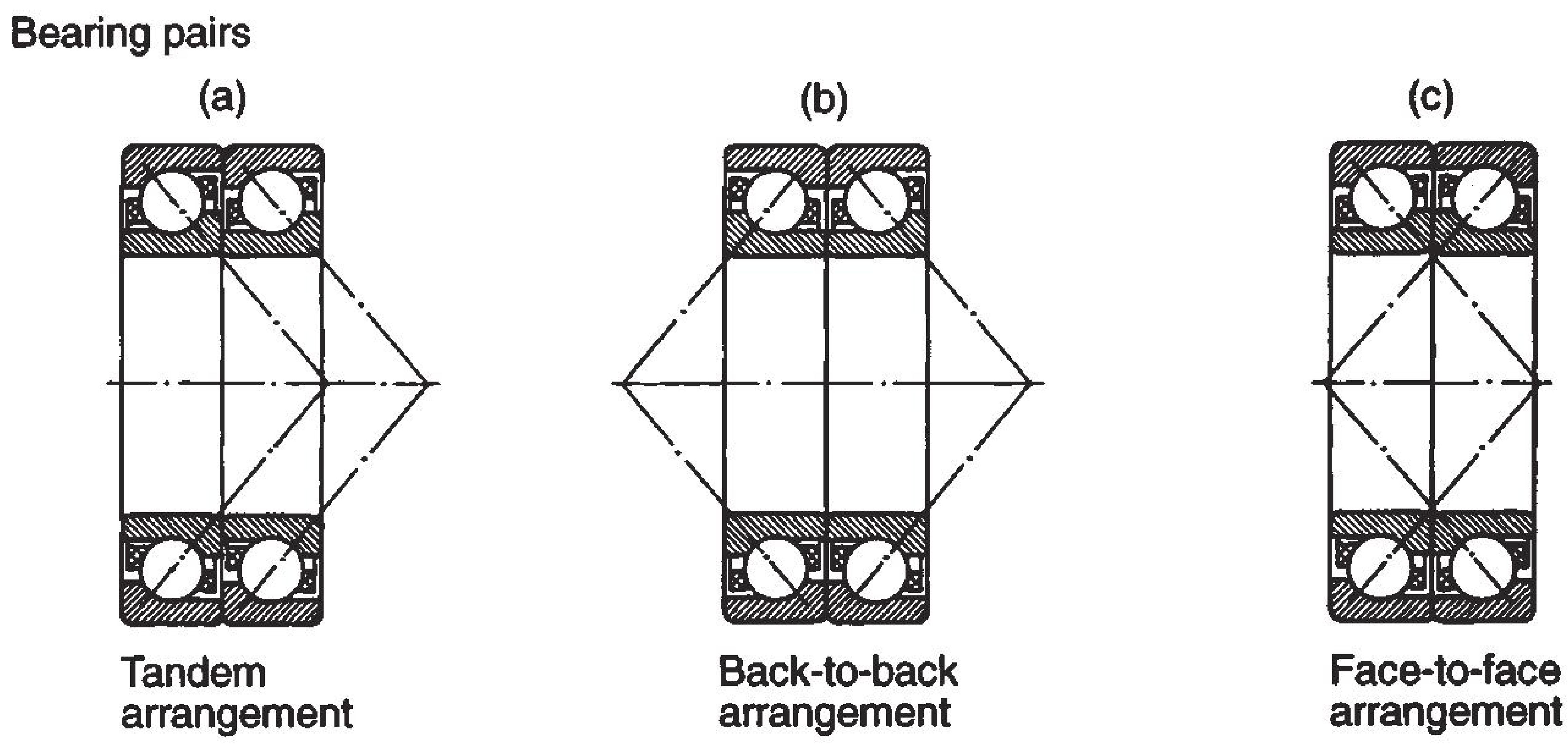


FIGURE 2.19 Angular contact mounting arrangements.

mounted on one of two types of sleeves, as shown in Figs. 2.21 and 2.22. European machinery builders are particularly partial to the use of sleeve mountings.

The adapter sleeve may be mounted as shown in Fig. 2.21 or with a shaft shoulder ring as shown in Fig. 2.23. With a removable type of sleeve, as shown in Fig. 2.22, the bearing must always be against a shaft shoulder.

The taper is 1 to 12 on diameter in all but the widest series of spherical roller bearings, in which a flatter 1 to 30 taper is used. Some four-row cylindrical roller rolling-mill bearings also will use a 1 to 30 taper in the bore of the inner ring.

MOUNTING AND DISMOUNTING OF ROLLER BEARINGS

The most important thing to remember when mounting or dismounting a roller bearing, of any type, is to apply the mounting or dismounting force to the side face of the ring with the interference fit. Keep this force from passing from one ring to the other through the ball or roller set. This is particularly

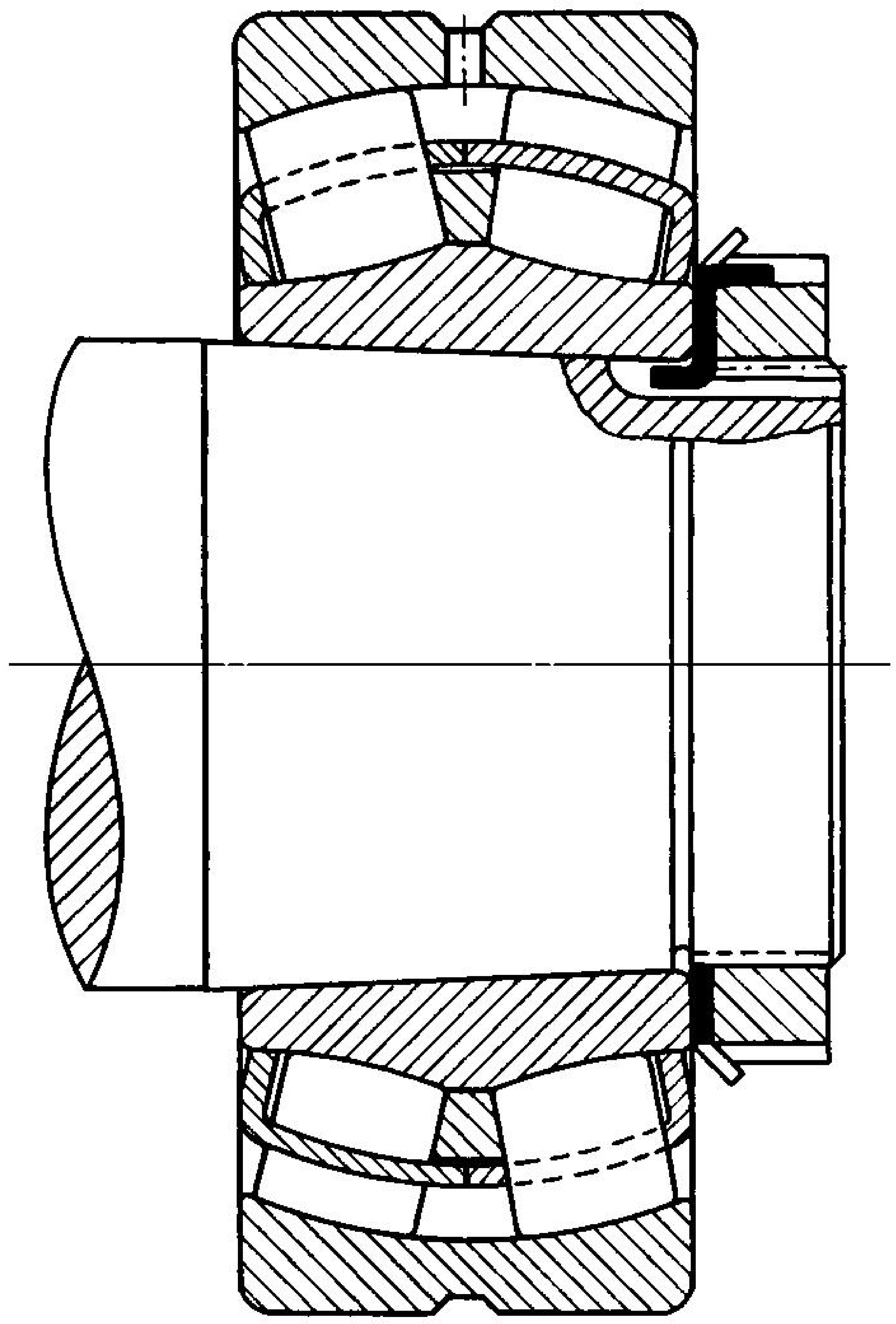


FIGURE 2.20 Direct shaft mounting of a spherical roller bearing.

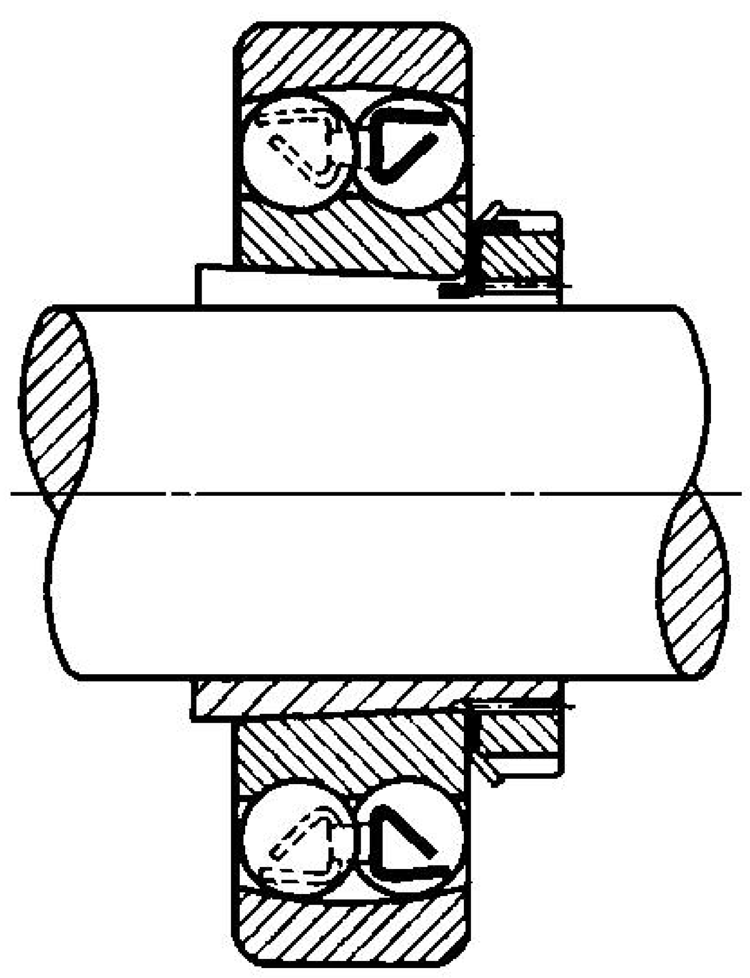


FIGURE 2.21 Mounting of a self-aligning ball bearing with an adapter sleeve.

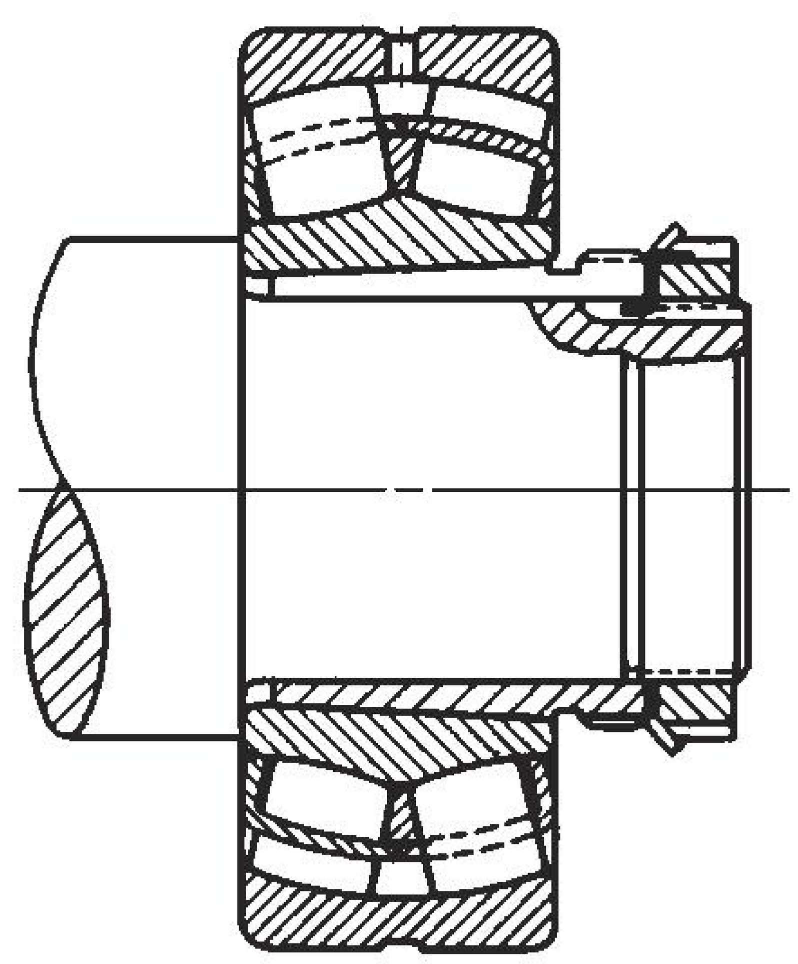


FIGURE 2.22 Mounting of a spherical roller bearing with removable type of sleeve.

important during mounting, since damage can easily occur internally to the bearing. Cleanliness is, of course, extremely important. Not only the bearing but also the shaft housing must be free from chips, burrs, dirt, and moisture.

Bearings should be kept wrapped or covered until the last possible moment. Since most modern rust preventives used by bearing manufacturers are compatible with petroleum-based lubricants, the slushing compound is normally not removed. However, there are exceptions to this rule. If oil-mist lubrication is to be used and the slushing compound has hardened in storage or is blocking lubrication holes

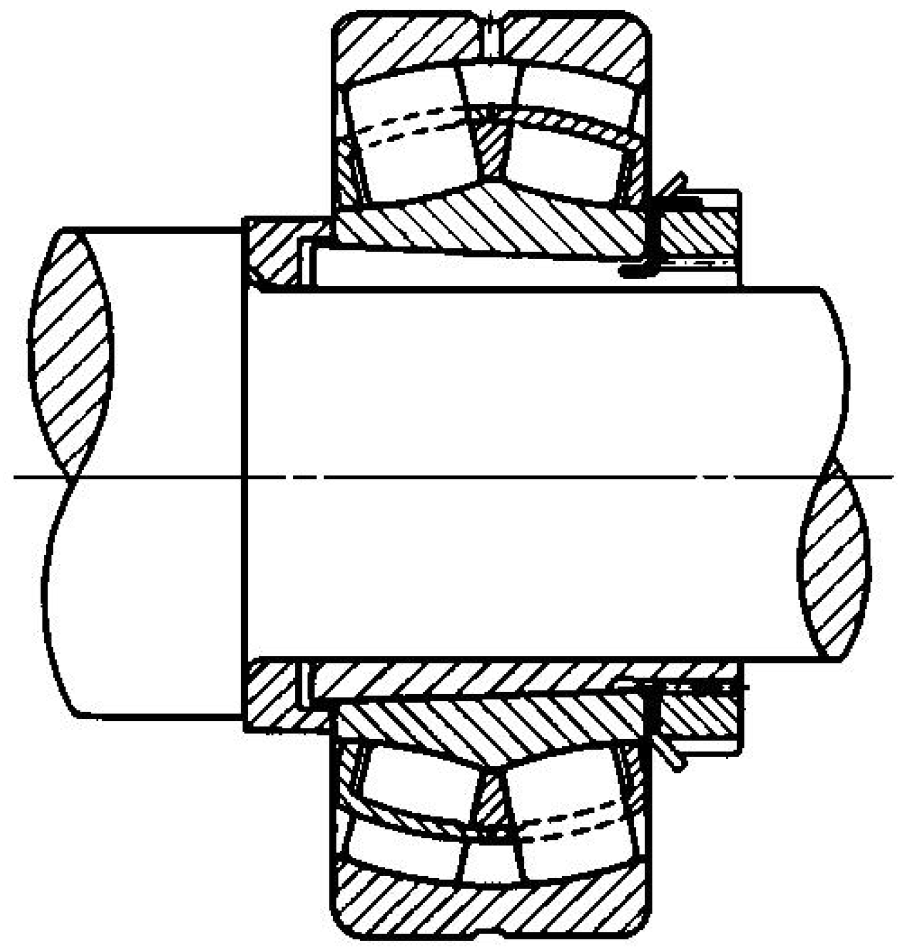


FIGURE 2.23 Mounting with shaft shoulder ring.

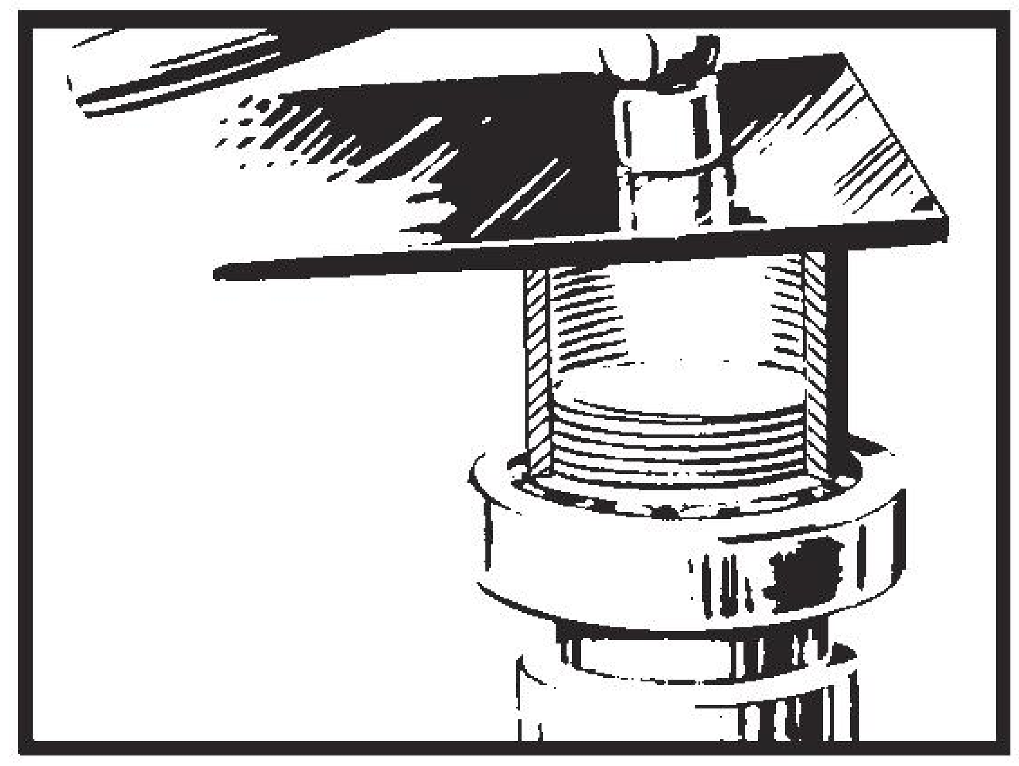


FIGURE 2.24 Mounting using flat plate.

in the bearing rings, it is best to clean the bearing with kerosene or other appropriate petroleum-based solvent. Obviously, the other exception would be if the slushing compound has been contaminated with dirt or foreign matter before mounting. It is also permissible and sometimes desirable to wipe the rust preventive from the bore or outside diameter of the bearing, depending on which surface will have the tight fit. Before mounting or dismounting a bearing, always take the time to collect the proper tools and accessories. The use of inappropriate tools is a major cause of bearing damage. Also remember, never strike a bearing directly with a hammer, sledge, or mallet.

Cold Mountings. All small bearings (4-in. bore and smaller) may and sometimes must be mounted cold by simply forcing them on the shaft or into the housing. However, it is important that this force be applied as uniformly as possible around the side face of the bearing and to the ring to be press-fitted. Mounting fixtures should be used. These can be a simple piece of tubing of appropriate size and a flat plate as shown in Fig. 2.24. Do not try to use a drift and hammer, because the bearing will become cocked. Force may be applied to the simple fixture described above by striking the plate with a hammer or by an arbor press, as shown in Fig. 2.25. It is a good idea to apply a coat of light oil to the bearing seat on the shaft and bore of the bearing itself before forcing on the shaft. It should be noted that all sealed and shielded ball bearings should be mounted cold in this manner.

Temperature Mountings. The simplest way to mount any open straight-bore bearing, no matter what size, is to heat the entire bearing and simply push it on its seat and hold in place until it cools enough to start gripping the shaft. For tight outside-diameter fits, the housing may be heated if practical; if not, the bearing may be cooled by dry ice. However, if the ambient conditions are humid, cooling the bearing introduces the possibility of condensation on the bearing, which will induce corrosion later.

There are several acceptable ways of heating bearings. Some of these are as follows:

1. Hot plate. A bearing is simply laid on an ordinary hot plate until it reaches the approved temperature. The disadvantage of this method is that the temperature is difficult to control. A Tempilstik or pyrometer should be used to make certain the bearing is not overheated.
2. The temperature-controlled oven. This method needs little comment. The bearings should be left in the oven long enough to heat thoroughly. However, never leave bearings in a hot over overnight or over a holiday or weekend.
3. Induction heaters are available which can be used to heat bearings for mounting. One of these is shown in Fig. 2.26. It must be remembered that this is a very quick method of heating and that some method of measuring the ring temperature must be used or the bearing may be damaged. A Tempilstik or pyrometer can serve this purpose. Bearings must be demagnetized after using this method.

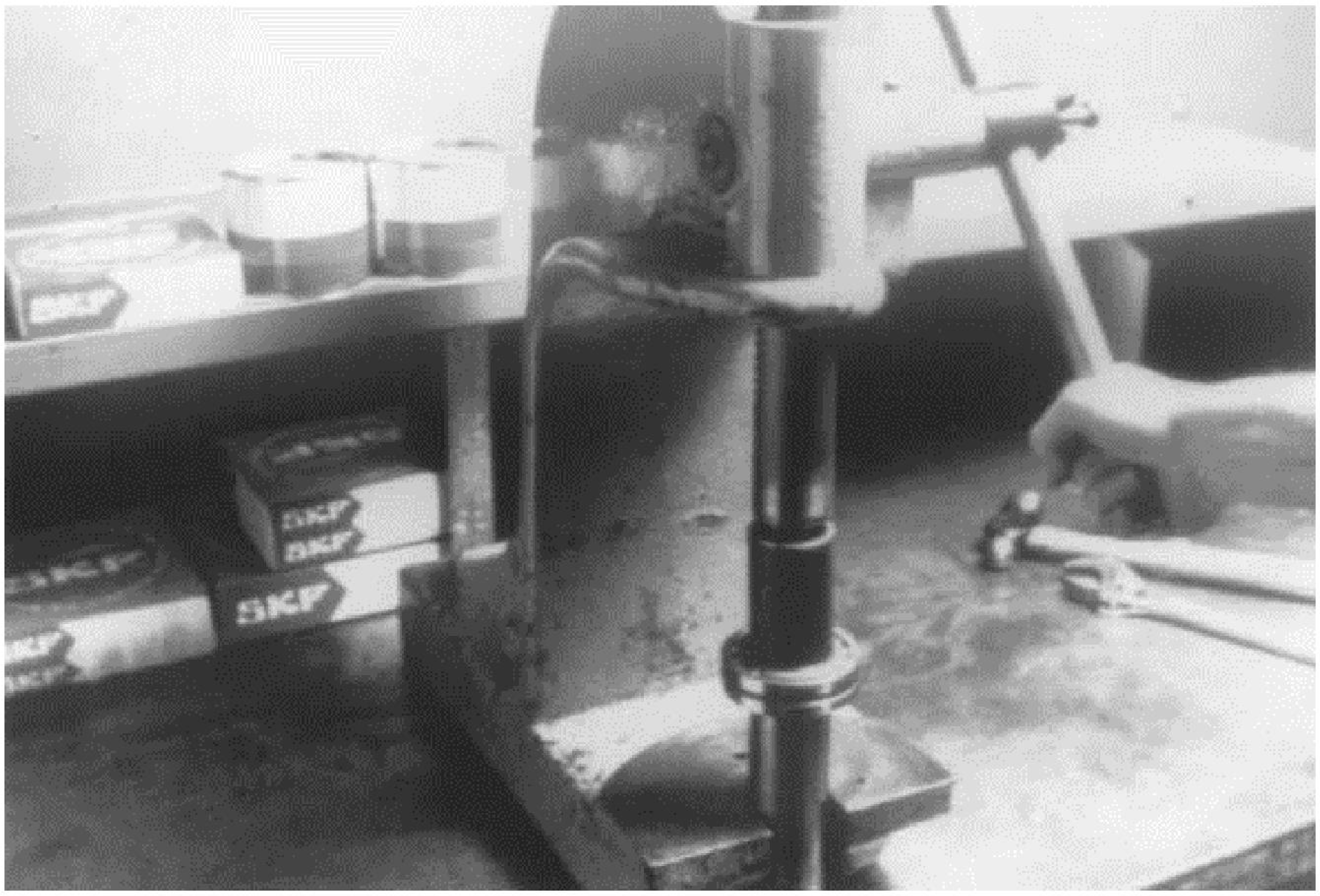


FIGURE 2.25 Arbor press.

FIGURE 2.26 Induction heater.

4. A hot-oil bath also may be used to heat the bearing and, in fact, is the most practical means to heat larger bearings. This method has some drawbacks, since the temperature of the oil is difficult to control and may overheat the bearing or even become a fire hazard. A mixture of soluble oil and water can eliminate both these disadvantages. Make the mixture 10 to 15 percent soluble oil. This solution will boil at approximately 210°F, which is hot enough for most bearing fits.

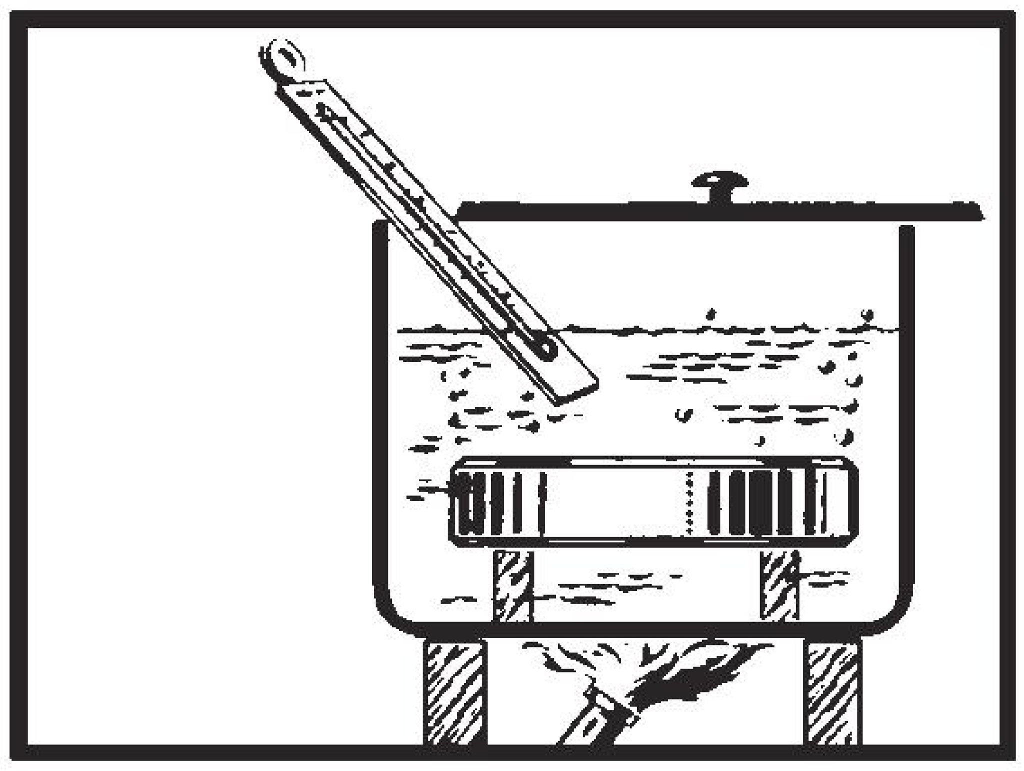


FIGURE 2.27 Hot oil for bearing.

The heating solution should be placed in a tank or container which has a grate or screen several inches off the bottom, as shown in Fig. 2.27. This will allow any contaminants to sink to the bottom and keeps the bearings off the bottom of the container.

As mentioned above, 210°F is not enough to mount most bearings. If you are using one of the other methods of heating or another solution, 250°F maximum will do the bearings no harm. However, this temperature should not be exceeded for small ball bearings (2-mm bore and smaller). Larger bearings can be heated somewhat higher than this without harm, but metallurgical damage will occur at approximately 300°F.

Mounting Tapered-Bore Bearings. Tapered-bore bearings can be mounted simply by tightening the locknut or clamping plate, which will locate it on the shaft until the bearing has been forced up the taper the proper distance. However, especially for large bearings, this technique will require a good amount of brute force. There are special techniques that may be used to reduce the amount of force required.

Before reviewing the mounting techniques for tapered-bore roller bearings, we will discuss the special case of self-aligning ball bearings. The bearing should be put on its tapered seat and the locknut hand tightened until all looseness is removed between adjacent parts. Then, using a spanner wrench, not a drift and a hammer, tighten the nut one-eighth turn further. Bend a lock-washer tab into the nut slot nearest to a washer tab in a tightened direction. At this point, the outer ring should rotate as well as swivel freely.

Tapered-bore spherical roller bearings can be mounted a bit more scientifically. Since the internal clearance in a roller bearing is significantly larger than in a ball bearing, this clearance can be measured with a thickness feeler gauge. As the bearing inner ring is pushed up the tapered seat, the inner ring expands, thereby reducing the internal clearance. Hence the amount of this reduction is a direct function of the interference fit between the bore of the bearing and the shaft. Therefore, if we measure the internal clearance of the bearing unmounted and control the amount the clearance is reduced during mounting, we control the shaft fit within very close limits. The internal clearance of a spherical roller bearing is measured as follows:

The bearing is unwrapped and placed on a table so that it can be easily handled. With one hand grasping the lower portion of the inner ring, oscillate the inner ring and roller set in a circumferential direction to seat the lower rollers properly in the sphere of the outer ring, on the roller paths of the inner ring, and against the separate guide ring between the two rows of rollers. Select a gauge blade of perhaps 0.003- or 0.004-in. thickness or less for small bearings. The usable length of the blade should be somewhat longer than the length of a roller. It should not be equal to or greater than the width of the bearing. While pushing the top roller against its guiding surface, inset the blade between two rollers and the outer ring and slide the blade circumferentially toward the roller at the top of the bearing, as shown in Fig. 2.28. The blade should pass between the uppermost roller and the inside of the outer ring. Do this with successively thicker feeler blades until a blade will not pass. Move it so that it approaches the bite between a roller and the outer ring sphere; then, with one hand grasping the inner ring as described earlier, slowly roll the uppermost roller under the feeler blade.

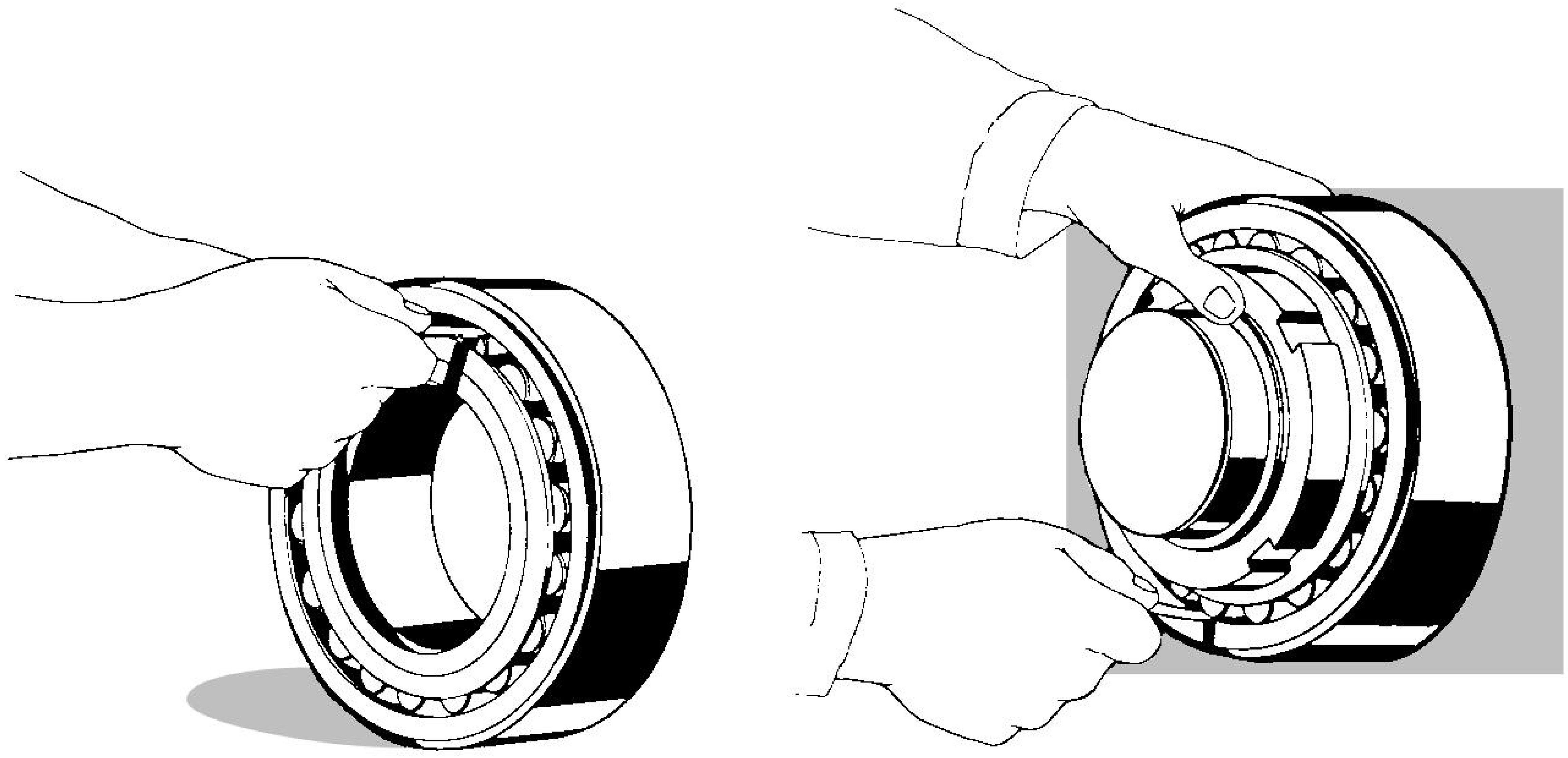


FIGURE 2.28 Determining internal bearing clearance for a spherical roller bearing.

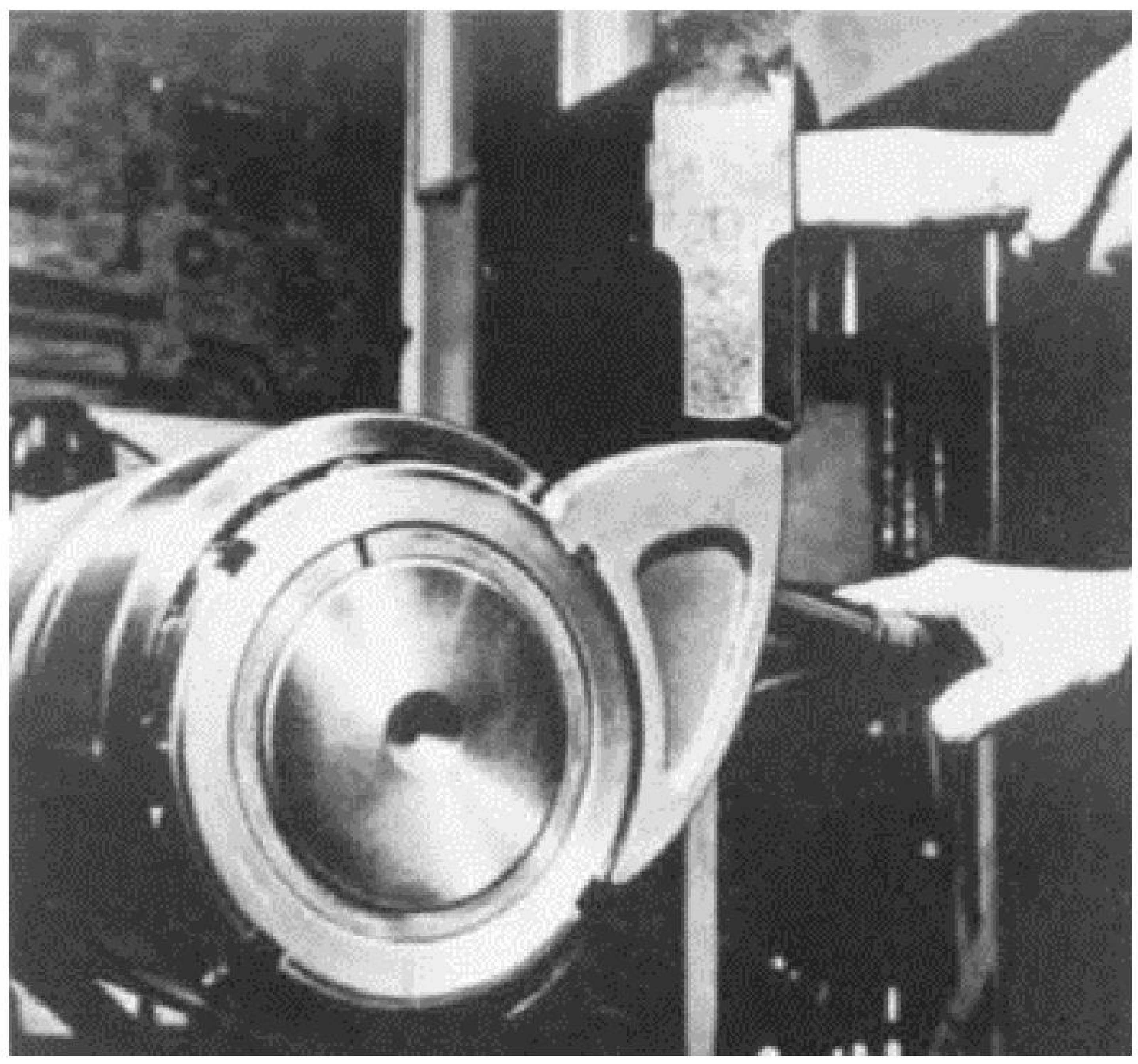


FIGURE 2.29 Use of impact-type spanner wrench.

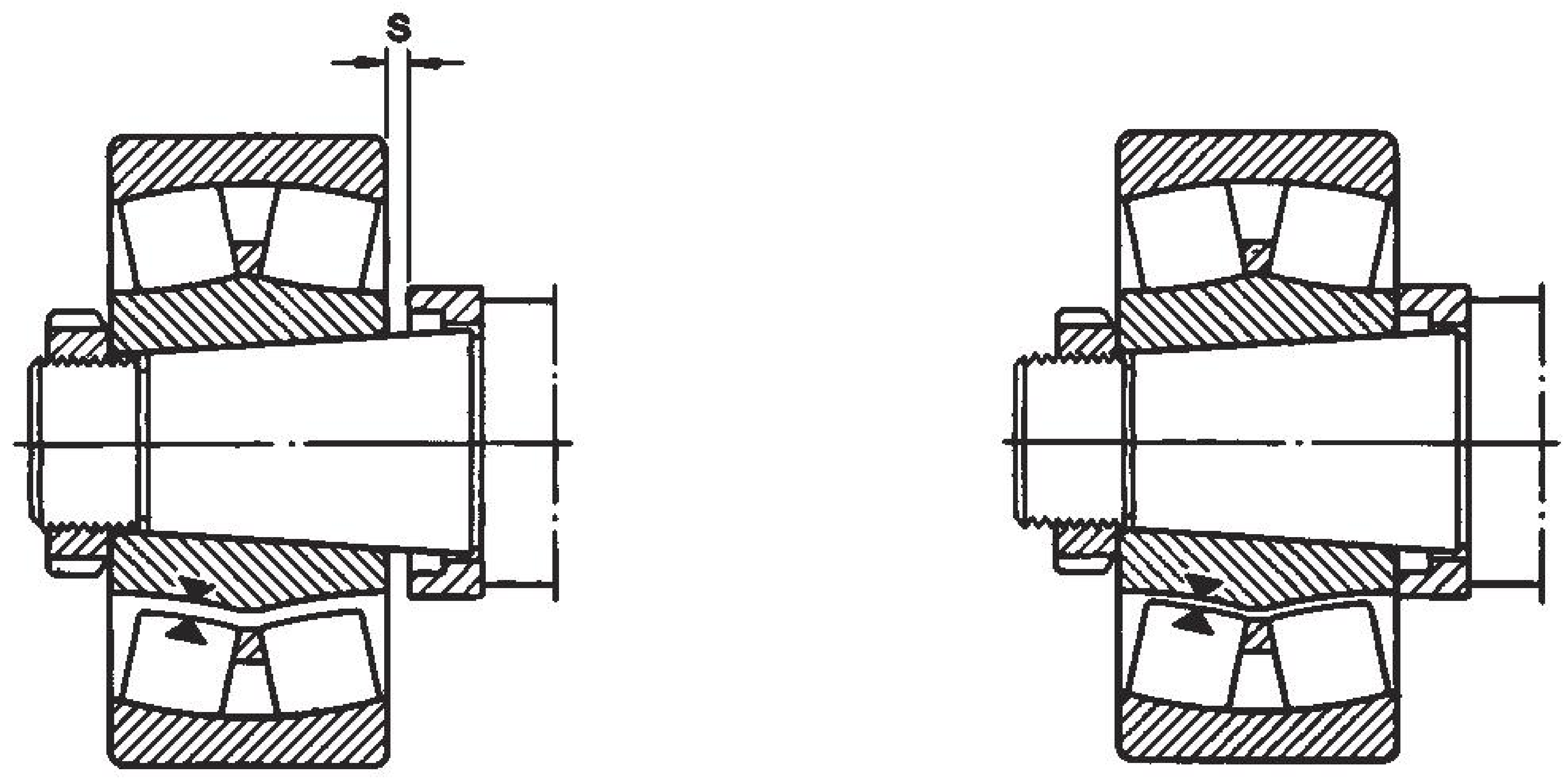
With the blade between the uppermost roller and the sphere, attempt to swivel the blade and withdraw it axially. The swiveling motion helps to center the roller in its proper operating position, and withdrawing it with the characteristic wiping feel of a line-to-line contact will show that thickness to be the looseness over that roller. If the blade becomes much looser during the swiveling and withdrawing process, attempt the same procedure with a blade 0.001 in. thicker and continue until a blade cannot be swiveled or withdrawn. The internal clearance over that roller will be the blade that can be swiveled and withdrawn after a thicker one has jammed.

Repeat this procedure in two or three other locations by resting the bearing on a different spot on its outside diameter and measuring over different rollers in one row. Either repeat the above procedure for the other row of rollers or measure each row alternatively in the procedure described above. Make a note of this unmounted internal clearance.

After the unmounted radial clearance is measured, the bearing is placed on its tapered seat. If the shaft provides for a locknut, it is then assembled, but the lock washer is left off the shaft at this point. The locknut should then be tightened against the bearing, pushing it up the taper until the internal clearance is reduced by the specified amount, as shown in Table 2.1. An impact-type spanner wrench as shown in Fig. 2.29 is ideal for tightening the nut.

Table 2.1. Recommendation for Driving a Spherical Roller Bearing on a Tapered Seat

Mounting of spherical roller bearings with tapered bore



2 220	2 250	0,000	0,000	0,0	0,0	00,0	00,0	0,000	0,000	
Bearing bore diameter d		Reduction in radial internal clearance		Axial drive-up s[1] Taper 1:12 on diameter		Taper 1:30 on diameter		Minimum permissible residual clearance[2] after mounting bearings with initial clearance		
over	incl.	min	max	min	max	min	max	Normal	C3	C4
mm		mm		mm				mm		
24	**30**	0.015	0.020	0.3	0.35	–	–	0.015	0.020	0.035
30	**40**	0.020	0.025	0.35	0.4	–	–	0.015	0.025	0.040
40	**50**	0.025	0.030	0.4	0.45	–	–	0.020	0.030	0.050
50	**65**	0.030	0.040	0.45	0.6	–	–	0.025	0.035	0.055
65	**80**	0.040	0.050	0.6	0.75	–	–	0.025	0.040	0.070
80	**100**	0.045	0.060	0.7	0.9	1.7	2.2	0.035	0.050	0.080
100	**120**	0.050	0.070	0.75	1.1	1.9	2.7	0.050	0.065	0.100
120	**140**	0.065	0.090	1.1	1.4	2.7	3.5	0.055	0.080	0.110
140	**160**	0.075	0.100	1.2	1.6	3.0	4.0	0.055	0.090	0.130
160	**180**	0.080	0.110	1.3	1.7	3.2	4.2	0.060	0.100	0.150
180	**200**	0.090	0.130	1.4	2.0	3.5	5.0	0.070	0.100	0.160
200	**225**	0.100	0.140	1.6	2.2	4.0	5.5	0.080	0.120	0.180
225	**250**	0.110	0.150	1.7	2.4	4.2	6.0	0.090	0.130	0.200
250	**280**	0.120	0.170	1.9	2.7	4.7	6.7	0.100	0.140	0.220
280	**315**	0.130	0.190	2.0	3.0	5.0	7.5	0.110	0.150	0.240
315	**355**	0.150	0.210	2.4	3.3	6.0	8.2	0.120	0.170	0.260
355	**400**	0.170	0.230	2.6	3.6	6.5	9.0	0.130	0.190	0.290
400	**450**	0.200	0.260	3.1	4.0	7.7	10	0.130	0.200	0.310
450	**500**	0.210	0.280	3.3	4.4	8.2	11	0.160	0.230	0.350
500	**560**	0.240	0.320	3.7	5.0	9.2	12.5	0.170	0.250	0.360
560	**630**	0.260	0.350	4.0	5.4	10	13.5	0.200	0.290	0.410
630	**710**	0.300	0.400	4.6	6.2	11.5	15.5	0.210	0.310	0.450
710	**800**	0.340	0.450	5.3	7.0	13.3	17.5	0.230	0.350	0.510
800	**900**	0.370	0.500	5.7	7.8	14.3	19.5	0.270	0.390	0.570
900	**1 000**	0.410	0.550	6.3	8.5	15.8	21	0.300	0.430	0.640
1 000	**1 120**	0.450	0.600	6.8	9.0	17	23	0.320	0.480	0.700
1 120	**1 250**	0.490	0.650	7.4	9.8	18.5	25	0.340	0.540	0.770

[1]Valid for solid steel shafts only. Larger axial displacements are necessary for hollow shafts depending on wall thickness.

[2]The residual clearance must be checked in cases where the initial radial internal clearance is in the lower half of the tolerance range and where large temperature differentials between the bearing range can arise in operation. The residual clearance must not be less than the minimum values quoted above.

The amount of force required to drive a tapered-bore bearing can be greatly reduced if the shaft is drilled and grooved as shown in Fig. 2.30. If these fittings are available, attach a hydraulic pump to the connection at the end of the shaft. Drive the bearing on the taper just enough so there is some interference; then build up hydraulic pressure under the bore of the bearing. A pressure of 3000 to

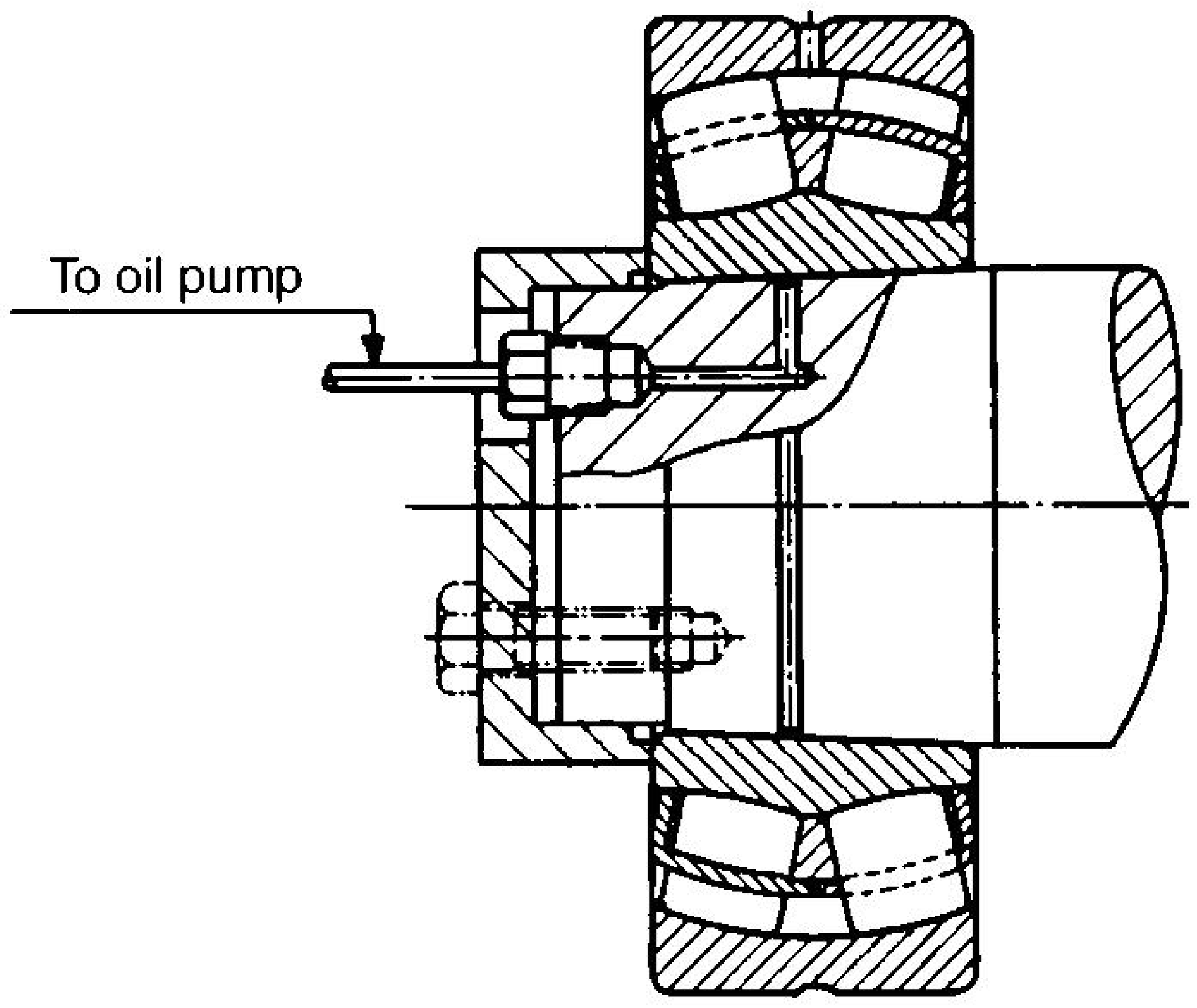


FIGURE 2.30 Drilling and grooving of the shaft to reduce bearing driving force.

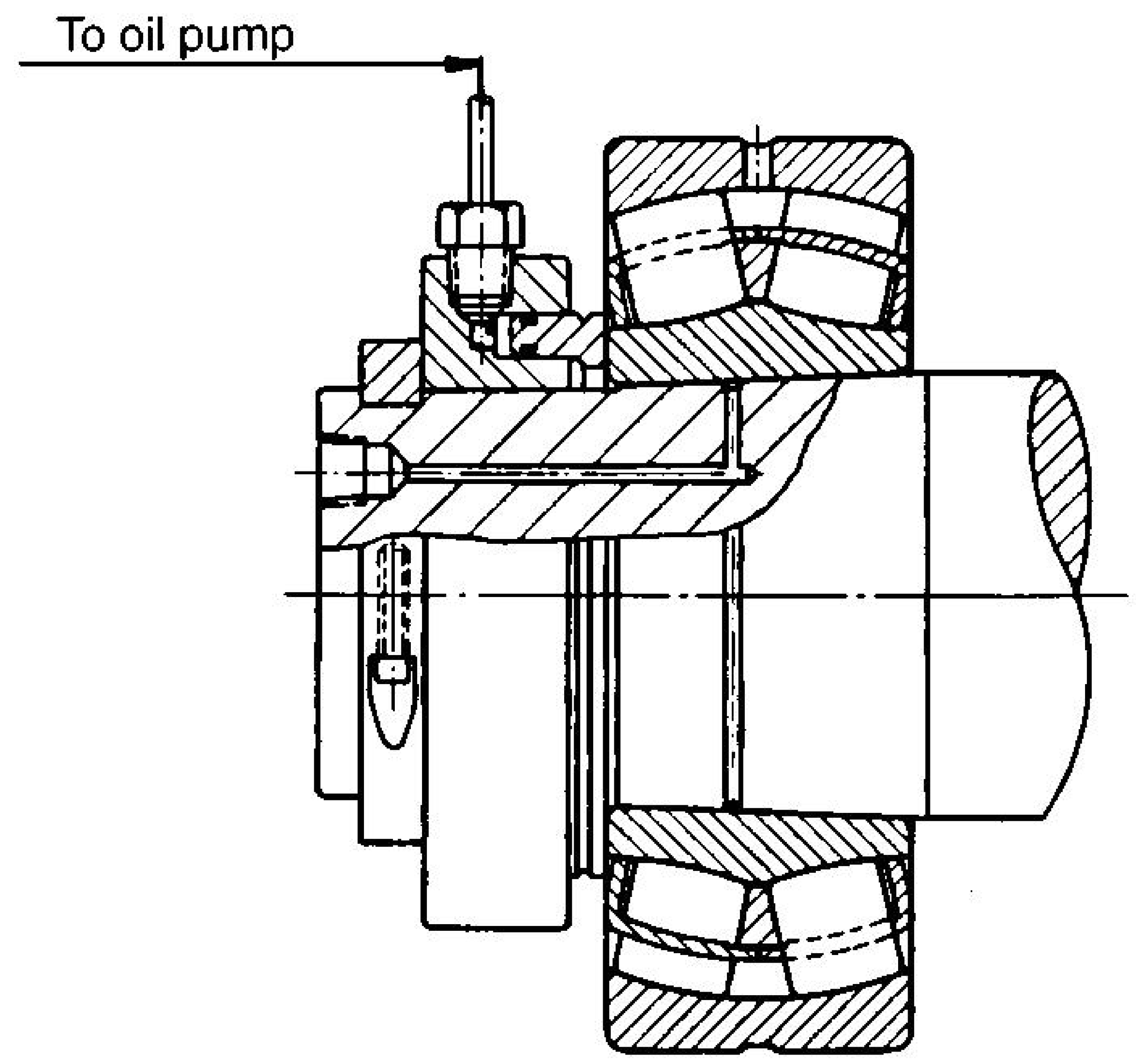


FIGURE 2.31 Use of a hydraulic nut to mount a spherical roller bearing.

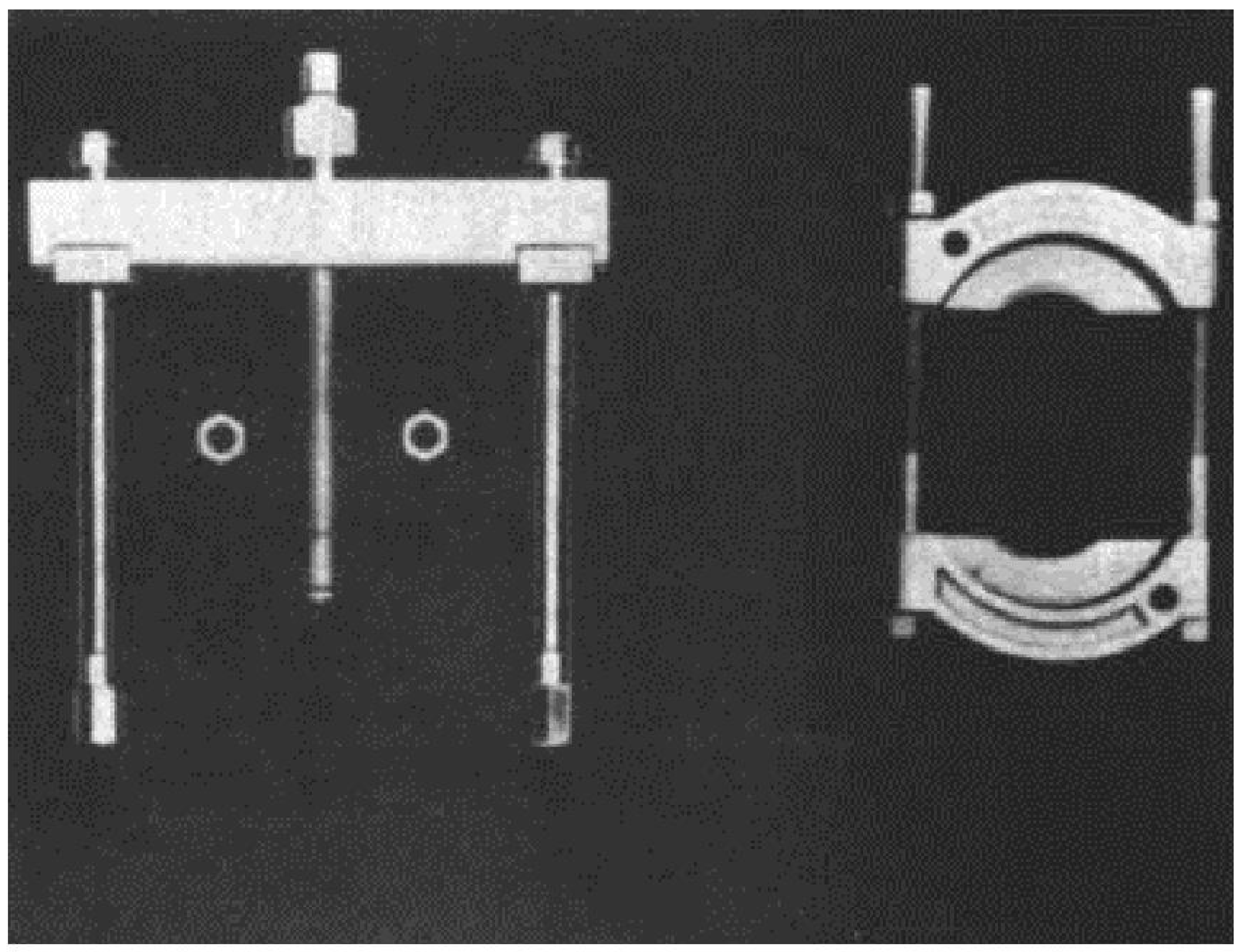


FIGURE 2.32 Typical bearing pullers.

6000 psi will be needed, but with this pressure between the bore of the bearing and the shaft it is possible to float the bearing up the taper with much less torque applied to the locknut or clamp plate than in a dry mounting.

Another convenient way to mount a tapered-bore bearing is to use a hydraulic nut or mounting tool, as shown in Fig. 2.31. This technique also can be adapted to sleeve mountings that are large enough to be drilled and grooved.

Cylindrical and tapered roller bearings with tapered bores are not as common as their spherical counterparts, and the manufacturer will have specific mounting instructions for each application.

Dismounting of Bearings. A wide variety of tools are available commercially which are designed to remove a rolling bearing from its seat without damage. Typical bearing pullers are shown in Fig. 2.32. In removal, we should again keep in mind the basic rule to apply force to the ring with the tight fit. Pullers normally can be applied to bearings so that this rule is observed. However, sometimes supplementary plates or fixtures may be required.

For smaller bearings, an arbor press is equally effective at removing as well as mounting bearings. Also, techniques such as the one shown in Fig. 2.33 may be used where size permits.

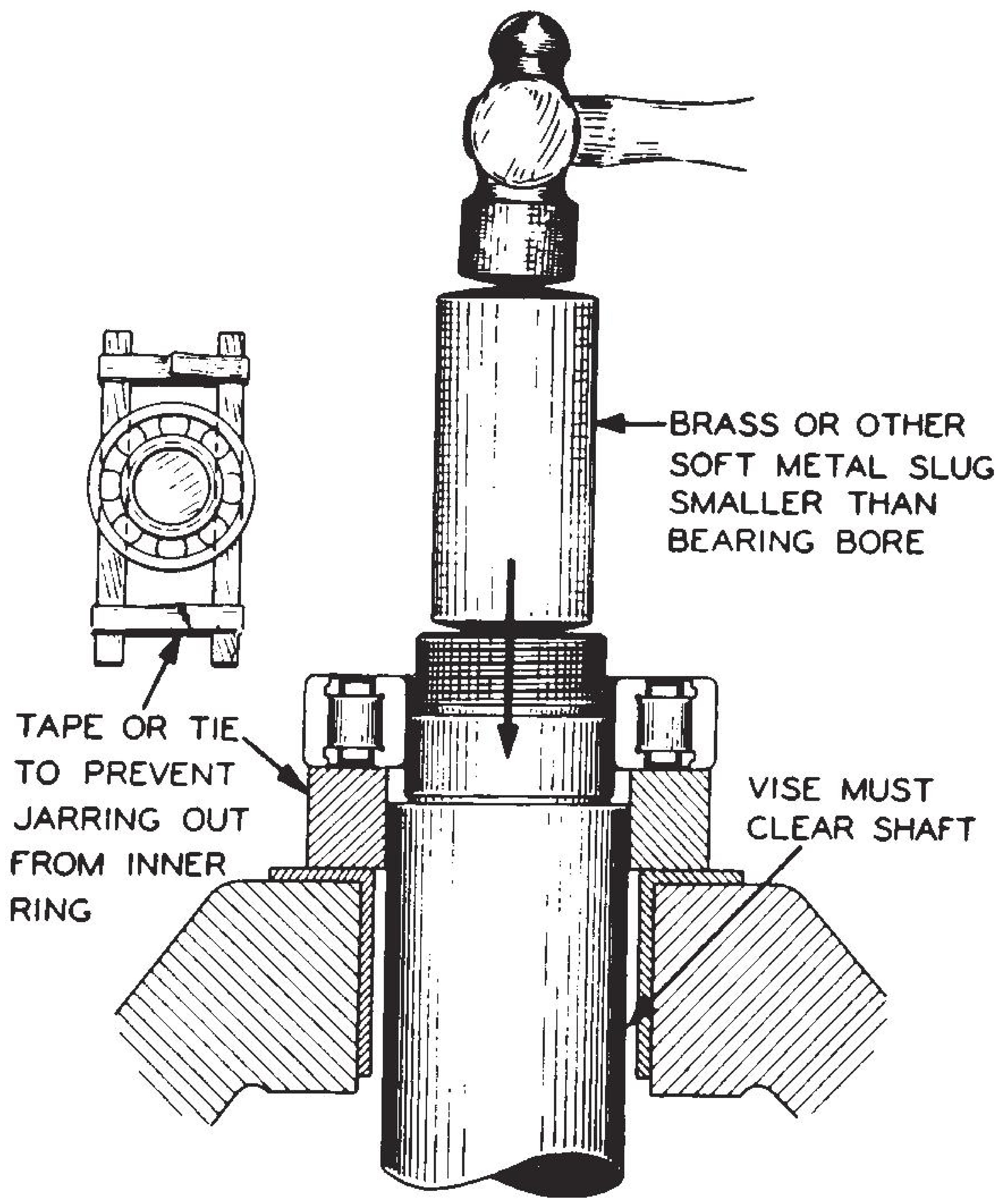


FIGURE 2.33 Method to remove small bearings by driving shaft through supported bearing.

Hydraulic Removal. Where shafts have been designed to apply hydraulic pressure to the fit between shaft and bearing, removal is quite simple. First, the locking device, whatever it is, should be backed off a distance greater than the axial movement of the mounting; ¼ in. will be sufficient in virtually every case. Then connect a hydraulic pump to the fitting provided at the end of the shaft, as shown in Fig. 2.34, and start building up pressure. When pressure becomes great enough to break the fit, usually about 3000 to 6000 psi, the bearing will literally jump off the taper with a sharp bang. The retaining device, still being loosely connected, will prevent the bearing from coming off the end of the shaft. Never completely remove the retaining device.

Hydraulic pressure may be used with straight-bore bearings, but a puller must be used in conjunction with the hydraulic pump, since there will be no axial component of the hydraulic pressure to blow the bearing off its seat. See Fig. 2.34.

Larger sleeve mountings also may be designed to utilize hydraulic pressure for dismounting. If this feature is available, follow the same procedure as outlined above. However, if the sleeve mounting does not have this feature, other techniques such as shown in Fig. 2.35, must be used. For withdrawal sleeves, a special nut must be used, as shown in Fig. 2.36. For large sleeves, a hydraulic nut is desirable for dismounting.

LUBRICATION

The primary purpose of lubrication in a rolling bearing is to separate the contacting surfaces, both rolling and sliding. This purpose is rarely achieved, and boundary lubrication or partial metal-to-metal contact frequently occurs. By far the most common lubricants are petroleum products in the form of grease or liquid oil. Synthetics are, however, finding more use in high-temperature applications.

Table 2.1.
Recommendation for Driving a Spherical Roller Bearing on a Tapered Seat

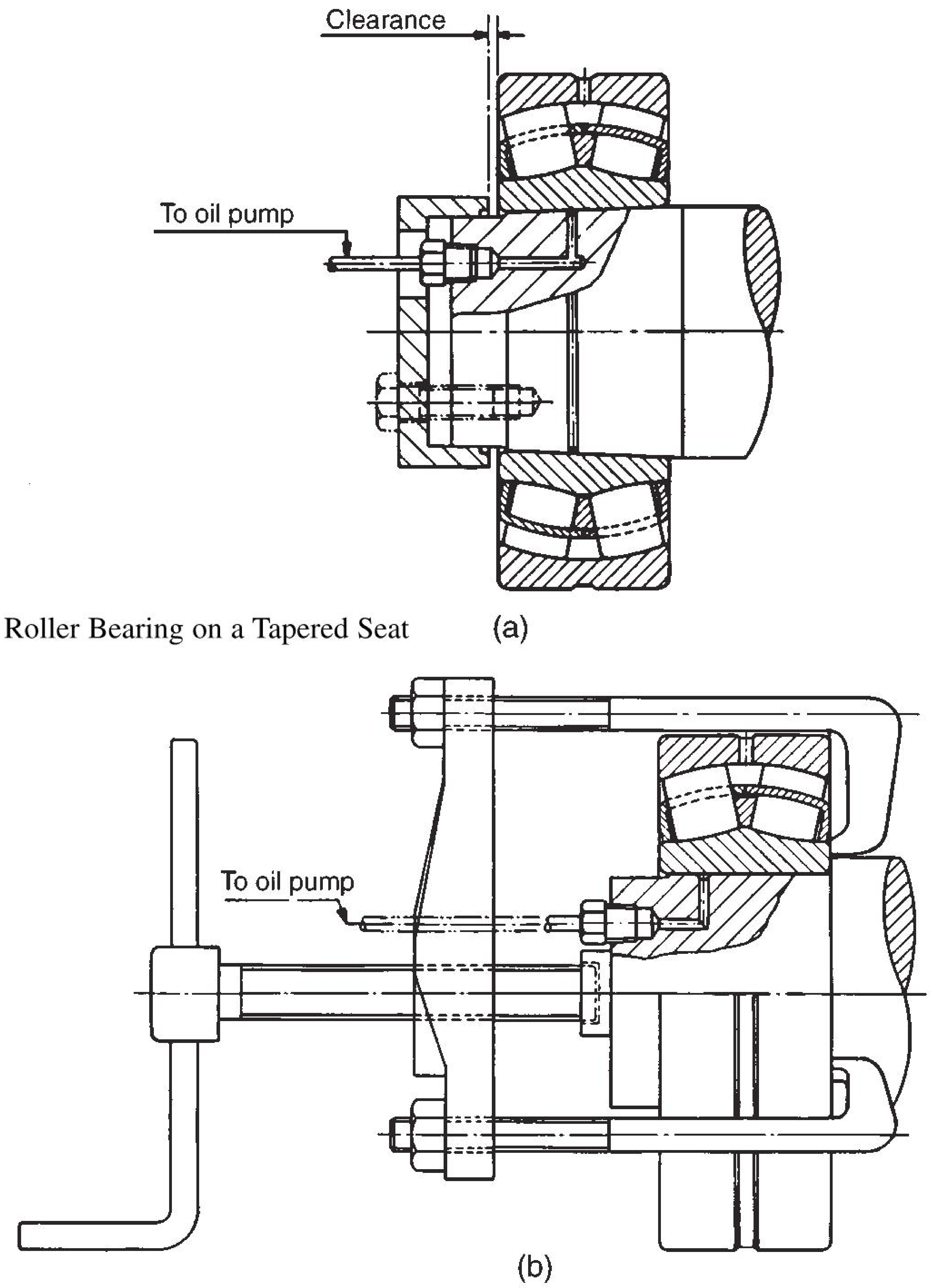


FIGURE 2.34 Hydraulic removal. (*a*) By connection to pump. (*b*) In conjunction with a puller.

Generally, the machine builder decides whether a bearing will be a grease- or oil-lubricated component and normally will recommend the basic specifications of the required lubricant. However, because the machine designer cannot foresee all the variable conditions under which the equipment will operate, some judgment is required on the part of maintenance personnel. Some knowledge of lubricants is therefore useful.

Oil Lubrication. For oil lubrication, the Annular Bearing Engineers Committee (ABEC) has issued the following recommendations:

> The friction torque in a ball bearing lubricated with oil consists essentially of two components. One of these is a function of the bearing design and the load imposed on the bearing, and the other is a function of the viscosity and quantity of the oil and the speed of the bearing.
>
> It has been found that the friction torque in a bearing is lowest with a very small quantity of oil, just sufficient to form a thin film over the contacting surfaces, and that the friction will increase with greater

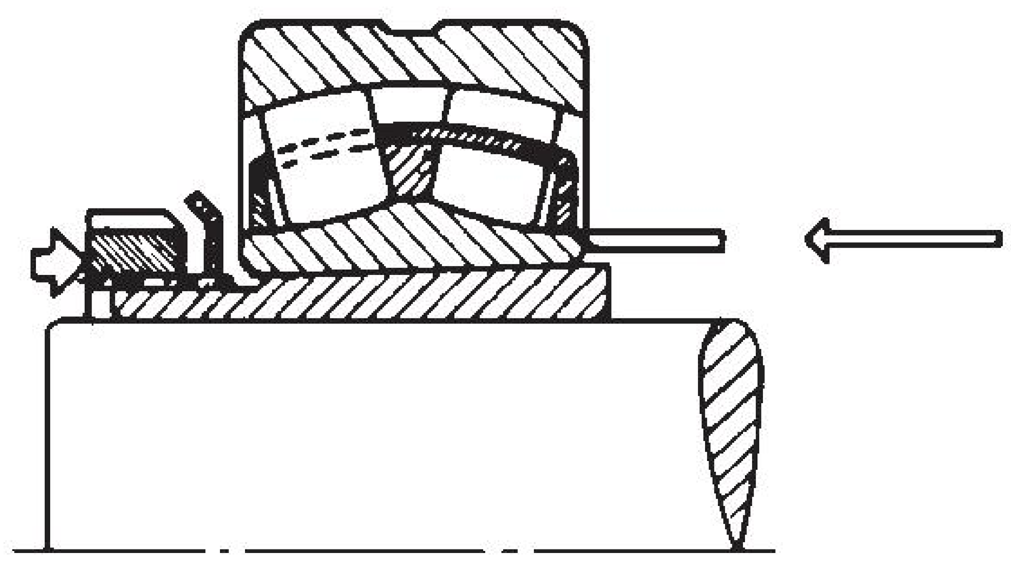


FIGURE 2.35 Bearing removal by driving bearing down tapered sleeve.

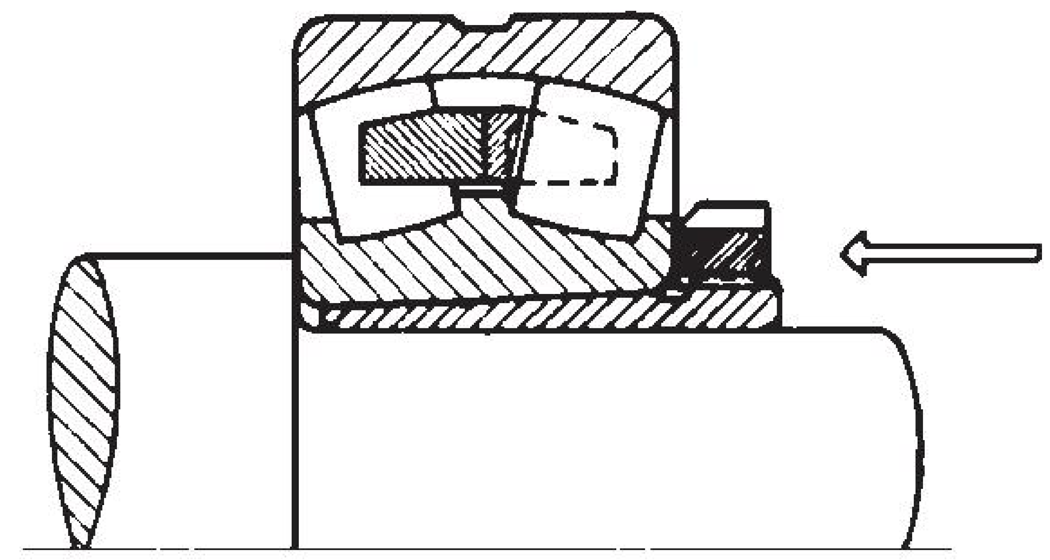


FIGURE 2.36 Bearing removal by pulling sleeve from under bearing.

quantity and with higher viscosity of the oil. With more oil than just enough to make a film, the friction torque will also increase with the speed.

The energy loss in a bearing is proportional to the product of torque and speed, and this energy loss will be dissipated as heat and cause a rise in the temperature of the bearing and its housing. This temperature rise will be checked by radiation, convection, and conduction of the heat generated to an extent depending upon the construction of the housing and the influence of the surrounding atmosphere. The rise in temperature, due to operation of the bearing, will result in a decrease in viscosity of the oil, and therefore, a decrease in friction torque compared with the friction of starting, but soon a balanced condition will be reached.

With so many factors influencing the friction torque, energy loss, and temperature rise in a bearing lubricated with oil, it is evidently not possible to give definite recommendations for selection of oil for all bearing applications, but two general considerations are dominant:

1. The desire to reduce friction to a minimum, which requires a small quantity of oil of low viscosity.
2. The desire to maintain lubrication safely without much regard for friction losses, which results in using larger quantities of oil and usually of somewhat greater viscosity in order to reduce losses from evaporation or leakage.

This second condition is most frequently met when bearings have to operate in a wide range of temperatures. An oil that has the least changes with changes in temperature, i.e., an oil with high viscosity index, should be selected.

In the great majority of applications, pure mineral oils are most satisfactory, but they should, of course, be free from contamination that may cause wear in the bearing, and they should show high resistance to oxidation, gumming, and deterioration by evaporation of light distillates, and they must not cause corrosion of any parts of the bearing during standing or operation.

It is self-evident that for very low starting temperatures an oil must be selected that has sufficiently low pour-point so that the bearing will not be locked by oil frozen solid.

In special applications, various compounded oils may be preferred, and in such cases, the recommendation of the lubricant manufacturer should be obtained.

Grease Lubrication. Where grease lubrication is used, we need to consider a few of the basic physical and chemical characteristics of the lubricant. Greases are a mixture of lubricating oil and usually a soap base. The base merely acts to keep the oil in suspension. When moving parts of a bearing come in contact with the grease, a small quantity of oil will adhere to the bearing surfaces. Oil is therefore removed from the grease near the rotating parts. Bleeding of the oil from the grease obviously cannot go on indefinitely, so new grease must come in contact with the moving part or a lubrication failure will result.

Many maintenance departments want to use one grease to lubricate all bearings in the plant. Some lubricant suppliers even advocate this technique. However, it is a risky procedure at best, since there is no true universal ball and roller bearing grease. A ball bearing is best lubricated with a fairly stiff grease which will channel. On the National Lubricating Grease Institute (NLGI) code, greases of the number 2 consistency, or 265 to 295 worked penetration, are normally recommended. For roller bearings, a grease stiff enough to channel is not desirable, since the full width of the roller track would soon be starved for lubricant if the grease is not soft enough to slump back into the bearing when it is pushed aside. This generally means greases in the number 0 or 1 consistency class with worked-penetration numbers of 355 to 380 for grade 0 and 310 to 340 for a number 1 grease.

Whatever the consistency of the grease, it is still the properties of the oil compounded in the grease that determine if the bearing will be satisfactorily lubricated. All statements and guidelines outlined above in the discussion of oil lubrication also apply to grease-lubricated bearings.

Another characteristic of a grease that must be considered is its *drop point.* This is the temperature at which the grease passes from a semisolid to a liquid. Typical dropping points are as follows:

Calcium	$+14 \pm 140°F$
Sodium	$-22 \pm 176°F$
Lithium	$-22 \pm 230°F$
Bentone	$-22 \pm 266°F$
Silicone	$-22 \pm 266°F$
Calcium complex	$-4 \pm 266°F$
Aluminum complex	$-22 \pm 230°F$

The drop point is the characteristic referred to when a grease is advertised as being good up to 400°F. Whether it will lubricate a bearing or not is still a function of the viscosity of the lubricating oil, not of the drop point of the base. In fact, common industrial bearings made of standard through-hardened or case-hardened materials have temperature limitations of 200 to 300°F depending on the material and how it was heat-treated. The bearing manufacturer should be consulted for specific information.

Never mix greases that are incompatible. If two such greases are mixed, the resulting mixture usually has a softer consistency which will eventually cause failure through leakage. If you don't know what type of grease a bearing was lubricated with originally, do not regrease without first removing the old grease both from the bearing and the surrounding environment.

Generally, bearings are not lubricated until after mounting. The most important reason for this is cleanness. The later grease is applied, the greater are the chances of avoiding contamination. The bearing should be lubricated prior to mounting only when pregreasing is the only way to obtain an even distribution of grease.

The right quantity of grease is as important as the right type of grease. Follow these general rules for quantity:

A bearing should be filled completely with grease, but free space in the housing should only be partially filled (between 30 and 50 percent). However, in nonvibrating applications, many lithium soap greases, also called *total-fill greases,* can fill up to 90 percent of the free space in the housing without any risk of a rise in temperature. Thus impurities can be prevented from entering the bearing, and relubrication intervals can be extended. Bearings that have to operate at high speeds, e.g., machine tool spindles, where it is desirable to keep the temperature low, should be lubricated with small quantities of grease.

In vibrating applications, such as wheel hubs, vehicle axle boxes, and vibrators, grease fill should be no more than 60 percent of the housing.

When relubrication intervals are long, then the housings should be easy to open. If more frequent lubrication is required, the housing should be fitted with some kind of grease-filling device, preferably a lubrication duct with a nipple.

In the optimal situation, grease can be injected with a grease gun. Some bearings are provided with grooves and ducts for relubrication; others have to be relubricated from the side.

Only the grease in the bearing should be replaced. The amount of grease, therefore, depends on the bearing size. If relubrication instructions are available from the original manufacturer, follow them. If not, or if you suspect the lubrication amount is inadequate, use the following formula to determine the correct amount.

$$Gq = 0.114 \times D \times B \qquad \text{(in ounces)}$$

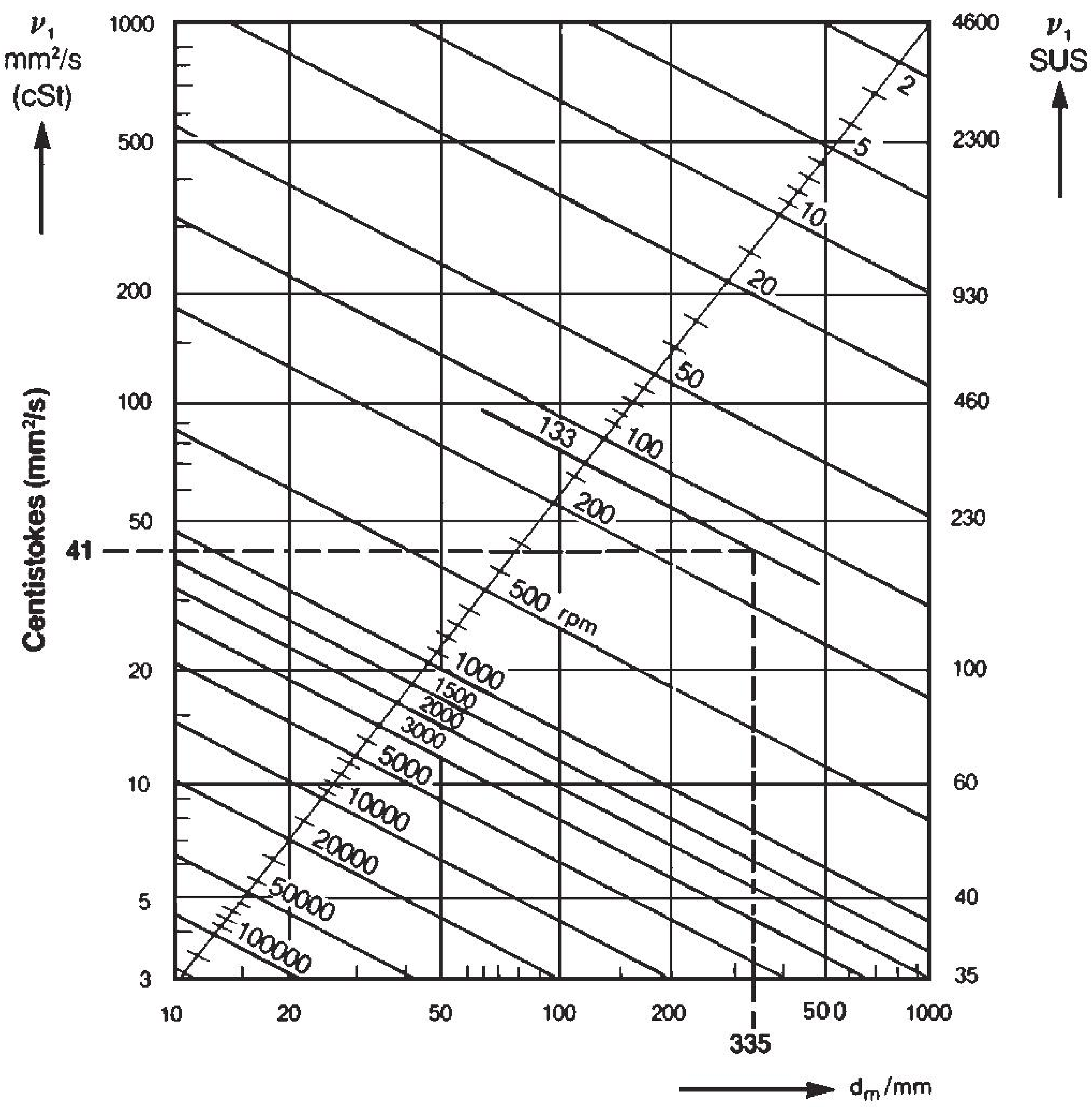
where G_q = grease quantity in ounces
D = bearing outside diameter in inches
B = total bearing width in inches

Selection of Lubricant. Research in elastohydrodynamics (EHD) has contributed greatly to the knowledge of lubricants, rolling bearings, and how they work. Results of this work have been published in various forms that may be used as a guide in the selection of the correct lubricant. Figure 2.37 is an example of these data in graph form which plots required viscosity of the lubricant in centistokes at operating temperature as a function of bearing size and speed. The abscissa of the curve is the bearing size, expressed as mean diameter in millimeters; the diagonal lines are the speed in rpm; and the ordinate is the required viscosity at the operating temperature of the bearing. It is obvious from an examination of this chart that the larger the bearing and the slower the speed, the higher is the required viscosity. This characteristic of the EHD theory sometimes produces a paradox where the lubricant required may not be realistically used for other reasons. Lubrication experts should be consulted when this situation exists.

When grease lubrication is used, the values obtained from the chart apply to the base oil of the grease. The example shown on the graph indicates that a rolling bearing with a mean diameter of 335 mm, running at 133 rpm, will require a lubricant having 41 centistokes at its operating temperature.

Reasonable estimates of bearing operating temperature can be made ahead of time based on analytical calculations or experience. In most cases, a combination of the two gives the most realistic estimate. The machine builder and/or the bearing supplier can help in this area and will normally make lubrication recommendations for new equipment. Some adjustment may have to be made to the original selections after operating experience is obtained.

In current literature and specification sheets, lubricants are normally grouped into viscosity grades according to standards established by the International Standards Organization (ISO). Lubricants are rated from ISO 2 to ISO 1500, the numbers indicating the mean value of a specified



d_m = (bearing bore + bearing O.D.) ÷ 2
ν_1 = required lubricant viscosity for adequate lubrication at the operating temperature

FIGURE 2.37 Minimum required lubricant viscosity for adequate lubricant film.

range of viscosity at a temperature of 104°F (40°C). Figure 2.38 plots these lubricants on a temperature-viscosity diagram.

In the selection of a lubricant, it should be kept in mind that the oil temperature in the bearing may vary 5 to 10°F (3 to 6°C) from the temperature of the housing. There are no exact rules for this estimate, but it is wise to add these degrees if housing temperature is used as a criteria for determining operating level of the bearing.

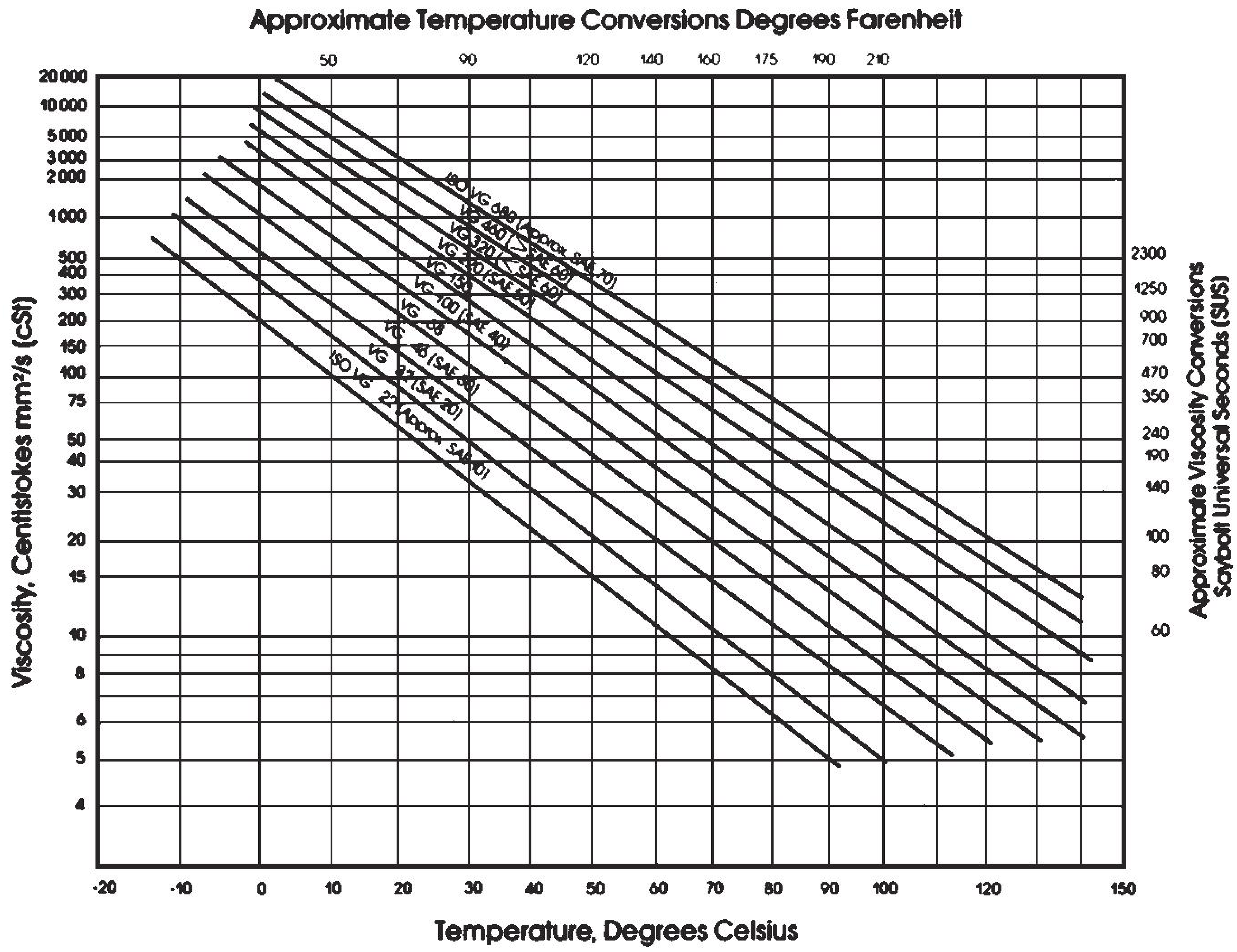


FIGURE 2.38 Approximate temperature conversions. Viscosity classification numbers are according to International Standard ISO 3448-1975 for oils having a viscosity index of 95. Approximate equivalent SAE viscosity grades are shown in parentheses.

CHAPTER 3
FLEXIBLE COUPLINGS FOR POWER TRANSMISSION

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GENERAL

A *flexible coupling* is a mechanical device used to connect two axially oriented shafts. Its purpose is to transmit torque or rotary motion without slip and at the same time compensate for angular, parallel, and axial misalignment. There are many supplementary functions, which include providing for or restricting axial movement of the connected shafts; minimizing or eliminating the conduction of heat, electricity, or sound; torsional dampening; and torsional tuning of a system. Basically, all flexible couplings can be categorized as either mechanical flexing or material flexing. While most available flexible couplings fall strictly into one or the other of these basic categories, a few combine both principles.

The mechanical-flexing group provides flexibility by allowing the components to slide or move relative to each other. Clearances are provided to permit movements to within specific limits. Lubrication is usually required to reduce wear within the coupling and to minimize the cross-loading in the connected shafts. The most prominent in this category are the chain, gear, grid, and Oldham flexible couplings.

The material-flexing group provides flexibility by having certain parts designed to flex. These flexing elements can be of various materials, such as metal, rubber, plastic, or composite. Couplings of this type generally must be operated within the fatigue limits of the material of the flexing element. Most metals have a predictable fatigue limit and permit the establishment of definite boundaries of operation. Elastomers (rubber, plastic, etc.) usually do not have well-defined fatigue limits, and service life is determined primarily by the operational conditions. The material-flexing group includes laminated-disk, diaphragm, spring, and elastomer flexible couplings.

DESCRIPTION OF TYPES

A wide variety of concepts is available within each of the two basic groups. A detailed account of each would be impractical. Only the most well-established forms will be described and discussed in generalities.

Chain couplings (Fig. 3.1) are compact units capable of transmitting proportionately high torques at low speeds.They consist of two hubs having sprocket teeth which are connected by a strand of

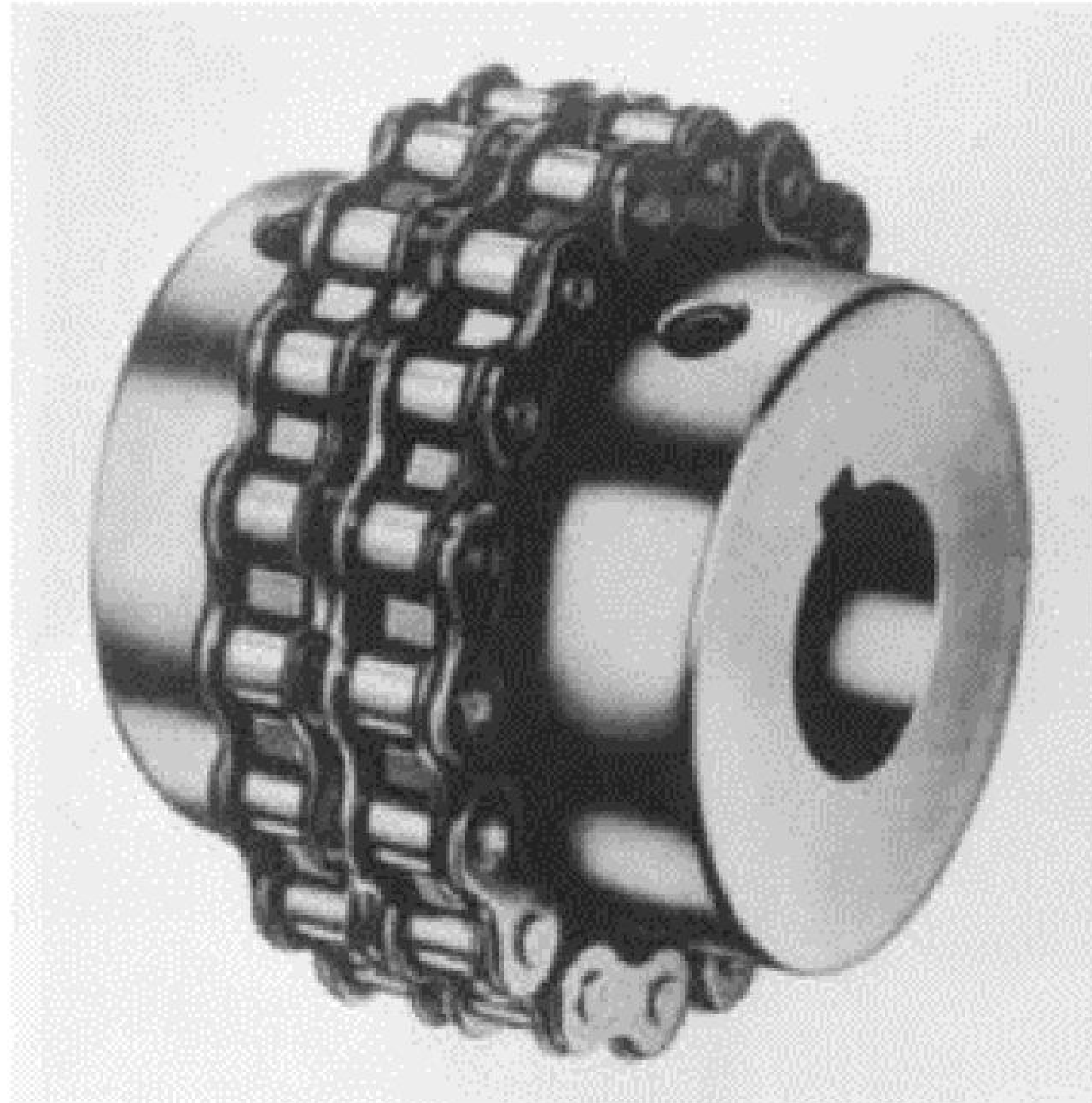


FIGURE 3.1 Roller-chain coupling. (*Rexnord Corporation, Coupling Division*)

single-roller, double-roller, or silent chain. Shaft misalignment is accommodated by clearances between the chain and the sprocket teeth and/or clearances within the chain itself. A number of special features such as hardened sprocket teeth, special tooth forms, and barrel-shaped rollers are available which are designed to increase flexibility and reduce wear. Nonmetallic chains are used on light-duty drives where the use of a lubricant is prohibited.

Coupling covers are recommended for all drives where the rotating speed is capable of slinging the lubricant or where the atmosphere is wet, corrosive, or abrasive. They protect the coupling and greatly extend its life by retaining the lubricant and preventing dirt or other foreign materials from coming in contact with or between the sliding parts. Most covers rotate with the coupling. Grease holes permit lubrication without disturbing the gaskets or seals. Covers should be half-filled with light grease (no. 1 is normally recommended). A heavier grease should be used if the coupling is operated in a high-temperature environment. Routine flushing and relubricating are required. It is generally recommended that a roller-chain coupling be relubricated every 6 months or sooner depending on the conditions of operation. Stationary oil-bath covers are available for large, slow-speed chain couplings.

Where a cover is not required, the chain and sprockets should be coated thoroughly with a good-quality bearing grease. The use of a stiff brush is suggested because it will give better penetration of the grease into critical areas of the chain. This is generally required on a weekly basis.

To obtain maximum service with roller-chain couplings, misalignment of the connected shafts must be restricted to the manufacturer's recommendations. Excessive amounts can cause rapid wear of the chain and sprockets as well as early failure of the cover seals and a resultant loss of lubricant. A straightedge and caliper can usually be used to check angular and parallel misalignment as well as the shaft gap. A properly aligned coupling will allow the chain to be wrapped around the sprockets and the connecting pin inserted without any significant force.

Gear couplings (Figs. 3.2 to 3.5), the most prominent type in the mechanical-flexing group, are available in a wide range of sizes and styles. They are capable of transmitting proportionately high torques at either low or high speeds. In their most common form, they are compact and consist of two identical hubs with external gear teeth and a sleeve or sleeves with matching internal gear teeth. Shaft misalignment is accommodated by clearances between the matching gear teeth. Special tooth forms are available which are designed to reduce wear and increase flexibility without increasing clearances. These include crowned tips, curved flanks, and curved roots. The sleeve may be a single tubular piece, or it may consist of two flanged halves bolted rigidly together.

Floating-shaft gear couplings usually consist of a standard coupling with a two-piece sleeve. The sleeve halves are bolted to rigid flanges to form two single-flexing couplings. These are connected

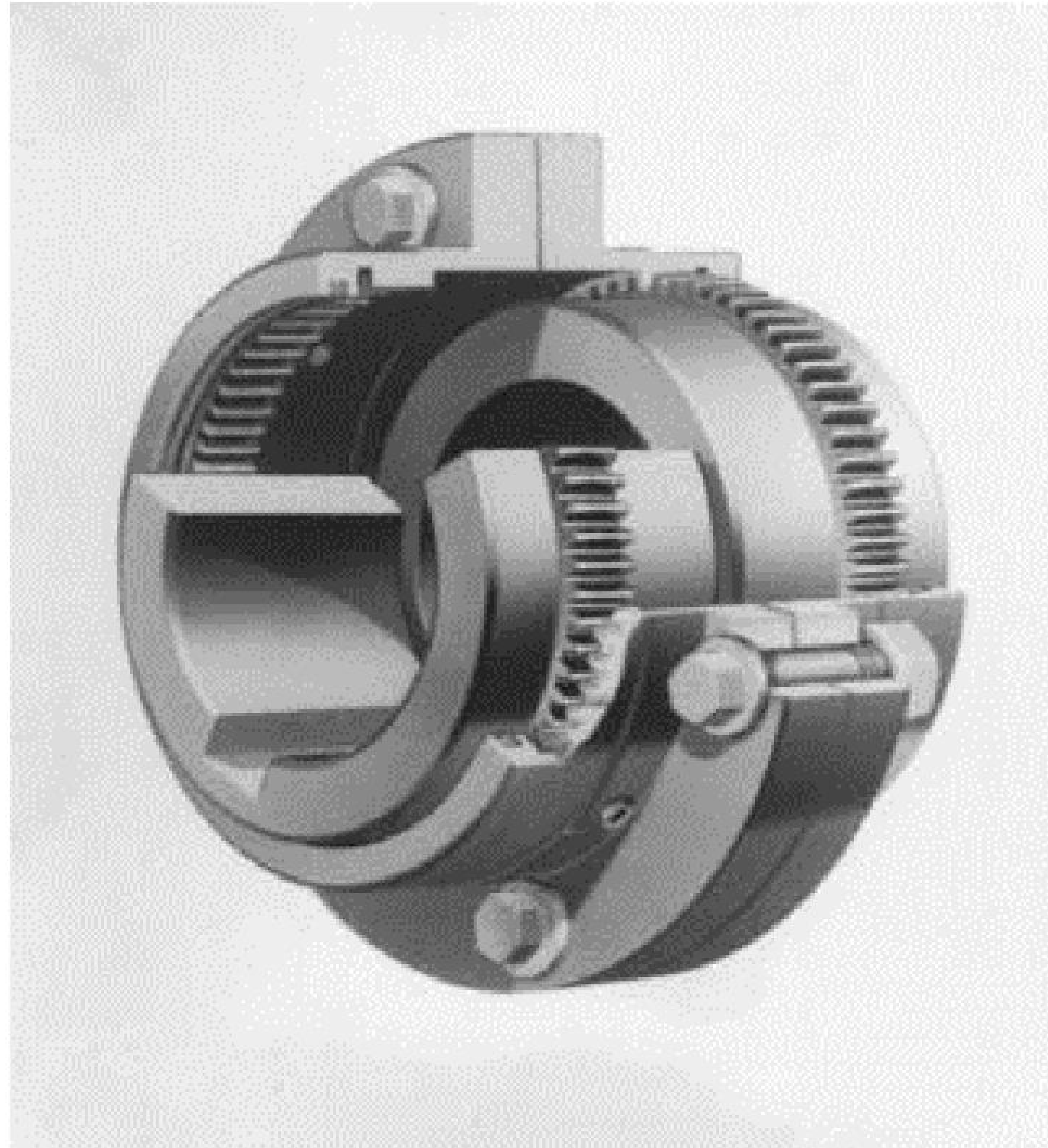


FIGURE 3.2 Gear-tooth coupling, standard double-engagement type. (*The Falk Corporation.*)

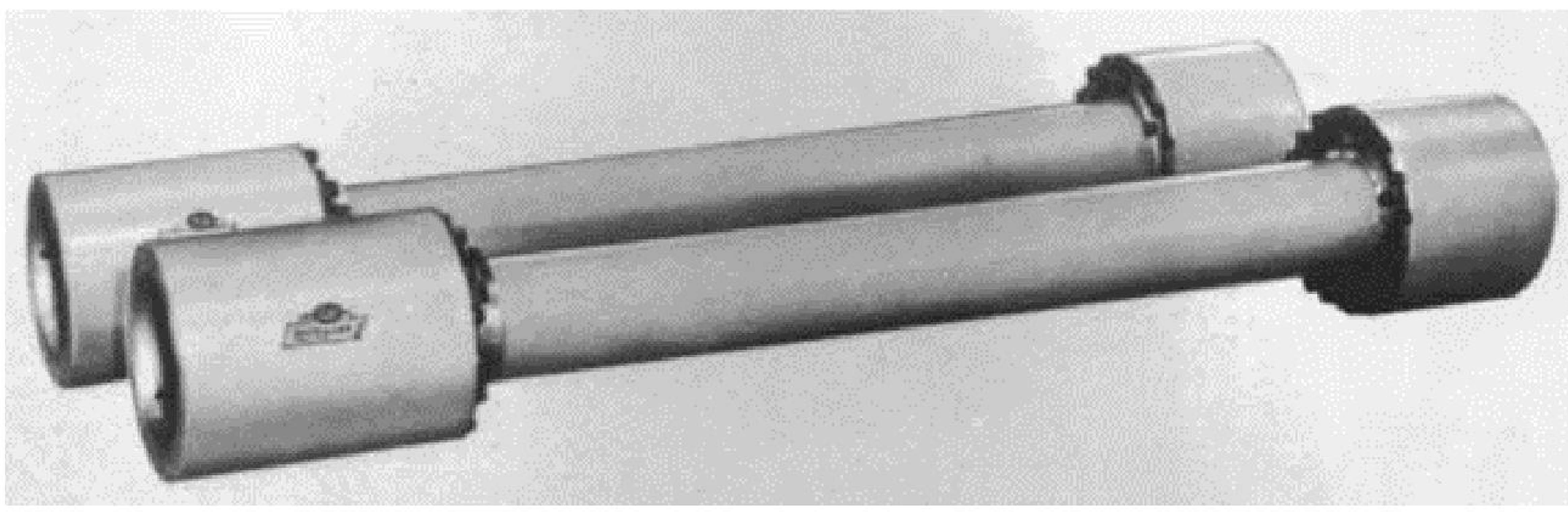


FIGURE 3.3 Gear-tooth coupling, spacer type. (*Zurn Industries, Inc.*)

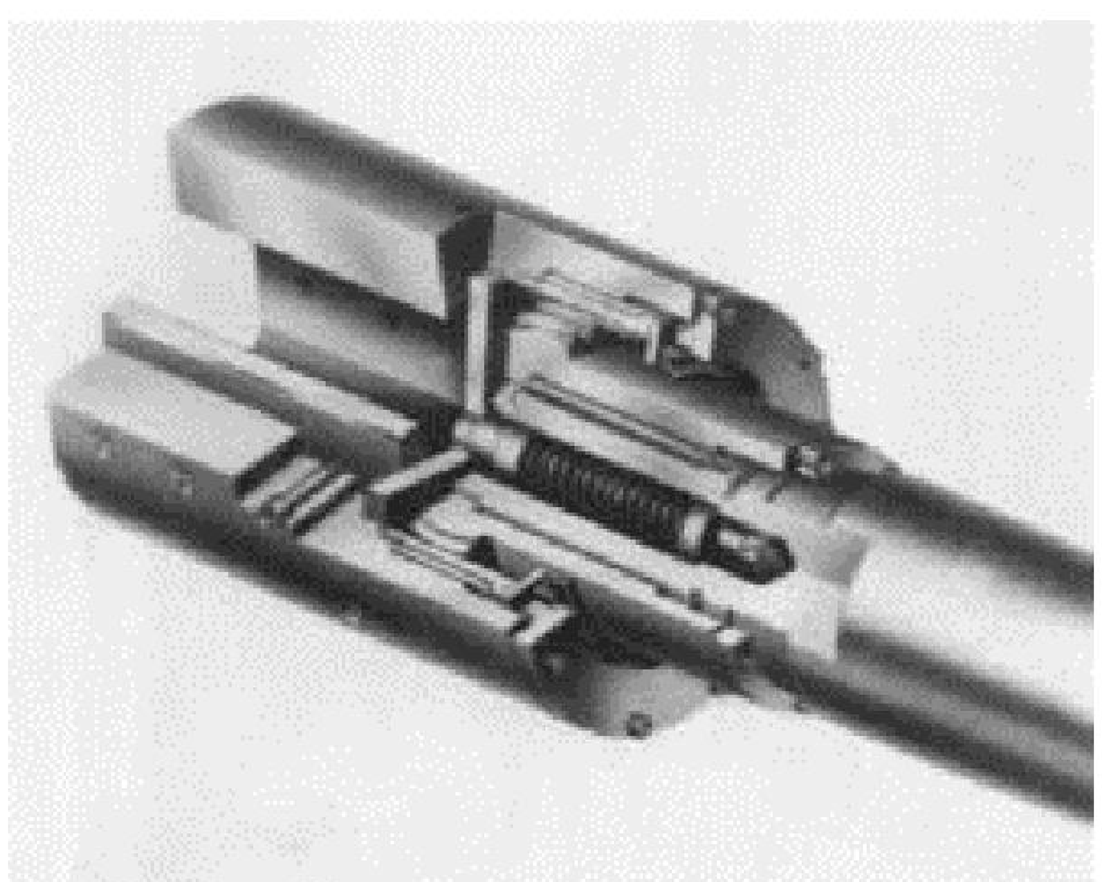


FIGURE 3.4 Gear-tooth coupling, spindle type. (*Zurn Industries, Inc.*)

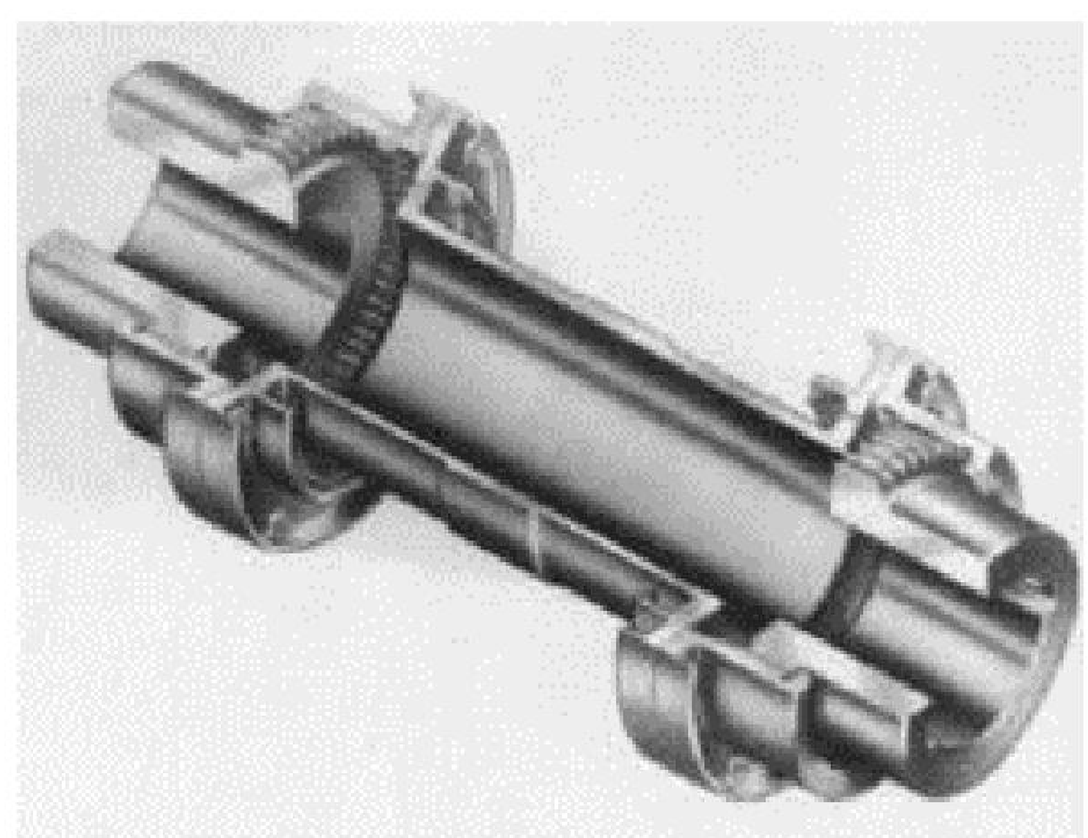


FIGURE 3.5 Gear-tooth coupling, high-speed spacer type. (*Zurn Industries, Inc.*)

by an intermediate shaft which permits the transmission of power between widely separated machines. On high-speed or short-span drives, spools are used to separate the two half couplings.

Spindle couplings are a modification of the floating-shaft gear coupling. They are used extensively on mill-roll drives and other related equipment which has unavoidable offsetting of the driving and driven shafts. In addition to accepting large angles of misalignment, they must operate with a relatively uniform angular velocity. These couplings are subject to severe operating conditions and are therefore a relatively high maintenance item. Numerous special features are available which are designed to reduce maintenance and downtime.

Some light-duty gear couplings have nonmetallic sleeves such as nylon or urethane, which eliminates the need for lubrication. Generally, all other gear couplings require proper lubrication in order to realize their potential life and are designed to have a static supply of lubricant inside. As the coupling is rotated, this lubricant is thrown outward by centrifugal force to engulf the load-carrying teeth. The relative sliding motion between the hubs and the sleeve permits the establishment of a film between the mating gear teeth.

There are several methods of lubricating gear couplings. These are grease pack, oil fill, oil collect, and continuous oil flow. The vast majority of drives operate at 3600 rpm or less and use grease as the lubricant. Both grease and oil are used at speeds of 3600 to 6000 rpm. Oil is normally used as the lubricant in couplings operation over 6000 rpm. Most high-speed couplings use a continuous oil flow to carry away the heat generated within the coupling. Conditions of operation should be reviewed in selecting the method of lubrication. In addition to the available maintenance program, consideration should be given to torsional loads and their characteristics, high or low temperatures, rotating speeds, and environmental conditions.

The grease-packed and oil-filled units have end rings or seals which are used to retain the lubricant and restrict the entry of dust, grit, moisture, or other contaminants. Sleeves are provided with lubrication holes which permit flushing and relubrication without disturbing the sleeve gasket or seals.

The oil-collector couplings have extended lips on the sleeve through which oil can be induced. The proper oil level can be maintained without shutdown.

The continuous-oil-flow couplings also have extended lips on the sleeve through which oil is induced, while discharge holes in the sleeve permit the excess oil to escape. This arrangement requires that an oil-tight enclosure encompasses the coupling and that an oil-circulating and oil-filtering system be provided.

Lubrication within the coupling is susceptible to heat, centrifugal force due to high speeds, water, foreign oil, solids, or other contaminants. In addition to ambient temperatures, heat can be transferred from the connected equipment or generated within the coupling itself. High centrifugal forces can cause a soap separation and oil bleeding. Soap separation results in improper lubrication and sludging, which can cause the coupling to lock up. Oil bleeding can lead to leakage. When low temperatures are encountered, special low-temperature grease may be required.

An adequate supply of the proper type of lubricant is essential to good coupling performance and long life. In view of the many problems that could arise with the use of an improper lubricant, it is necessary that the coupling manufacturer's instructions be adhered to closely.

The retention of the coupling's lubricant is essential. Leakage can result in high costs of lubrication, reduced coupling life, and the creation of hazardous conditions around the connected equipment.

It is equally important to prevent any abrasive materials or contaminants from entering the coupling. Abrasives will cause excessive wear and prevent proper performance. Contaminants such as water, solvents, and foreign oil can cause a deterioration of the coupling's lubricant.

Care should be used during installation of a new coupling or reassembly of an existing unit. Keyways and keys should be coated with a sealing compound that is resistant to the coupling's lubricant. Seals should be checked for pliability and condition. They must be seated properly in the sleeve, and the lip must be in intimate contact with the hub. Sleeve flange gaskets must be whole and in good condition. Sleeve flange faces and rabbet must be clean and free of nicks. This joint should be tightened in accordance with the manufacturer's recommendations. Lubricant plugs must be clean and tight in the sleeve.

Periodic inspection of the coupling is recommended. Whenever possible, inspection and relubrication of the coupling are done on the same schedule as that established for connected machines. If, during inspection, leakage is evident, plugs, gaskets, seals, and keyways should be checked and the situation corrected. The condition of the lubricant also should be noted. If it is abnormal, the coupling should be flushed and refilled as required.

Metallic-grid couplings (Fig. 3.6) are compact units capable of transmitting proportionately high torques at moderate speeds. They consist of two flanged hubs with special grooves or slots cut axially on the outside. The flanges are joined by interlacing a serpentine metallic grid. Flexibility is achieved by sliding movement of the grid in the slots. Flexure of the grid in the curved slots provides some torsional resilience. The grid may be of one piece or may be provided in two or more sections. Grids with tapered cross sections are available from some manufacturers and are designed to ease installation and removal.

Covers are provided to retain the coupling's lubricant and to prevent dust, grit, and other foreign materials from coming in contact with or between the sliding parts. The cover may be split either horizontally or vertically. Grease holes permit lubrication without disturbing the gaskets or seals.

Proper lubrication is essential. The manufacturer's recommendations as to the type of lubricant should be followed. Seals must be in good condition and properly seated. The gasket must be whole and the cover joint tight. Plugs must be tight. Regular intervals of inspection for leakage and condition of lubricant are recommended. If the lubricant is abnormal, the cover should be opened and all parts thoroughly flushed before the new lubricant is added.

The grid members on this type of coupling are replaceable. If they are significantly worn, they should be replaced.

Misalignment of the connected shafts should be kept within the manufacturer's recommendation. Excessive amounts can cause rapid wear of the grid and hub slots as well as early failure of the cover seals. A spacer bar of caliper and straightedge usually can be used to check angular and parallel misalignment as well as shaft gap.

Oldham couplings (Fig 3.7) are also known as *block-and-jaw couplings.* They are compact units normally relegated to light or medium duty and moderate speeds. They consist of two jaw flanges and a floating-block center member. The jaw flanges are positioned at right angles to each other and engage opposite parallel surfaces of the block.

Small-sized light-duty couplings usually have flanges of die-cast metal and blocks of nonmetallic material such as laminated phenolic plastic. These units do not require any lubrication. The

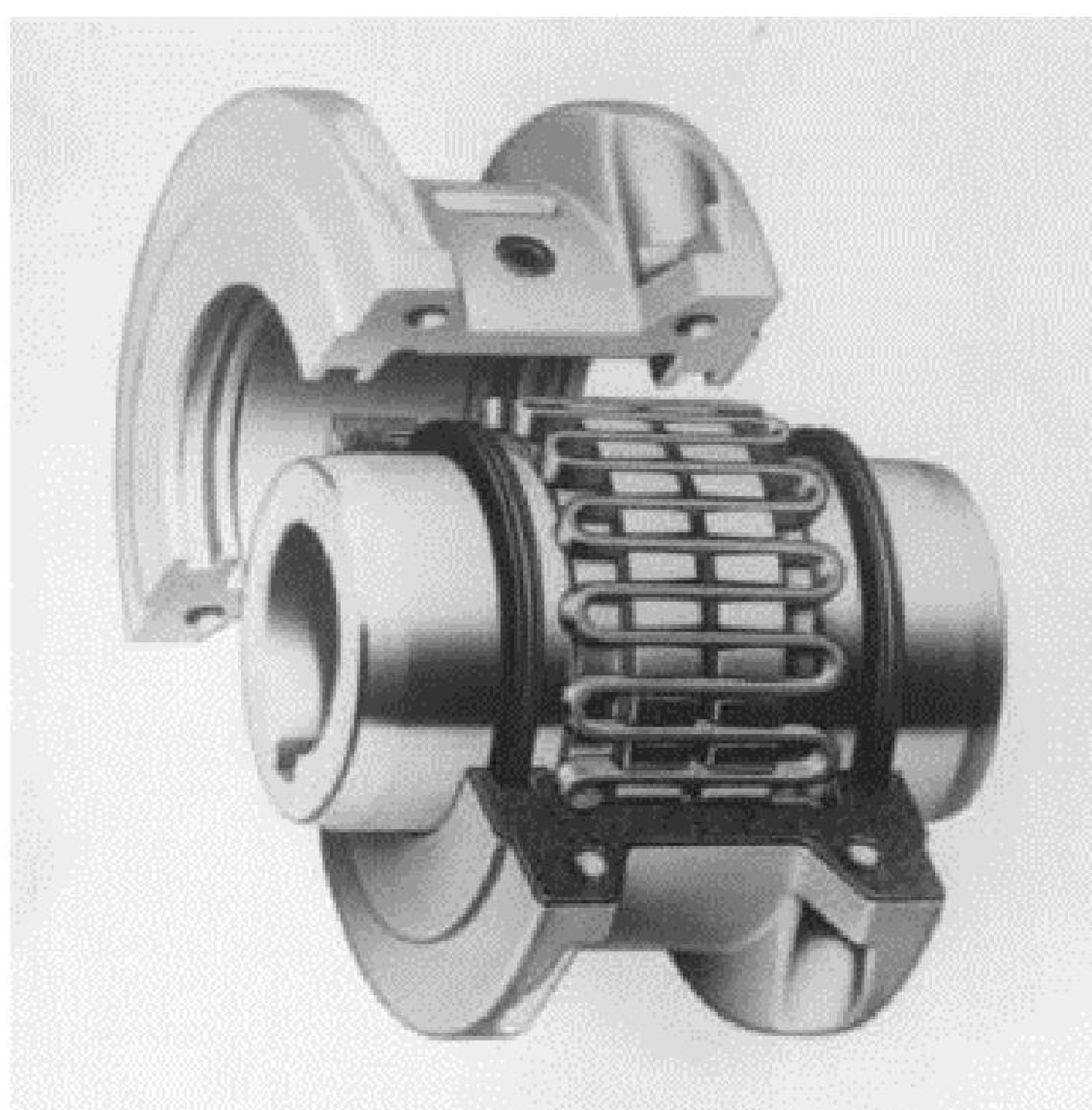


FIGURE 3.6 Metallic-grid coupling. (*The Falk Corporation.*)

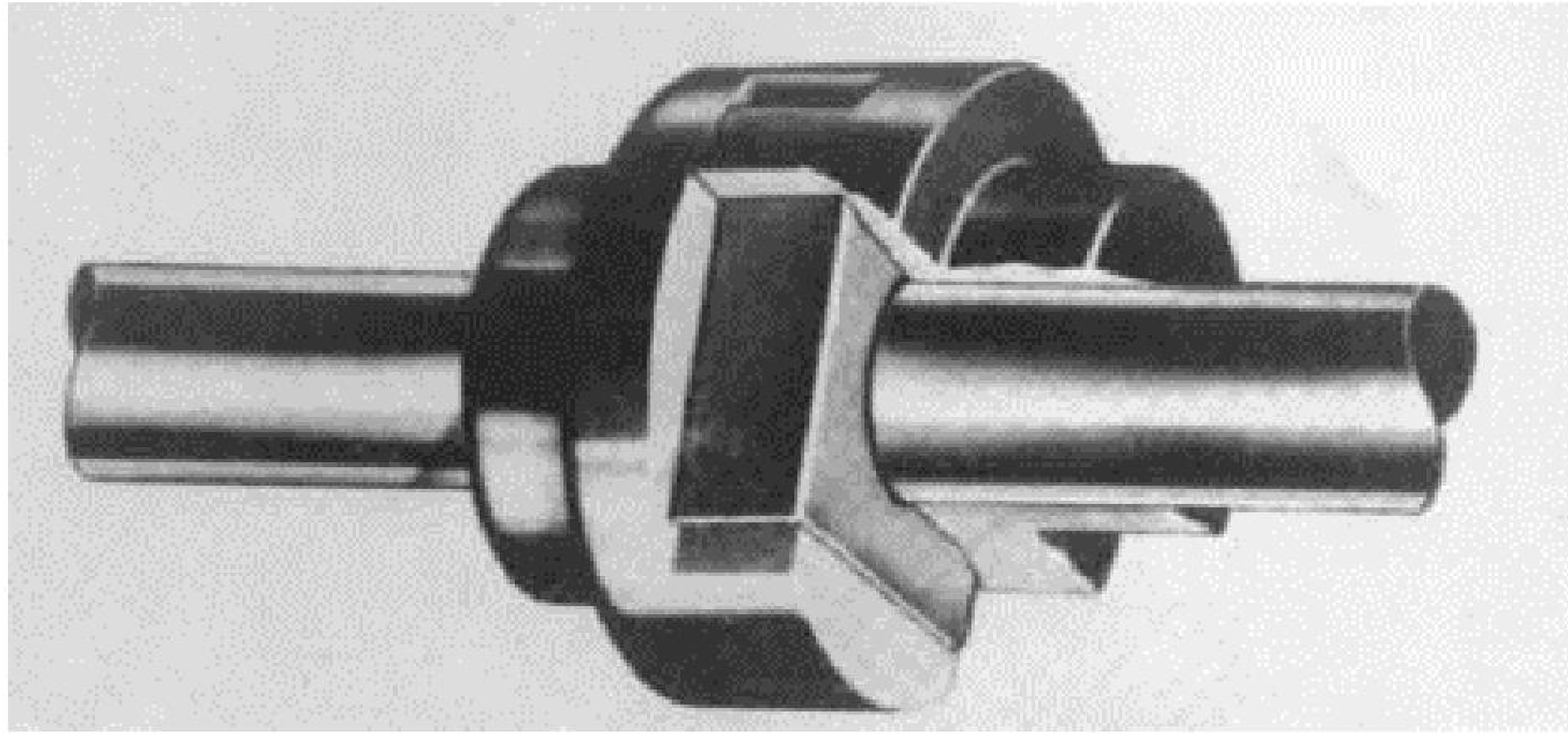


FIGURE 3.7 Oldham coupling. (*Zurn Industries, Inc.*)

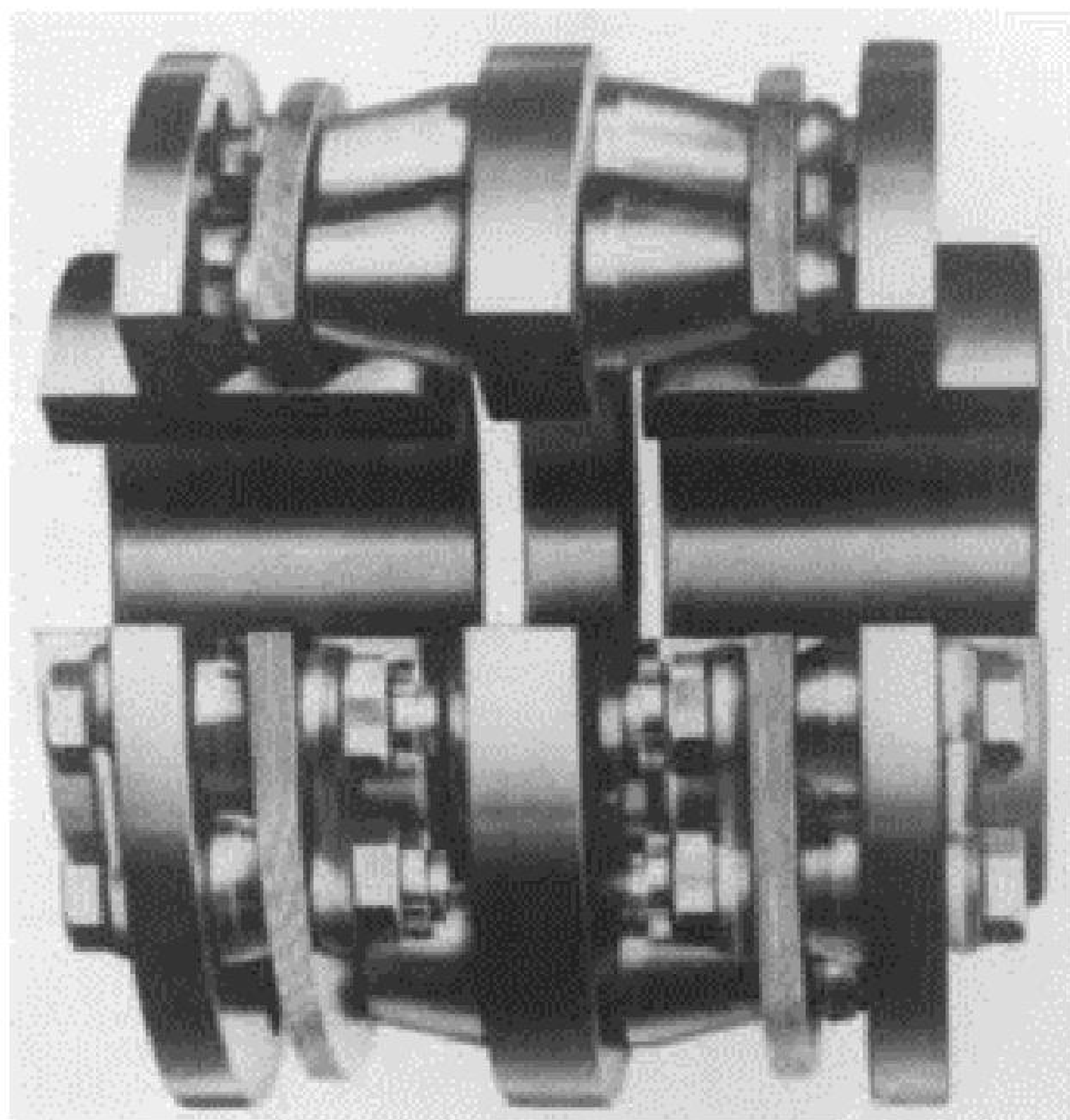


FIGURE 3.8 Laminated disc-ring coupling, standard double-engagement type. (*Rexnord Corporation, Coupling Division.*)

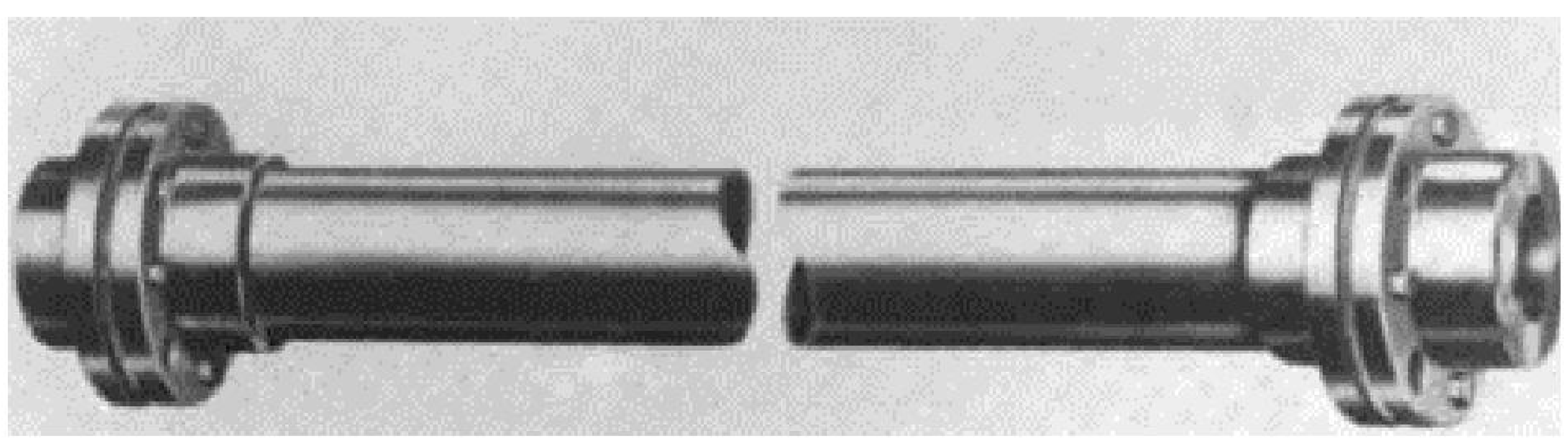


FIGURE 3.9 Laminated disc-ring coupling, spacer type. (*Rexnord Corporation, Coupling Division.*)

larger-sized couplings usually have hubs of cast iron and blocks of oil-impregnated sintered metal. On some styles, the bearing surfaces of the block are provided with replaceable nonmetallic strips. The block has a reservoir for lubricant which is fed to the bearing strips through orifices.

Shaft misalignment is provided for by slippage of the block between the jaw flanges. Alignment of the connected shafts should be within the coupling manufacturer's recommendation and usually can be accomplished with the use of a straightedge.

Laminated disk-ring couplings (Figs. 3.8 to 3.10) are the most prominent type in the material-flexing group and are available in a wide range of sizes and styles. They are capable of transmitting

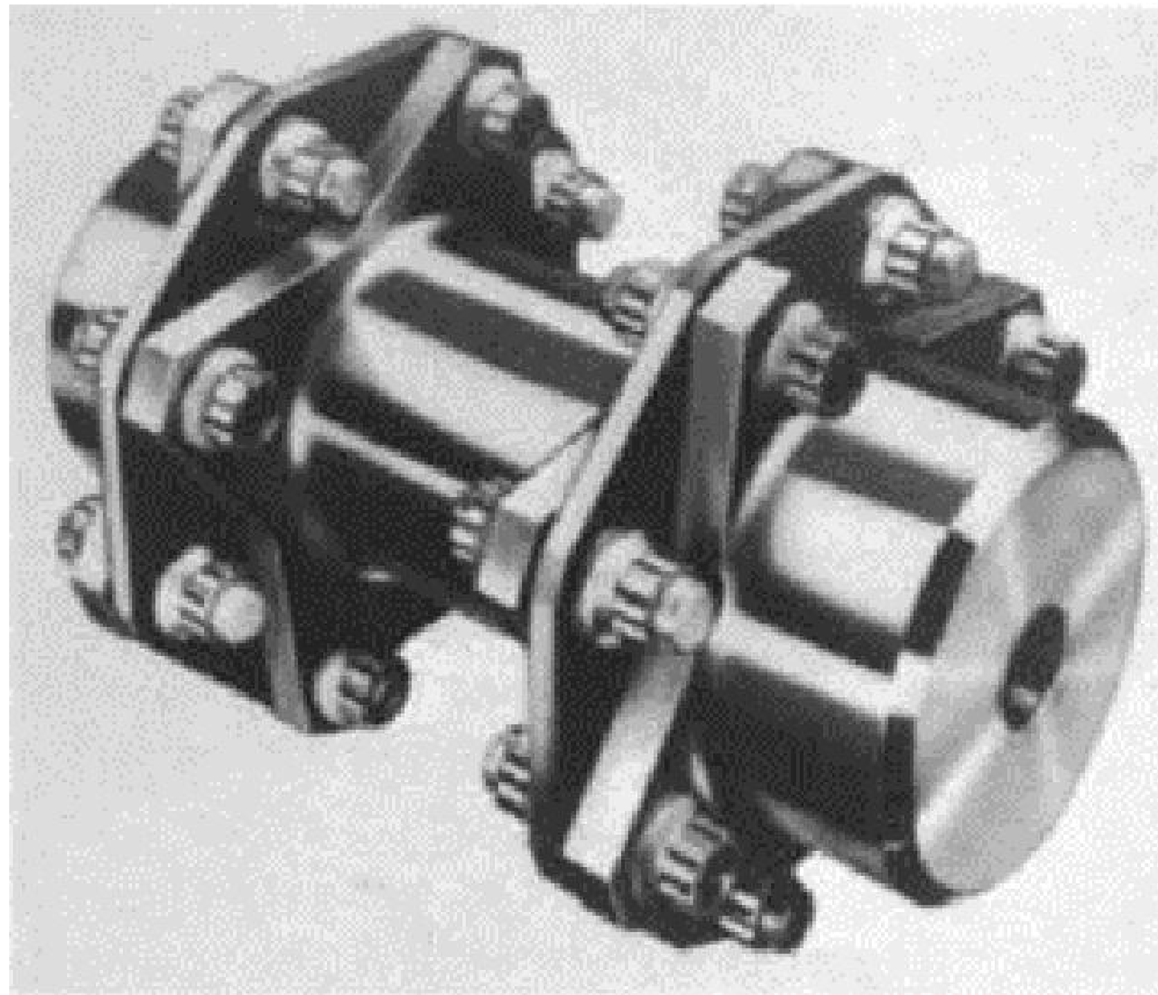


FIGURE 3.10 Laminated disc-ring coupling, high-speed spacer type. (*Rexnord Corporation, Coupling Division.*)

proportionately high torques at either low or high speeds. In their most common double-flexing form, two flanged hubs are connected to a floating center member through laminated disk rings. Each of the disk rings is alternately bolted or riveted to a hub flange and center member. The disk rings in tandem allow the coupling to accommodate angular and parallel misalignment as well as a limited amount of end float. In their single-flexing form, they consist of two flanged hubs and one laminated disk ring. The disk ring is alternatively bolted to the flanged hubs. These single-flexing units are capable of supporting a radial load and provide concentricity of connected three-bearing assemblies. They will accept only angular misalignment and a reduced amount of end float. Shaft misalignment is provided for by flexure of the disk rings. Since these units are normally of all-metal construction, they are free of backlash and are relatively rigid in a torsional plane.

Under normal conditions, the metal parts are not subject to deterioration. Most manufacturers have available couplings that are resistant to corrosion. These units usually have the components plated or are made of a corrosion-resisting material such as stainless steel. Laminated disk-ring couplings have no sliding parts that can wear, so no maintenance is required other than occasionally checking the condition of the laminated disk rings to make sure that all bolts are tight and that the equipment is still in proper alignment. Periodic visual inspection of the condition of the coupling is recommended. This can be done without disassembly or disturbing the connected equipment. When the equipment cannot be shut down conveniently, a stroboscopic light can be used. During inspection, special consideration should be given to the outer sheets of the disk ring. If any deterioration or broken sheets are found, the entire disk ring should be replaced. Significant deterioration and breaking of the sheets are normally indications of excessive flexure due to misalignments beyond the coupling's capacity. Realignment of the equipment must be done immediately. If a coupling has been operating with loose bolts, they should be removed and inspected. If there are significant scour marks or indentations on the body, the bolts should be replaced. Most couplings of this style are completely repaired.

Misalignment of the connected shafts should be restricted to within the manufacturer's recommendations. When accurate measurements are required, dial indicators should be used. Alignment with a caliper and straightedge is usually satisfactory for slow-speed drives.

Diaphragm couplings (Figs. 3.11 and 3.12) are also in the material-flexing group and are used primarily for the high-speed, high-horsepower applications. They are relatively light for the horsepower transmitted. The diaphragm coupling is available in many sizes and styles, including a reduced-moment design. This coupling uses two flexing elements separated by a floating center member. The diaphragm is normally attached at the outside diameter and the inside diameter by bolts or E.B. welding to connect the hubs to the floating center member. The torque goes through the diaphragm assembly for the outside diameter to inside diameter, or vice versa. The flexibility of

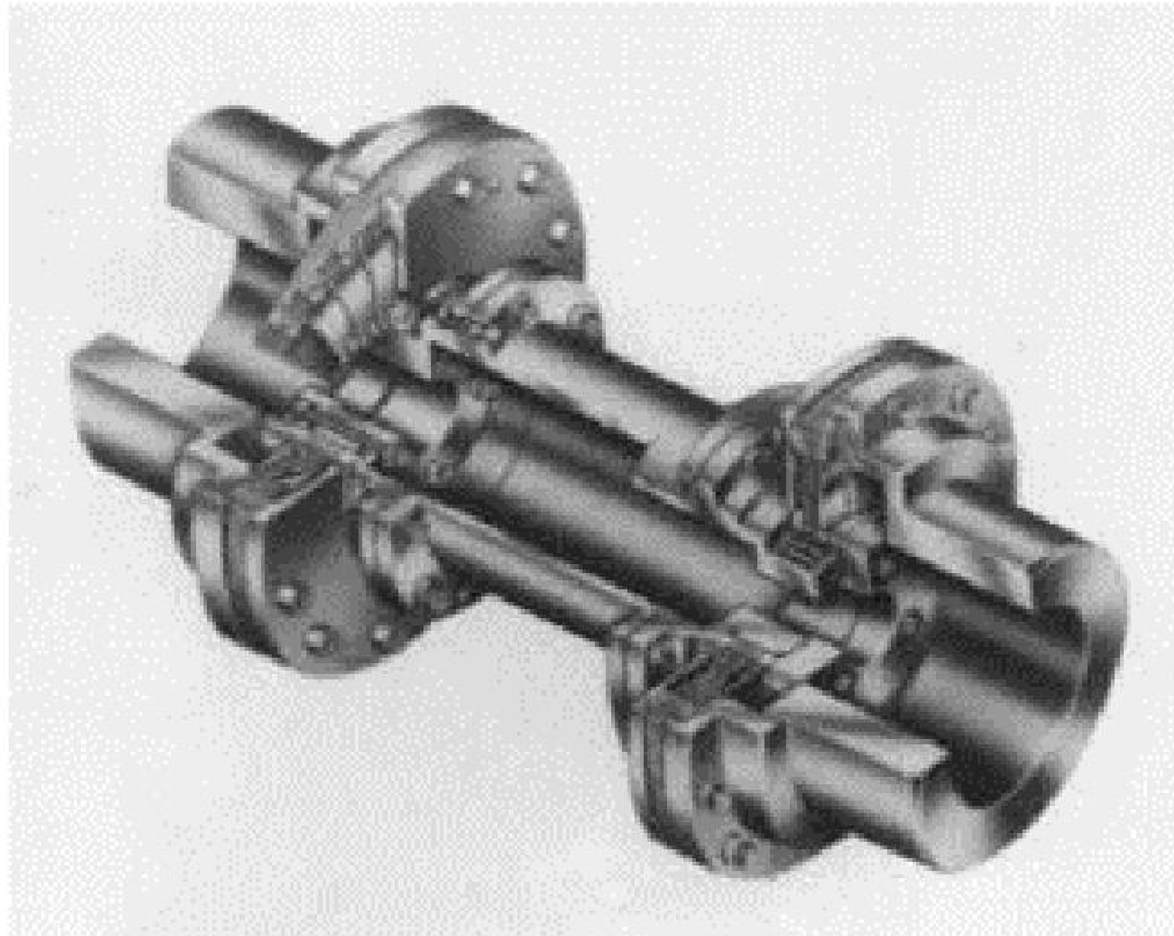


FIGURE 3.11 Multielement diaphragm coupling. (*Zurn Industries, Inc.*)

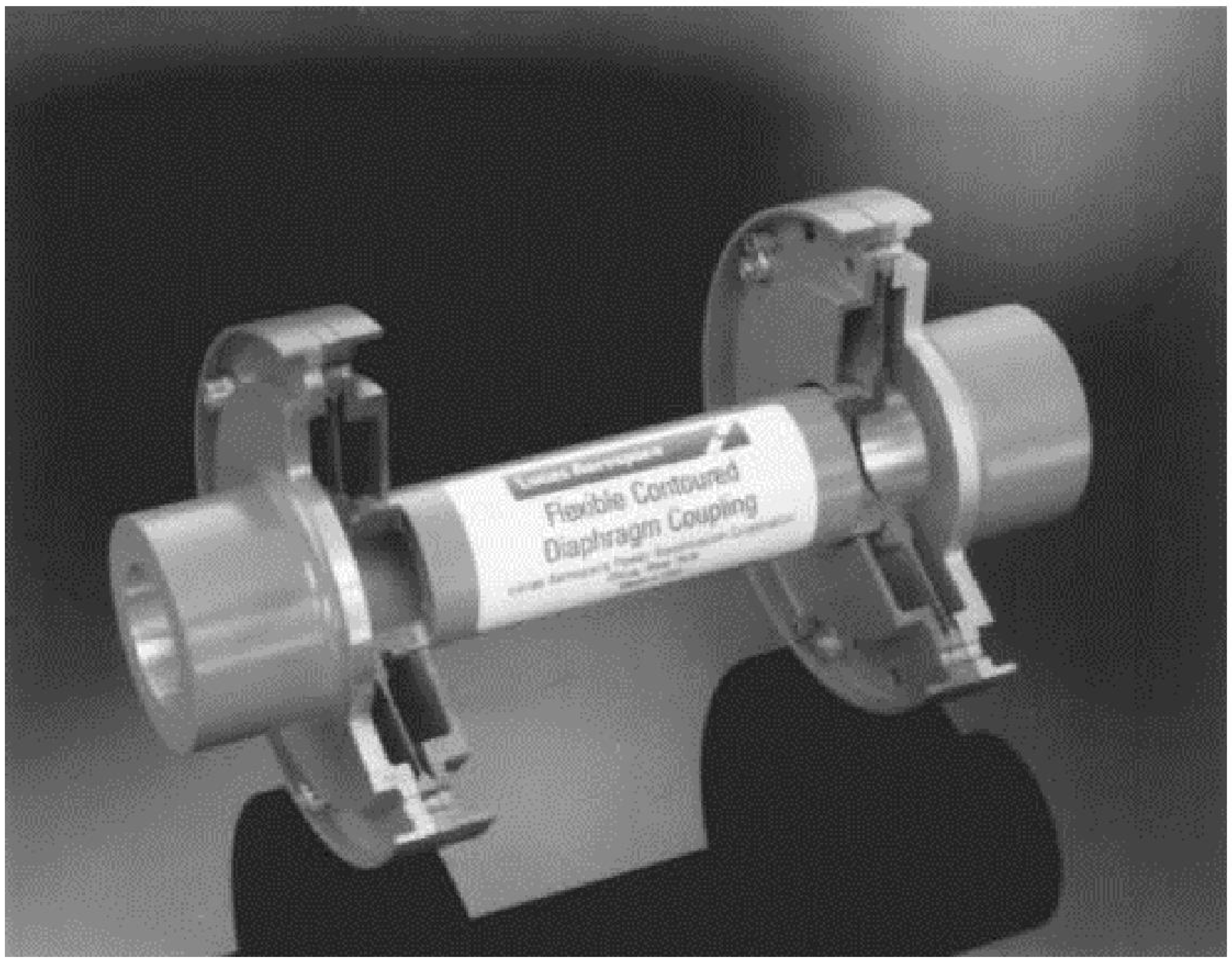


FIGURE 3.12 Single-element diaphragm coupling. (*Lucas Aerospace Corporation.*)

the diaphragm design accommodates angular and parallel shaft misalignment as well as a limited amount of end float. Each flexing element is made up of one or more diaphragm elements depending on the design. The coupling is radially rigid and maintains its original balance because there are no wearing parts.

Under normal conditions, the metal parts are not subject to deterioration. Most manufacturers have available couplings that are resistant to corrosion. These units usually have the components plated or are made of a corrosion-resisting material.

Misalignment of the connected shafts should be restricted to within the manufacturer's recommendation for long coupling life. If the connected equipment experiences high vibration, the coupling should be inspected for possible damage.

Elastomeric couplings (Figs. 3.13 and 3.14) are available in an almost infinite number of versions. They are generally categorized into two types. There are those in which the elastomer is placed

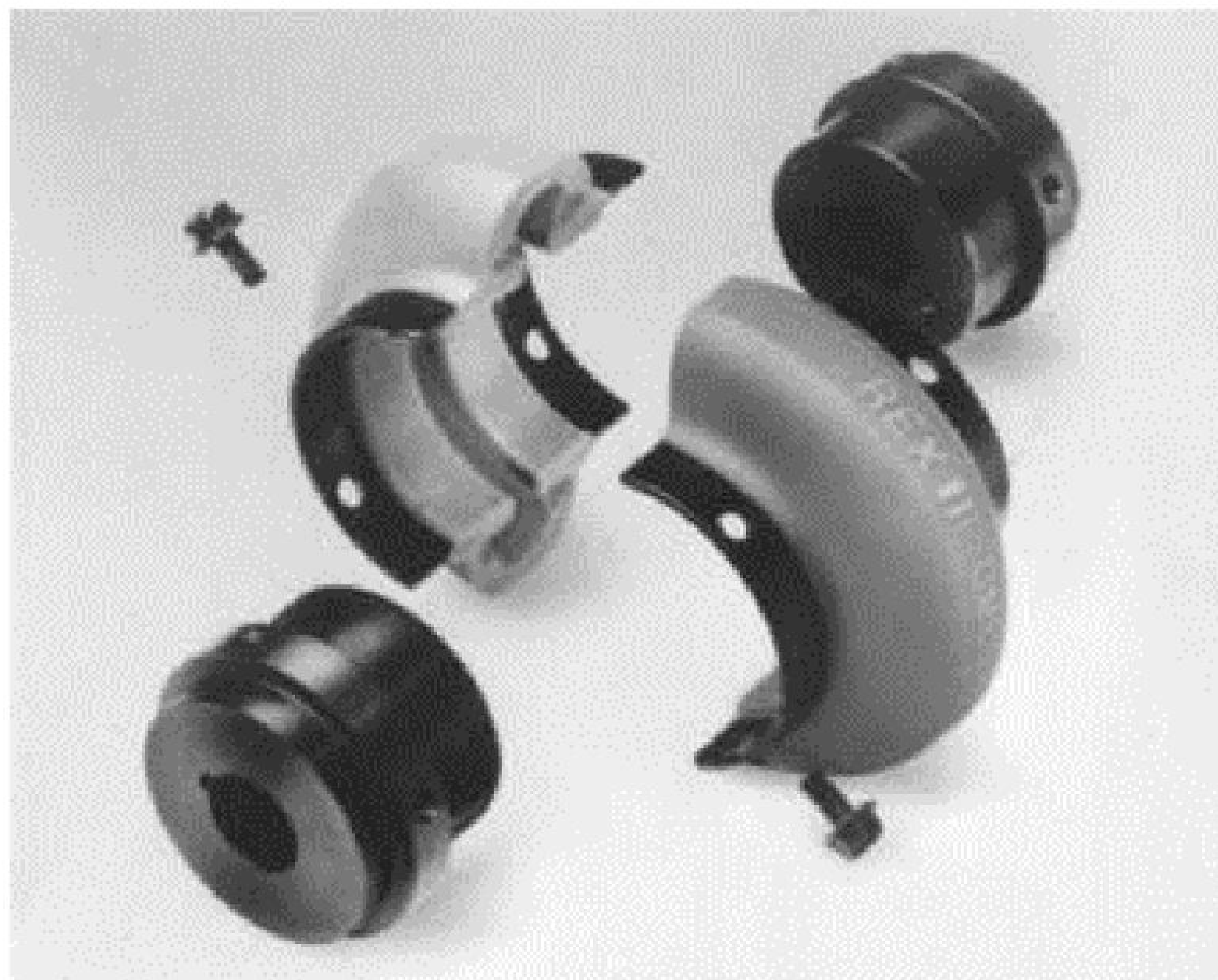


FIGURE 3.13 Elastomeric coupling, shear-type flexing element. (*Rexnord Corporation, Coupling Division.*)

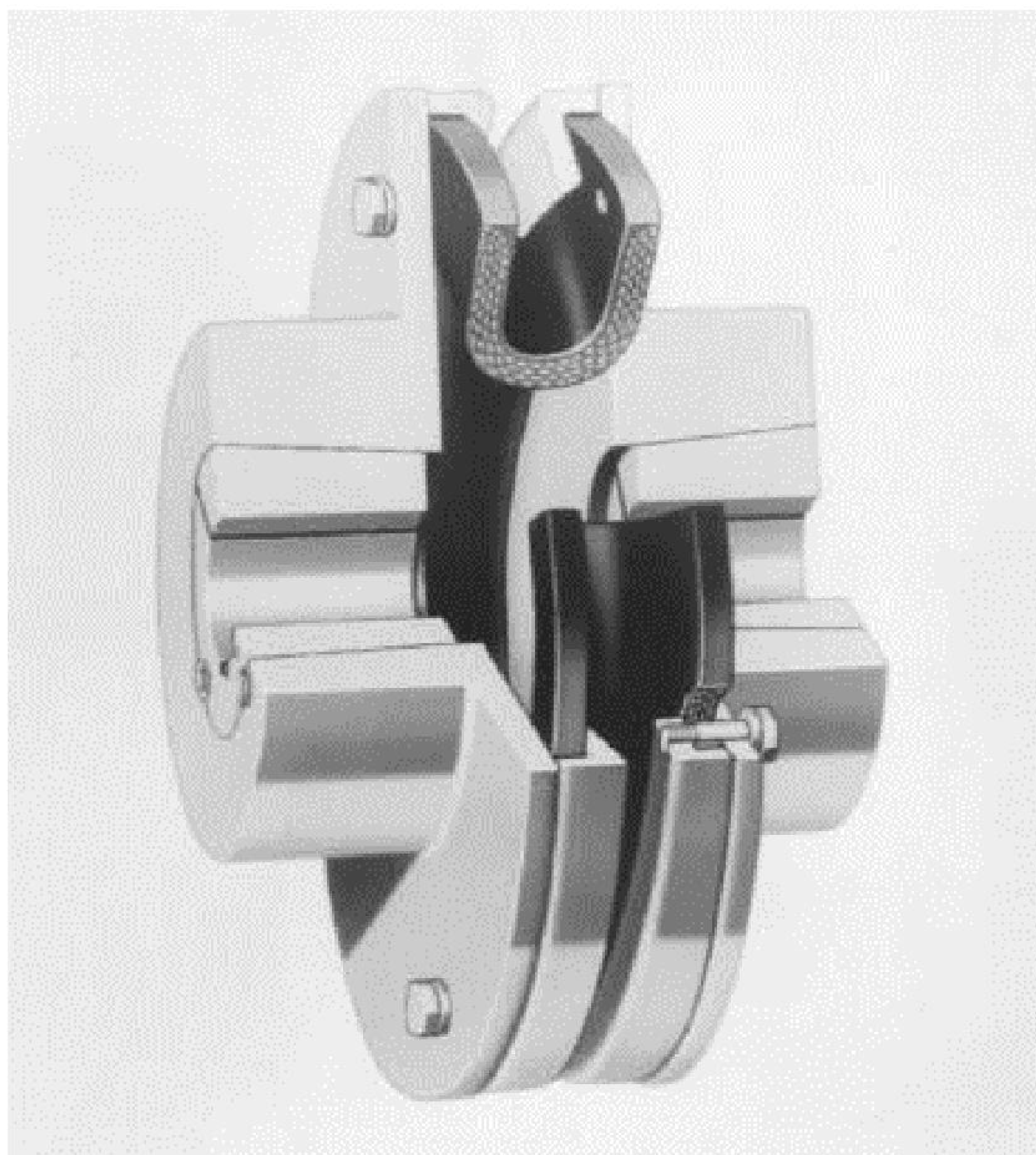


FIGURE 3.14 Elastomeric coupling, shear-type flexing element. (*The Falk Corporation.*)

in shear and those in which it is placed in compression. Their ability to compensate for shaft misalignment is obtained by flexure and/or displacement of the elastomeric element. These couplings are generally relegated to light- or medium-duty service at moderate speeds.

In their basic concept, they consist of two hubs separated and connected by the elastomeric element. On shear-type couplings, the elastomer may be bonded, clamped, or fitted to matching sections of the hubs. The compression-type couplings usually utilize projecting pins, bolts, or lugs to connect the components. The elastomeric flexing elements may be polyurethane, rubber neoprene, or impregnated cloths and fibers.

Elastomeric couplings are normally maintenance-free, but it is suggested that occasional checks be made of the elastomer's condition and equipment alignment. If the elastomer shows signs of deterioration or wear, it should be replaced and the equipment aligned to within the coupling manufacturer's recommendations. This usually can be accomplished with the use of a caliper and straightedge.

CAUSES OF COUPLING FAILURE

In the event of a coupling failure, a thorough investigation should be made to determine the cause. Failure may be due to either faults within the coupling itself or external conditions.

Most failures due to internal faults are the result of improper or poor machining. They usually have to do with concentricities, squareness of the mating face, and tolerances on the various piloting or registering diameters. Defective materials and materials with inadequate strength and/or hardness also have contributed to many premature failures. Another major cause of failure due to internal faults is improper product design. On mechanical-flexing couplings, the major problem is to provide adequate lubrication between the sliding contact faces, since lack of a lubricating film between these high-pressure surfaces will result in rapid wear. On material-flexing couplings, improper design of the flexing-element section and method of attachment to the hubs are the main causes of premature fatigue.

Most common causes of failure due to external conditions have to do with improper selection, improper assembly, and excessive misalignment. Consideration of these is given in the following pages.

Coupling Selection

Maintenance personnel are frequently faced with the problem of replacing a worn-out or broken coupling. After the cause of failure has been determined, careful consideration should be given to the type, size, and style of coupling that will be used as a replacement. Whenever possible, it should satisfy all the needs of the drive.

Proper selection as to type of coupling is the first step of good maintenance. A well-chosen coupling will operate with low cross-loading of the connected shafts, have low power absorption, induce no harmful vibrations or resonances into the system, and have negligible maintenance costs. The primary considerations in selecting the correct type of flexible couplings, as well as its size and style, are

1. Type of driving and driven equipment
2. Torsional characteristics
3. Minimum and maximum torque
4. Normal and maximum rotating speeds
5. Shaft sizes
6. Span or distance between shaft ends
7. Changes in span due to thermal growth, racking of the bases, or axial movement of the connected shafts during operation
8. Equipment position (horizontal, inclined, or vertical)
9. Ambient conditions (dry, wet, corrosive, dust, or grit)
10. Bearing locations
11. Cost (initial coupling price, installation, maintenance, and replacement)

The coupling should be selected conservatively for the torque involved. Consideration must be given to all peak and shock loads encountered in normal service. If the coupling is to operate at high

speeds, it should be dynamically balanced. Special coupling modifications dictated by the connected equipment should be made.

If any doubt exists as to the proper type or size of coupling to use, it is recommended that the manufacturer be consulted. Most manufacturers have representatives in all large cities, and they are usually qualified to make recommendations and assist in the coupling procurement.

Installation

On most applications, it is necessary to disassemble the coupling before installation. The arrangement of the components should be noted, since they must be replaced in the same order.

The driving and driven shafts, as well as the bore in the hubs, should be inspected to make sure they are free of burrs, dirt, and grit. Check the keys for proper fit in both shafts and hubs. Clearance over the key is essential. Normal practice uses 0.005 in. of clearance.

Next, the hubs should be mounted on their respective shafts. If an interference fit has been specified, it will be necessary to heat the hubs in water, oil, or a furnace and quickly position them on the shafts. Spot heating with a sharp, concentrated flame must be avoided because it will cause distortion and affect the capabilities of the material.

Finally, the equipment should be brought into its approximate operating position and the coupling reassembled. The equipment is now ready for alignment.

Alignment

Remember that misalignment is the major cause of coupling problems. Therefore, machines connected by a flexible coupling should be aligned with the greatest possible accuracy. The better the initial alignment, the more capacity the coupling has to take care of subsequent operational misalignment. Changes from the initial condition can occur through pipe strain, bearing wear, settling of foundations, base distortion due to torque, thermal changes, and vibrations in the connected equipment. To get the potential life of the coupling, the alignment of the connected machines should be checked at regular intervals and corrected as necessary.

The closer you get the hot running alignment, the better the connected equipment will run, giving longer bearing and seal life.

Normally, there are three conditions of misalignment that a flexible coupling must accommodate. These are angular misalignment, parallel misalignment, and axial misalignment (end float). These conditions combine to form the results shown in Fig. 3.15. No specific alignment procedure can be used on all drives. They must be worked out individually to suit the conditions at hand. There are fundamentals that do apply to all, however.

The first and most often overlooked step in equipment alignment is to bring the shafts into their proper axial position (Fig. 3.16). The shaft gap must be in accordance with the coupling manufacturer's recommendations. The mating surfaces or flexing elements of the coupling must be in their normal or relaxed position when the shafts of the connected equipment are in their operating position.

Some equipment has shafts with an inherent freedom to float. A typical example is the frequently encountered sleeve-bearing motor. Such motors may have as much as ½-in. free shaft movement. Usually, the shafts are marked to indicate where they should be located relative to the bearing housing. When the shafts are unmarked, it should be assumed that the motor rotor is on magnetic center when it is located halfway between the float limits.

If the exact shaft gap for the coupling is not known, it usually can be obtained from the manufacturer's catalogs, certified prints, or data sheets. Most manufacturers furnish dimensional sheets and installation instructions with all new couplings. In the case of old couplings, this information is usually available upon request. If the gap dimension is not available, it usually can be obtained by measuring the coupling or its components and calculating the normal position. Slight adjustments to the initial position may be required during the later processes of eliminating angular and parallel misalignment. After the equipment has been properly spaced, it is ready for alignment.

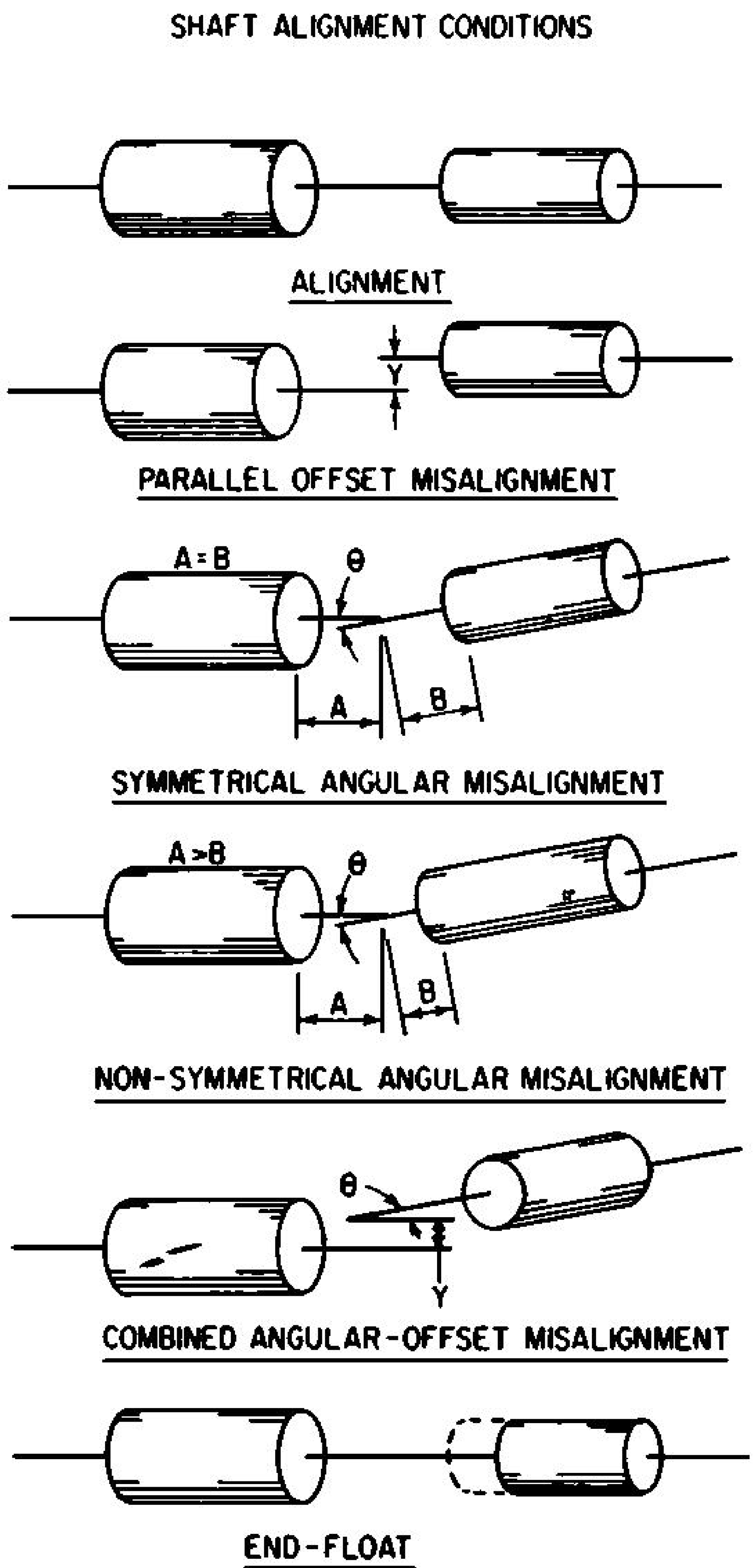


FIGURE 3.15 Shaft-alignment conditions.

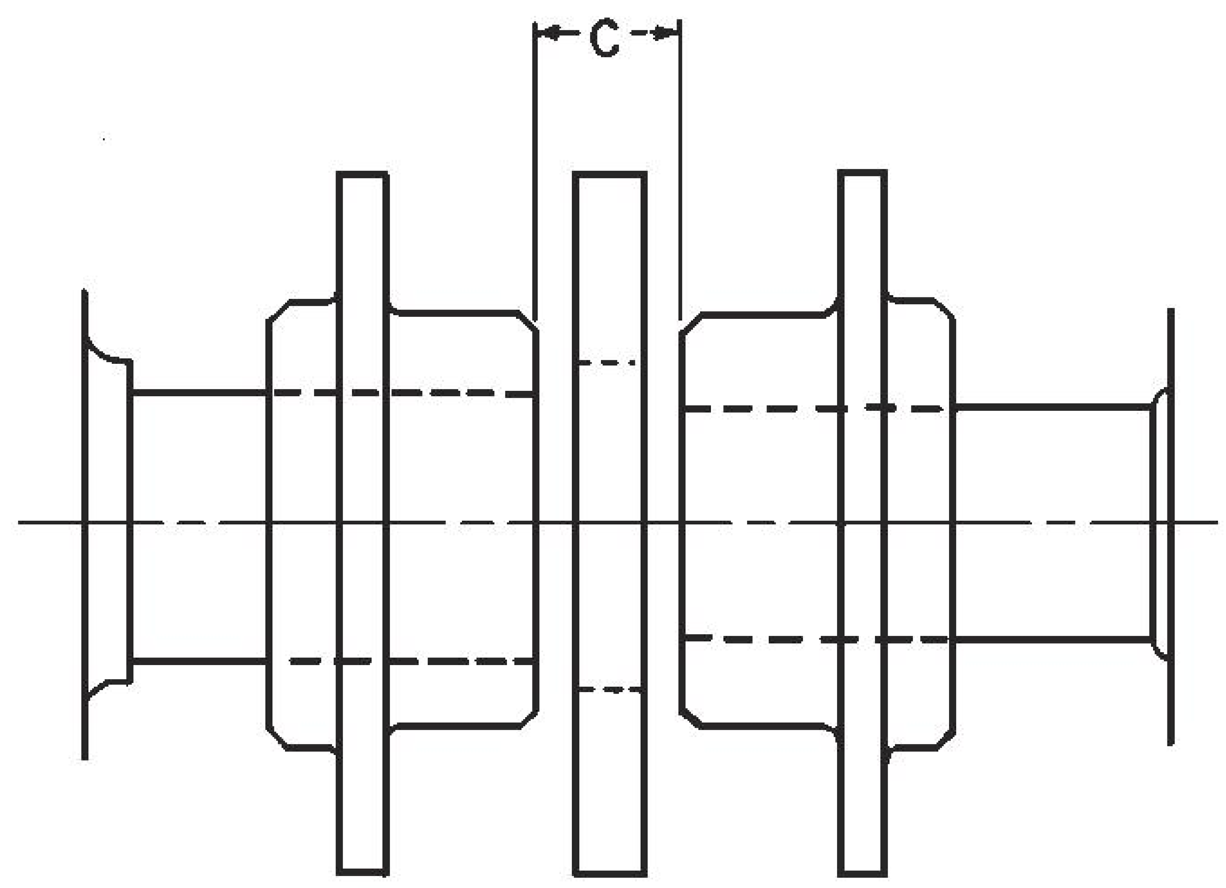


FIGURE 3.16 Axial spacing.

For many years, the accepted method of aligning equipment was with the use of a straightedge and caliper or scale, and this method is still frequently used with some types of couplings on low-speed drives. With the continuing development of small, high-speed equipment, however, the need for accurate measurements of misalignment has increased. On most drives, the use of a straightedge and scale has given way to dial indicators, optics, and lasers. When properly applied, the devices will give precise measurements as to the amount of misalignment as well as give its phase or direction.

No specific method can be established for mounting this equipment. This must be worked out to suit conditions at hand.

The preferred method of aligning equipment is to have the coupling mounted and completely assembled. The dial indicator then rotates with the equipment, but its stem remains in contact with a specific surface with no sliding of the stem over a large surface. In the following comments on alignment, it is assumed that the coupling is completely assembled and that the driving and driven pieces of equipment can be rotated together.

By graphically plotting the shaft misalignment, the solution becomes apparent. A picture is worth a thousand words.

Correct alignment is mandatory for successful operation of rotating equipment. A flexible coupling is no excuse for misalignment.

SHAFT CENTER LINE RELATIONSHIP

How do you easily determine the relationship of one shaft center line relative to the other? It is hard sometimes to visualize this.

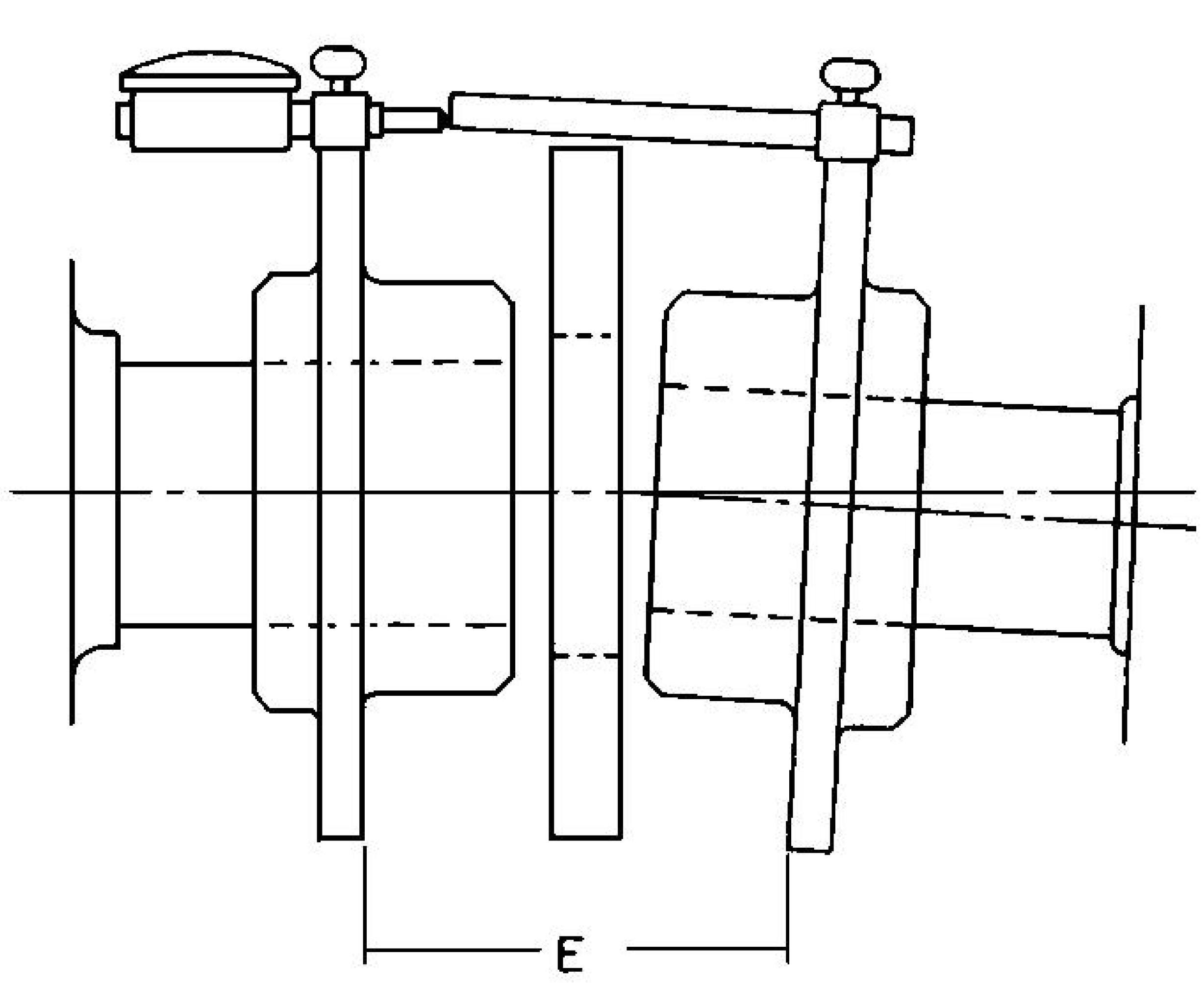


FIGURE 3.17 Using a dial indicator to check angular misalignment.

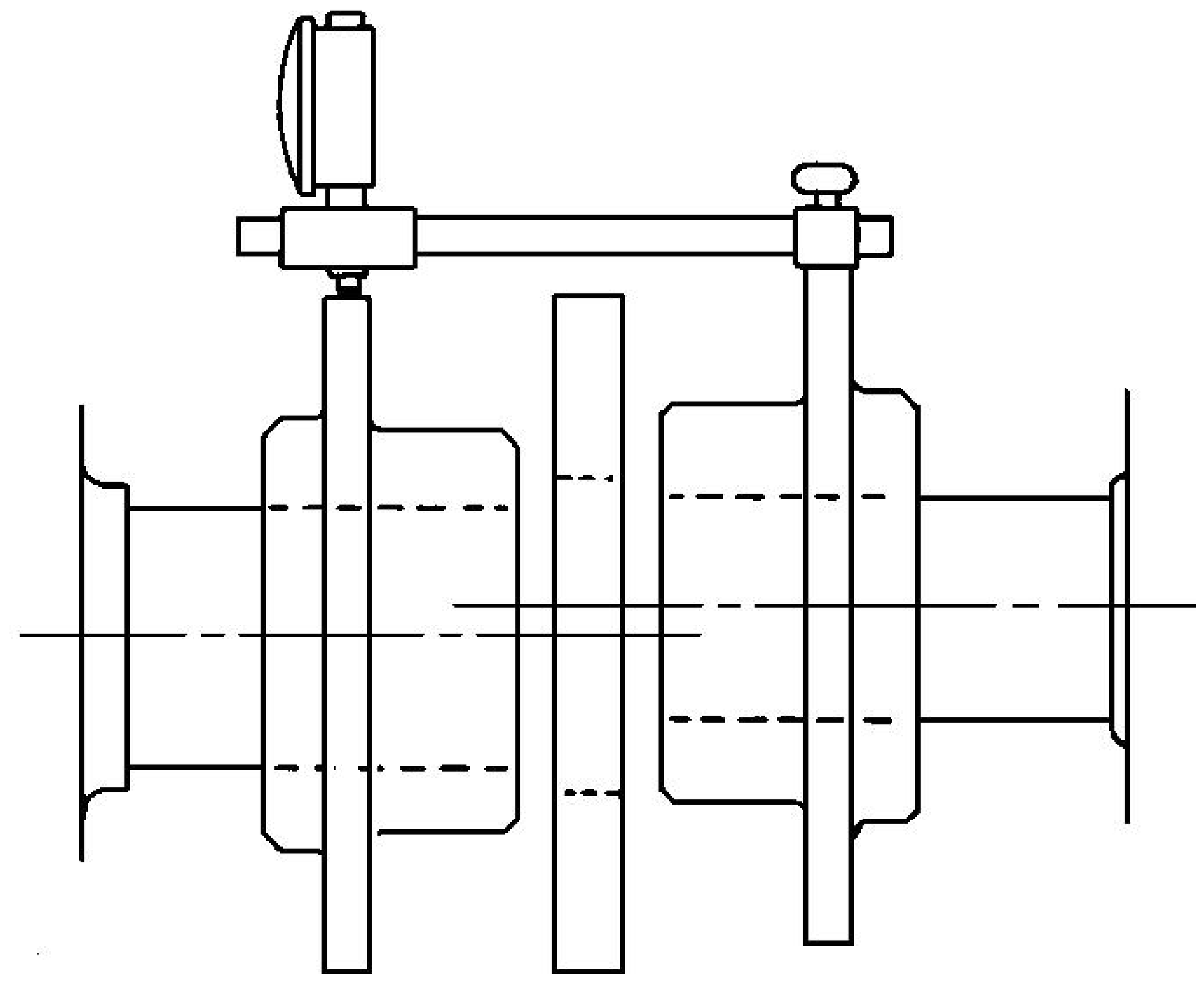


FIGURE 3.18 Using a dial indicator to check parallel misalignment.

This is done by drawing a picture on a piece of graph paper to show visually where the equipment is and how far it needs to be shifted to get it into perfect alignment.

There is some confusion between shaft alignment and coupling alignment. We will restrict the comments to couplings using two flexing elements. The coupling flex element sees angular misalignment only. It is possible, as you can see in Fig. 3.17, to have misalignment between two shafts, yet one end of the coupling can still be in perfect alignment with its center member.

This could be the situation if all the problems are at one end of the coupling. In Fig. 3.18, the two ends of the coupling share the misalignment equally.

SHAFT MISALIGNMENT

How do you go about correcting for shaft misalignment?

These instructions will be broken down into four sections:

1. Items that must be considered *before starting* any of the alignment procedures.
2. *Reverse indicator alignment* graphic analysis.
3. *Face and rim alignment* graphic analysis.
4. *Across the flex element* graphic analysis.

Before Starting

Before we get into the procedure itself, there are several items that must be considered before starting. See Figs. 3.19 and 3.20.

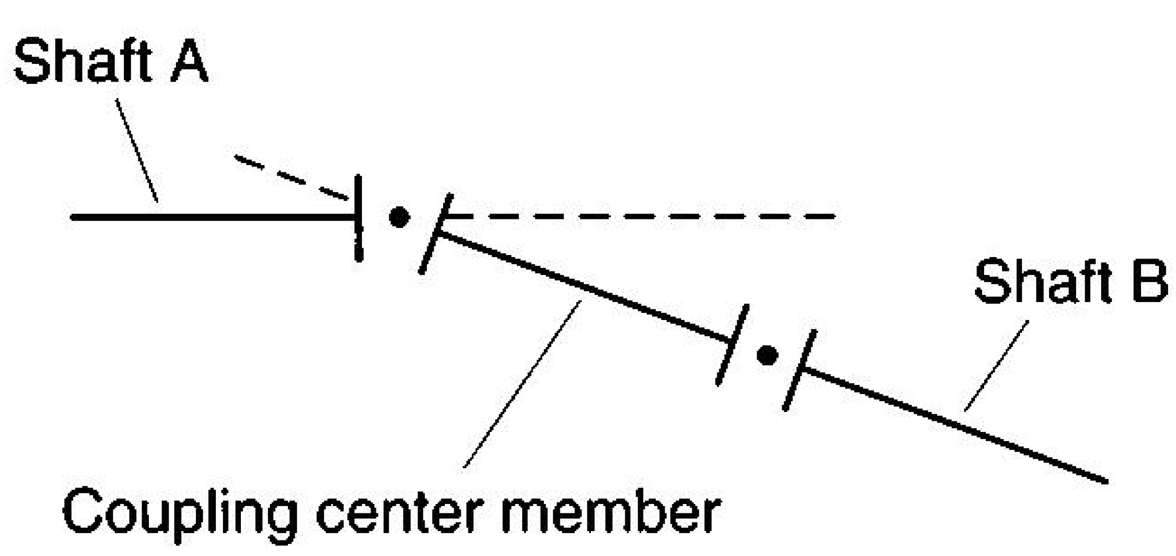


FIGURE 3.19 Coupling center member.

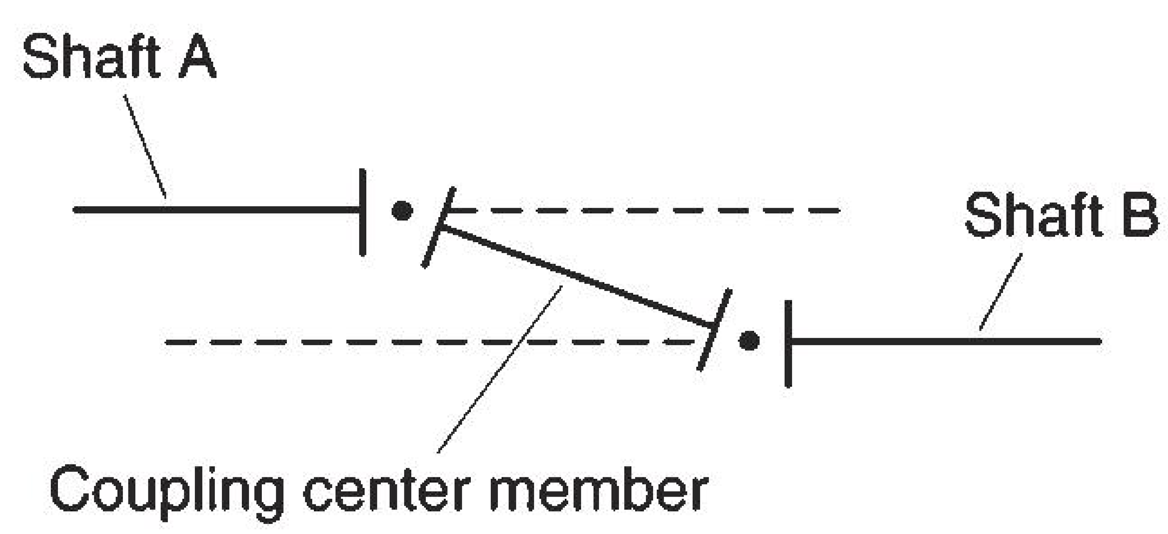


FIGURE 3.20 Coupling center member.

Soft Foot. Soft foot occurs when the equipment, say, a motor, is not sitting flat on its base or it rocks. Now this rocking can be eliminated by tightening all the hold-down bolts. What this does, however, is to put the motor bearing under strain. This, in turn, can cause vibration. It also may give erroneous alignment readings. The soft foot must be corrected first. This is easily done by shimming under the motor foot until it no longer rocks.

Indicator Sag. Calibrate the dial indicator setup sag. In other words, determine the difference in the dial indicator reading when it is on top of a shaft as opposed to when it is on the bottom. This is a gravitational effect. It is not necessary to eliminate sag but rather to know the amount of sag. The indicator setup should be as rigid as practical, and then it should be calibrated. It is easily calibrated by mounting the setup on a piece of pipe, allowing the dial indicator to ride on the pipe itself. See Fig. 3.21.

Set the indicator at 0 on top. Now roll the pipe over until the indicator is at the bottom of the pipe. The gravitational effect on the setup can be determined by reading the indicator deference from top to bottom. This delta reading can be subtracted algebraically from the alignment readings obtained at the bottom position.

In the example shown in Fig. 3.21, setup sag checked out to be 20.005 in. This reading will always be negative. The indicator setup sag does not have to be considered for the horizontal or side-to-side reading.

Alignment Readings. It is suggested that the dial indicator be zeroed at the top for convenience. The coupling hub should be marked at 0, 90, 180, and 270° with a reference mark on the equipment so that the units can be turned through 90° increments. This eliminates any runout that might exist between the point at which the indicator rides and the theoretical centerline of the shaft. Now rotate the coupling in 90° increments, recording all readings. It is important to keep the side-to-side readings straight. A suggestion is to refer to the sides of the unit as "near" and "far" ("near" being the side where you are standing). After making the four-position check, return to the top to make sure that the indicator returns to zero. If it does not, disregard the readings and repeat the procedure. It is a good practice to take several sets of readings to make sure they are consistent. It is a lot easier to take another set of readings than it is to move the units a second time. See Fig. 3.22.

Thermal Growth. Now consider any thermal growth values for the equipment. For example, if the pump is pumping hot water, it will grow vertically from the ambient to the hot running condition. The whole objective is to have the equipment in good alignment when it is running under normal operating conditions. These predicted thermal movements can be obtained from the equipment manufacturer and should be taken into account before making the alignment changes.

Shaft Relationship. How do you easily determine the relationship of one shaft center line relative to the other? It is hard sometimes to visualize this. To help, when using the reverse indicator method, refer to Fig. 3.22.

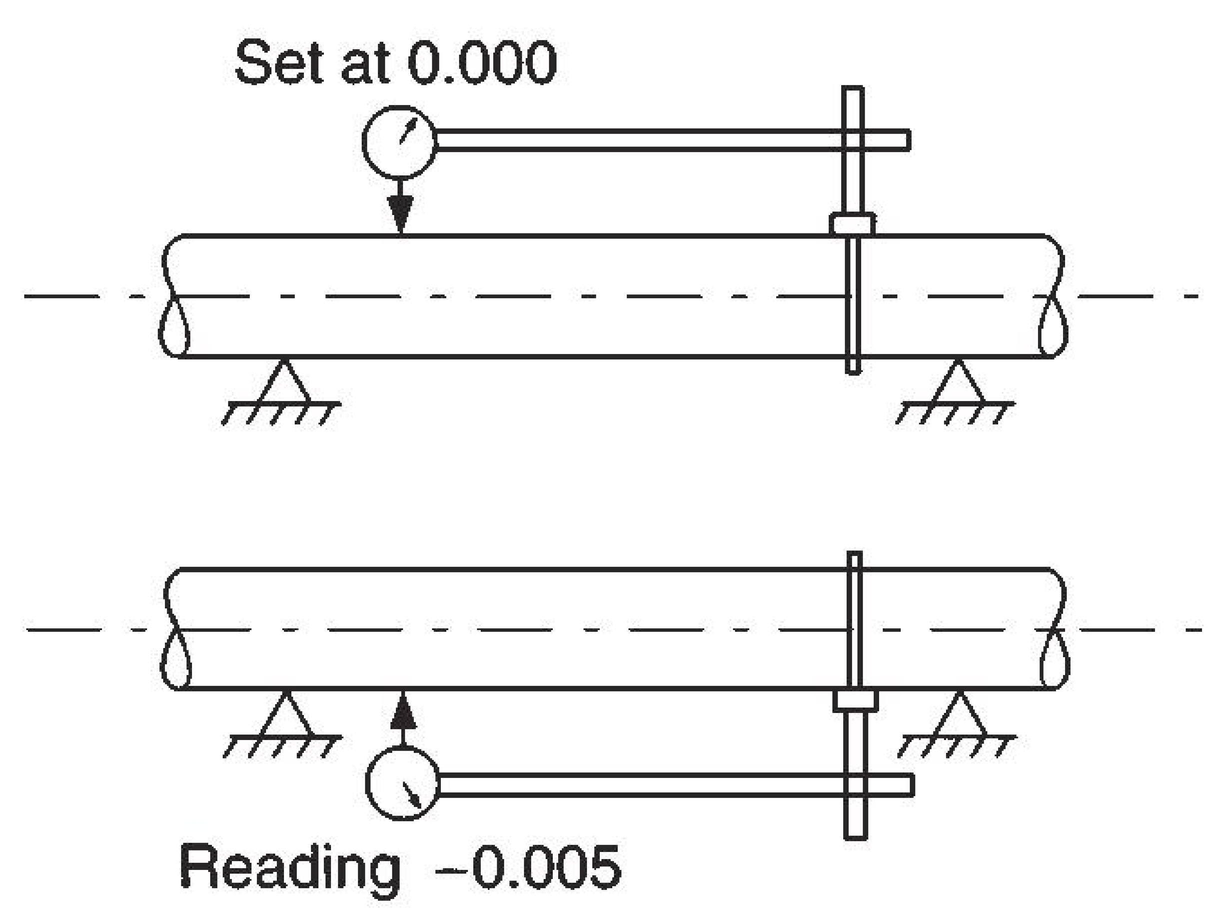


FIGURE 3.21 Indicator sag.

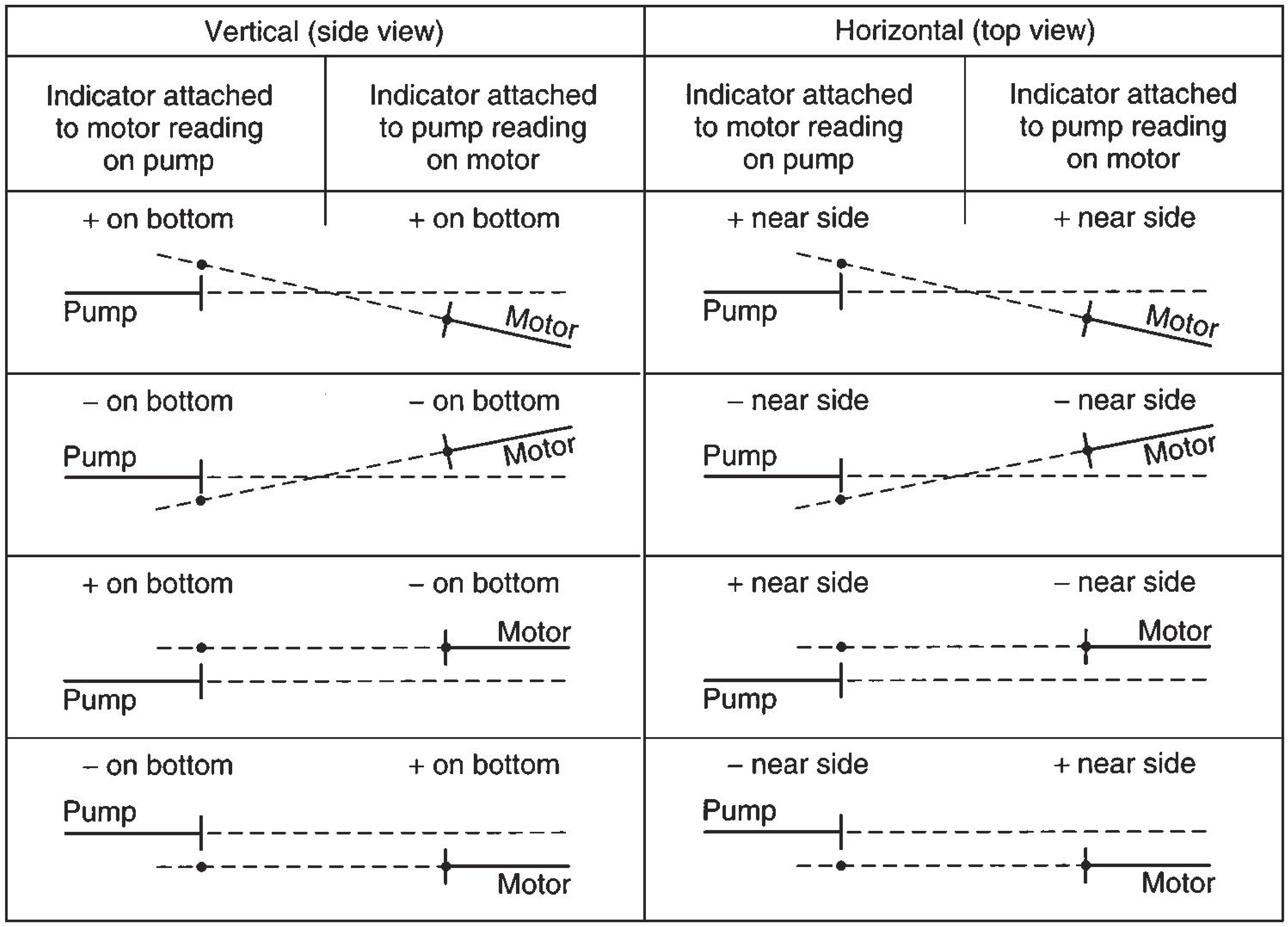


FIGURE 3.22 Pump-to-motor alignment guide.

Reverse Indicator

To explain the reverse indicator alignment procedure, a motor-to-pump example will be used. First, correct the vertical misalignment by shimming, and then correct the horizontal misalignment by sliding the equipment from side to side. With proficiency, these two steps can be done together.

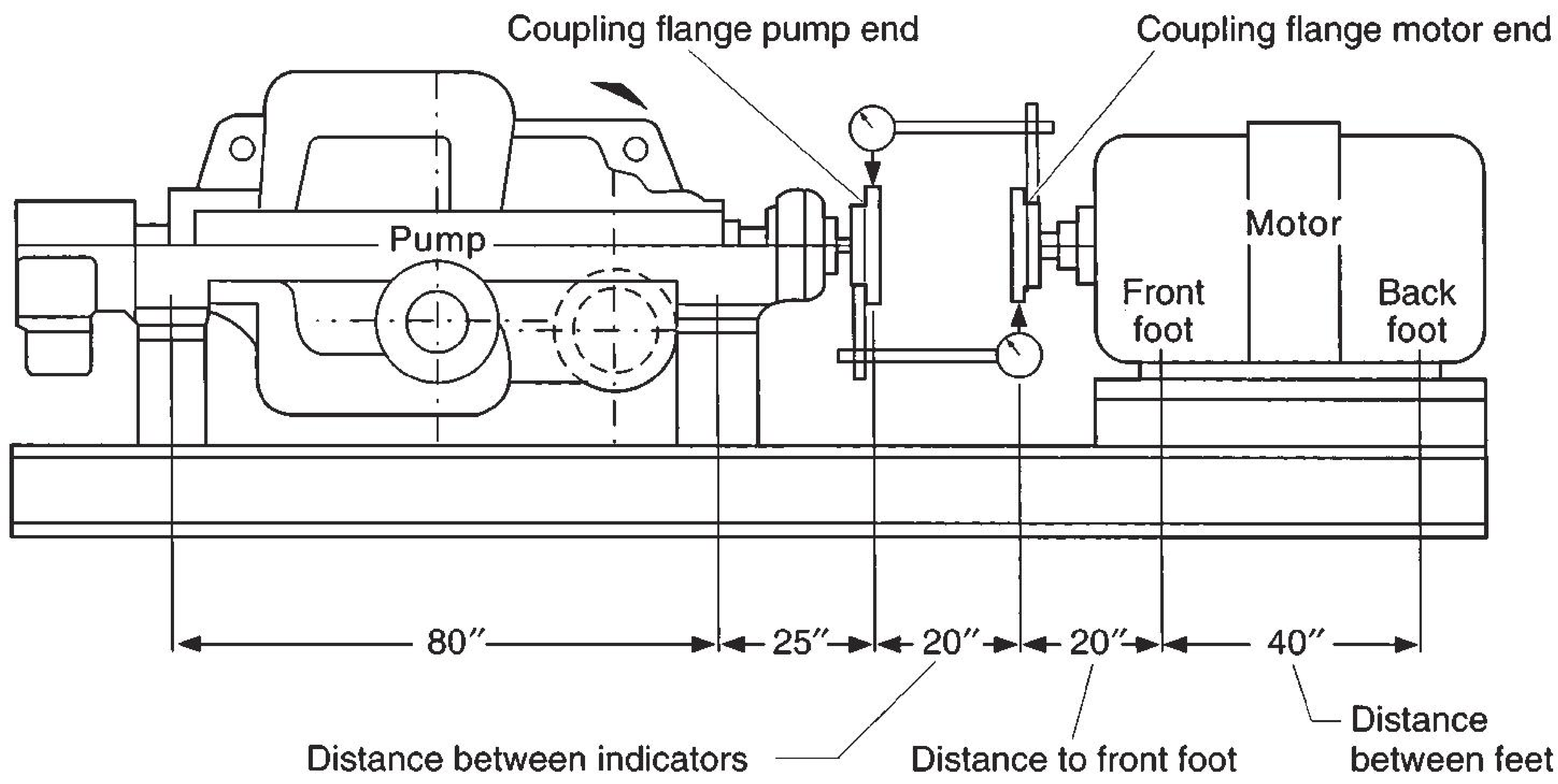
Before starting the alignment work, determine which piece of equipment is easiest to move. This is not to eliminate the option of moving both units if a problem occurs. The pump, in this example, will be fixed. Therefore, the motor will be moved into alignment with the pump.

Now, on a sheet of graph paper, lay out the equipment being aligned. See Fig. 3.23.

The horizontal scale on the graph used here is one small division equals 1 in. The distances needed are

1. The distance from the first indicator riding on the pump hub to where the second indicator is riding on the motor hub. In the example, this is 20 in.
2. The axial distance between the motor hub where the second indicator is riding and the center of the motor front foot. In the example, this is 20 in.
3. The distance from the center of the front motor feet to the center of the back motor feet. In this example, this is 40 in.

Vertical Alignment Solution. The alignment can be done either with the coupling totally installed, with the coupling hubs mounted, or with the coupling totally removed from the shafts. Find a spot to mount the dial indicator bracket. The shaft behind the hub or the hub itself is good. A chain-clamp alignment bracket usually fits in well. With the indicator bracket attached to the motor hub and the dial indicator reading off the pump hub, rotate both units in 90° increments and take readings. These are shown in Fig. 3.24.



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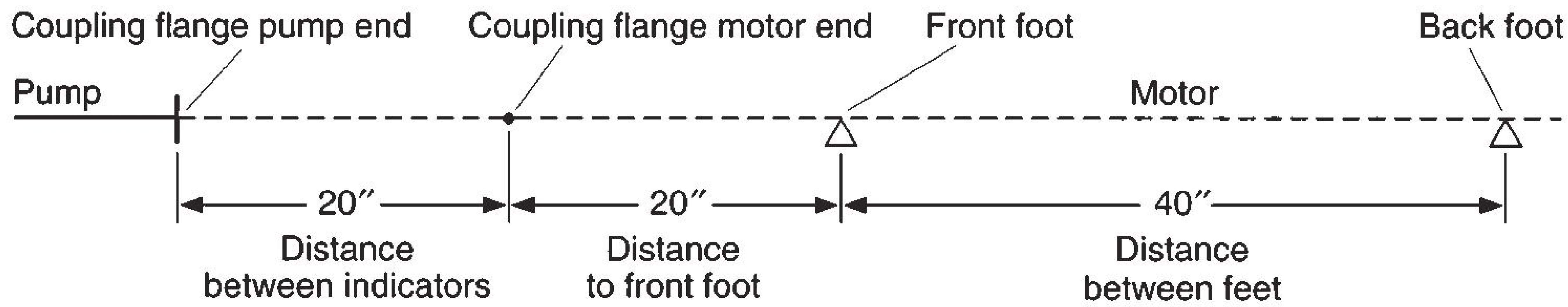

(*b*)

FIGURE 3.23

The bottom reading is then corrected for indicator setup sag. The indicator setup sag in this example was determined to be −0.005 in. Thus −0.005 was subtracted algebraically from the −0.025 indicator reading. The corrected reading is −0.025 −(−0.005) = −0.020. The readings taken are total indicator readings (T.I.R.), which are two times the actual shaft-to-shaft relationship. This means that the readings taken must be divided by 2. To solve for the vertical (up and down) part of the problem, take the corrected bottom reading of −0.020 and divide it by 2, that is, −0.020/2 = −0.010. With a minus reading on the bottom of the pump flange, the motor center the extension must be lower than the center line of the pump. For further clarification, see Fig. 3.22.

Use a convenient scale in the vertical of 0.001 in. per small division. Plot this point as shown in Fig. 3.24. Do nothing with the horizontal (near/far) readings at this time.

Reverse the bracket setup by attaching it to the pump hub, as shown in Fig. 3.25, with the dial indicator reading off the motor hub.

Take a set of alignment readings. In this case, the indicator shows a reading on the bottom of +0.005 in. Correct for indicator sag by algebraically subtracting the −0.005 indicator setup from the +0.005, which gives a +0.010 corrected reading. Now divide this number by 2, since it is a total indicator reading. With a plus indicator reading on the bottom, the motor is low relative to the theoretical center line of the pump. To help you to see the shaft-to-shaft relationship, refer again to Fig. 3.22. Use the same scale of 0.001 in. per small division. Now plot the +0.005 on the graph, as shown in Fig. 3.25.

The solution to the problem now becomes easy. Draw a line through the two points plotted, and extend it beyond the plane of the motor feet, as shown in Fig. 3.26.

Then, by counting the graph graduations in line with each of the motor feet, it can be seen that the motor must be lowered by 0.010 in. at the back foot and left alone at the front foot. By making this correction, the motor center line extension falls on top of the pump center line. This would solve

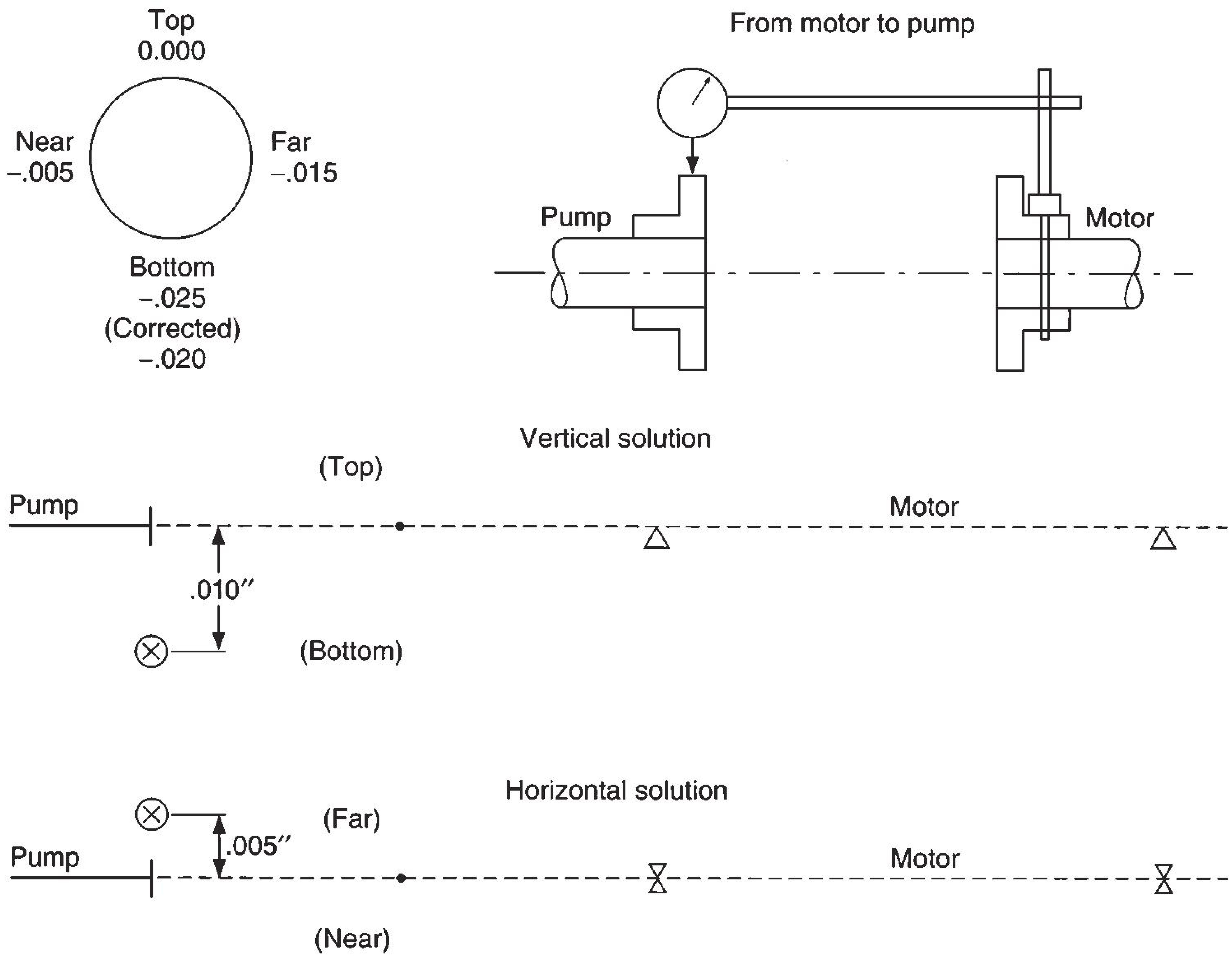

FIGURE 3.24

the vertical alignment problem if there was no thermal correction to be made for either the pump or the motor.

Note: If the vertical scale chosen is too big, it may realize some minor shimming errors.

Let's say that the pump does grow 0.005 in. at both ends because it is handling a hot liquid. In order to compensate for this, add an additional 0.005 in. to each motor foot in order to have the motor in line with the pump when the pump is running under normal operating conditions. The total solution now is to add 0.005 in. of shims to the front motor feet and subtract 0.005 in. of shims from the back motor feet. The vertical alignment is now solved.

Horizontal Alignment Solution. In a similar manner, the side-to-side movements can be plotted. Refer to Fig. 3.24, and imagine looking straight down on the pump-motor combination. The alignment readings obtained were −0.005 in. on the "near" side (this is the side you are standing on) and −0.015 in. on the "far" side. By adding a value of 0.015 to each side, the "far" side becomes 0.000 and the "near" side becomes +0.010 T.I.R., or +0.010/2 × +0.005 in. actual. With a plus on the "near" side, the motor center line extension falls on the "far" side of the pump center line. See Fig. 3.22.

Now plot this point as shown in Fig. 3.24. Moving on to Fig. 3.25, the side-to-side alignment readings obtained were +0.006 in. on the "near" side and +0.004 in. on the "far" side. By subtracting 0.004 from each side, the "far" side becomes 0.000 and the "near" side becomes +0.002 T.I.R., or 0.002/2 = 0.001 in. actual. With a plus on the "near" side, the motor center line is on the "near" side of the pump center line extension. See Fig. 3.22.

Now plot this point as shown in Fig. 3.25

Now draw a line through the two plotted points, extending it past the front and back motor feet, as shown in Fig. 3.26.

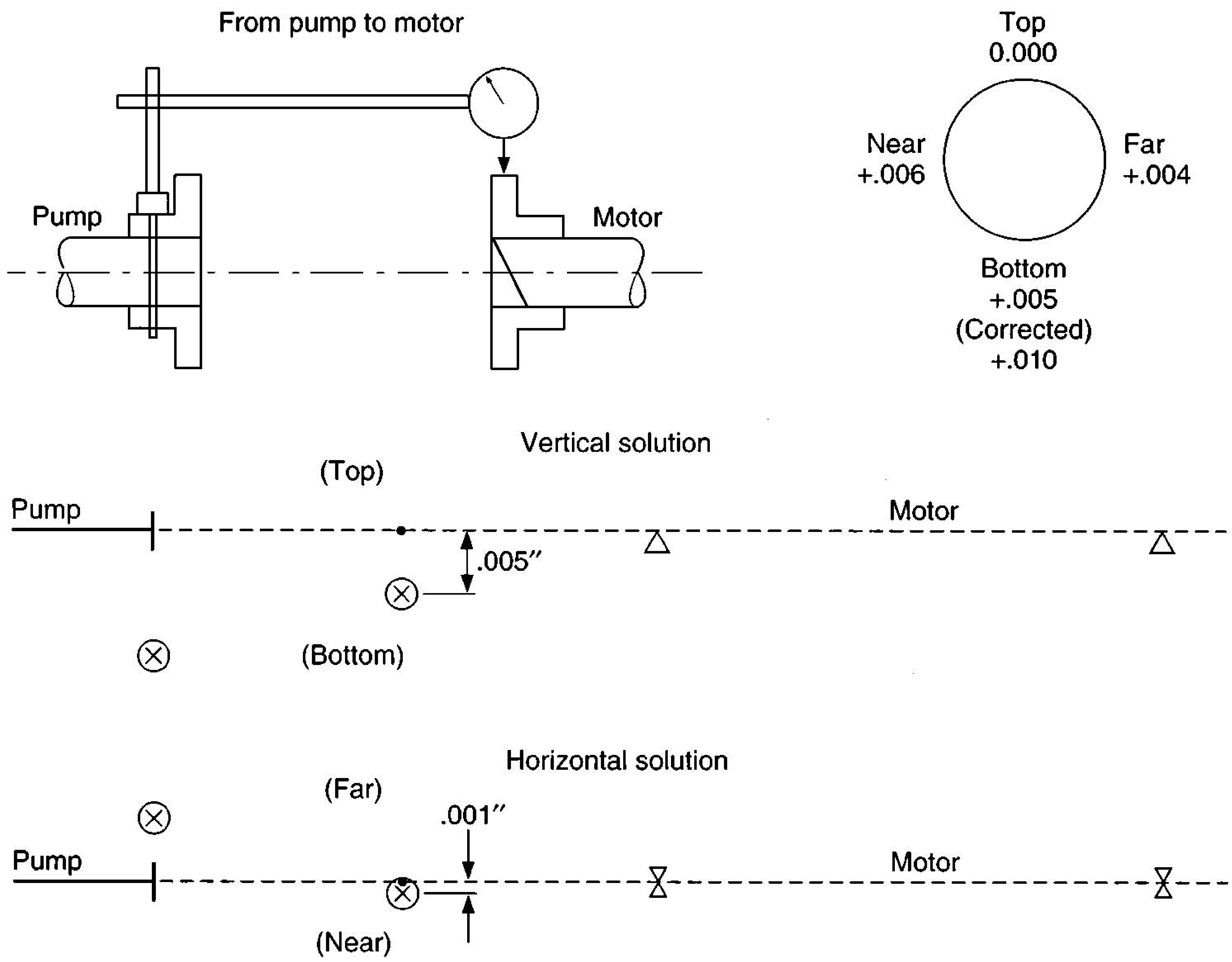

FIGURE 3.25

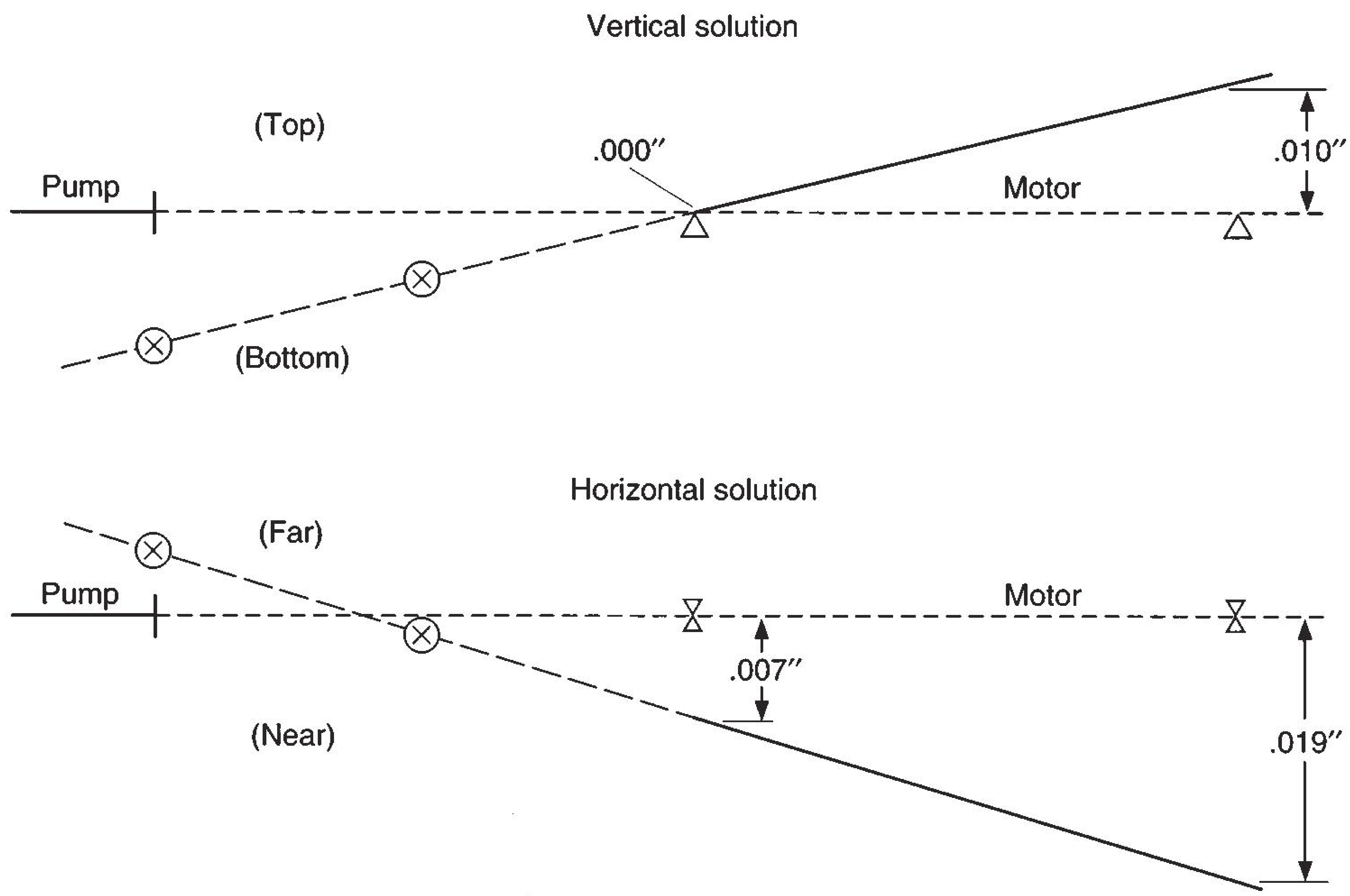

FIGURE 3.26

As the graphic plot shows, the motor must be moved 0.007 in. toward the "far" side at the front foot and 0.019 in. toward the "far" side at the back foot. This solves the horizontal alignment provided there is no side-to-side movement caused by operating conditions.

Where this graphic procedure really has its greatest value is when there are more than two units in one train to be aligned. As shown in Fig. 3.27, there are three pieces of equipment to be aligned.

If you start by aligning the second unit to the first unit and then the third unit to the second unit, the third unit may have to be moved a considerable distance. In the actual installation, there may not be room to move that third piece of equipment the required amount. For example, there may not be enough shims under the third unit to lower it, or there may not be clearance in the bolt holes to move it sideways. The approach that should be taken is shown in Fig. 3.27. Plot all three units on one piece of graph paper after taking reverse indicator alignment readings across both couplings. As can be seen, the third unit is a long way off the original reference line. By drawing an alternate reference line close to the actual position of the three units, it can be seen that minimal movement is required to get all the units in alignment with respect to the alternate reference line.

Alignment problems can be made easier. There are a lot of tools on the market to make the mathematics simple; however, a graphic picture shows the whole problem at a glance and makes the solution apparent. The alignment chart in Fig. 3.28 may be helpful.

Face/Rim

To explain the face/rim alignment procedure, a motor-to-pump example will be used. First, correct the vertical misalignment by shimming, and then correct the horizontal misalignment by sliding the equipment from side to side. With proficiency, these two steps can be done together.

Before starting the alignment work, determine which piece of equipment is easiest to move. This is not to eliminate the option of moving both units if a problem occurs. The pump, in this example, will be fixed. The motor will be moved into alignment with the pump.

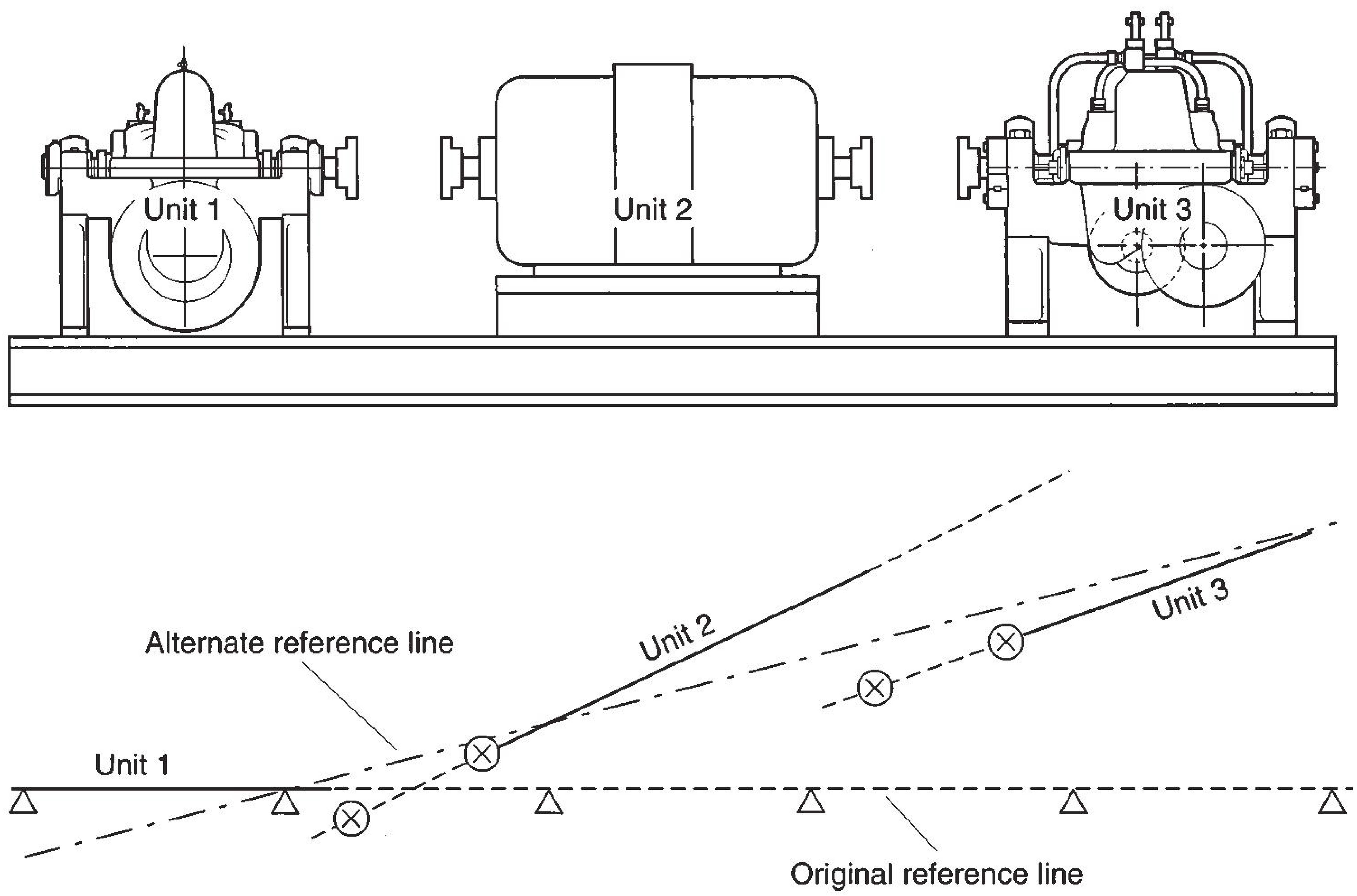

FIGURE 3.27 Multiunit alignment.

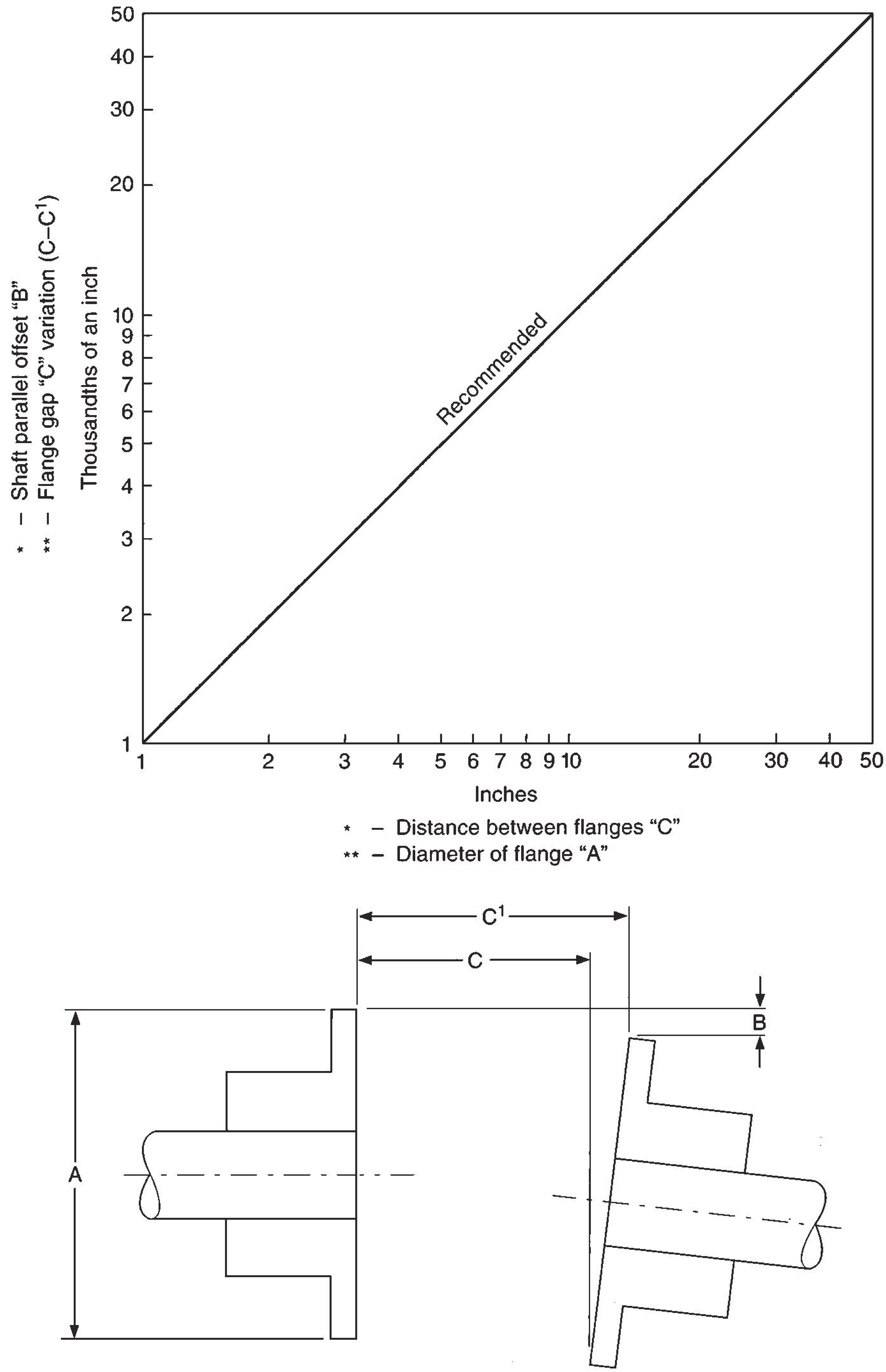

FIGURE 3.28 Alignment chart.

Vertical Alignment. Now, on a sheet of graph paper, lay out the equipment being aligned. See Fig. 3.29.

The horizontal scale on the graph used here is one small division equals 1 in. The distances needed are

1. Distance from where the indicator rides radially on the pump hub to the center of the motor front feet. In the example, this is 15 in.
2. Diameter of the pump hub flange at the location the face indicator rides. In the example, this is 10 in.
3. Distance from the center of the motor front feet to the center of the motor back feet. In the example, this is 25 in.

The alignment can be done either with the coupling totally installed or with just the coupling hubs mounted. Find a spot to mount the dial indicator bracket. The shaft behind the hub or the hub itself is good. A chain-clamp alignment bracket usually fits in well.

Angular (Face) Solution. With the indicator bracket attached to the motor hub, reading off the pump hub face, rotate the shafts in 90° increments and take readings (see Fig. 3.30).

Note: Make sure that neither of the equipment shafts being aligned moves axially, since this will distort the face readings.

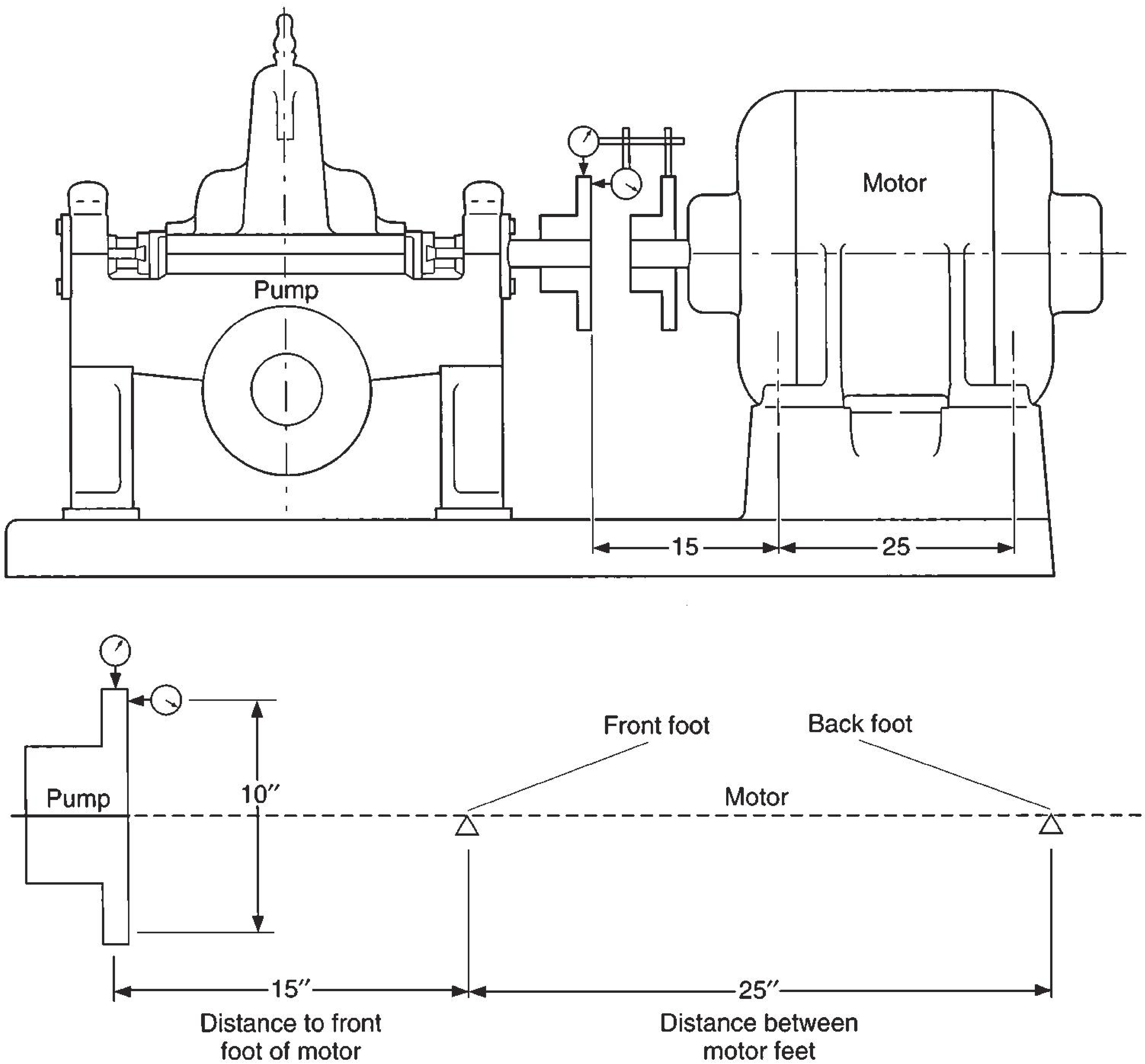

FIGURE 3.29

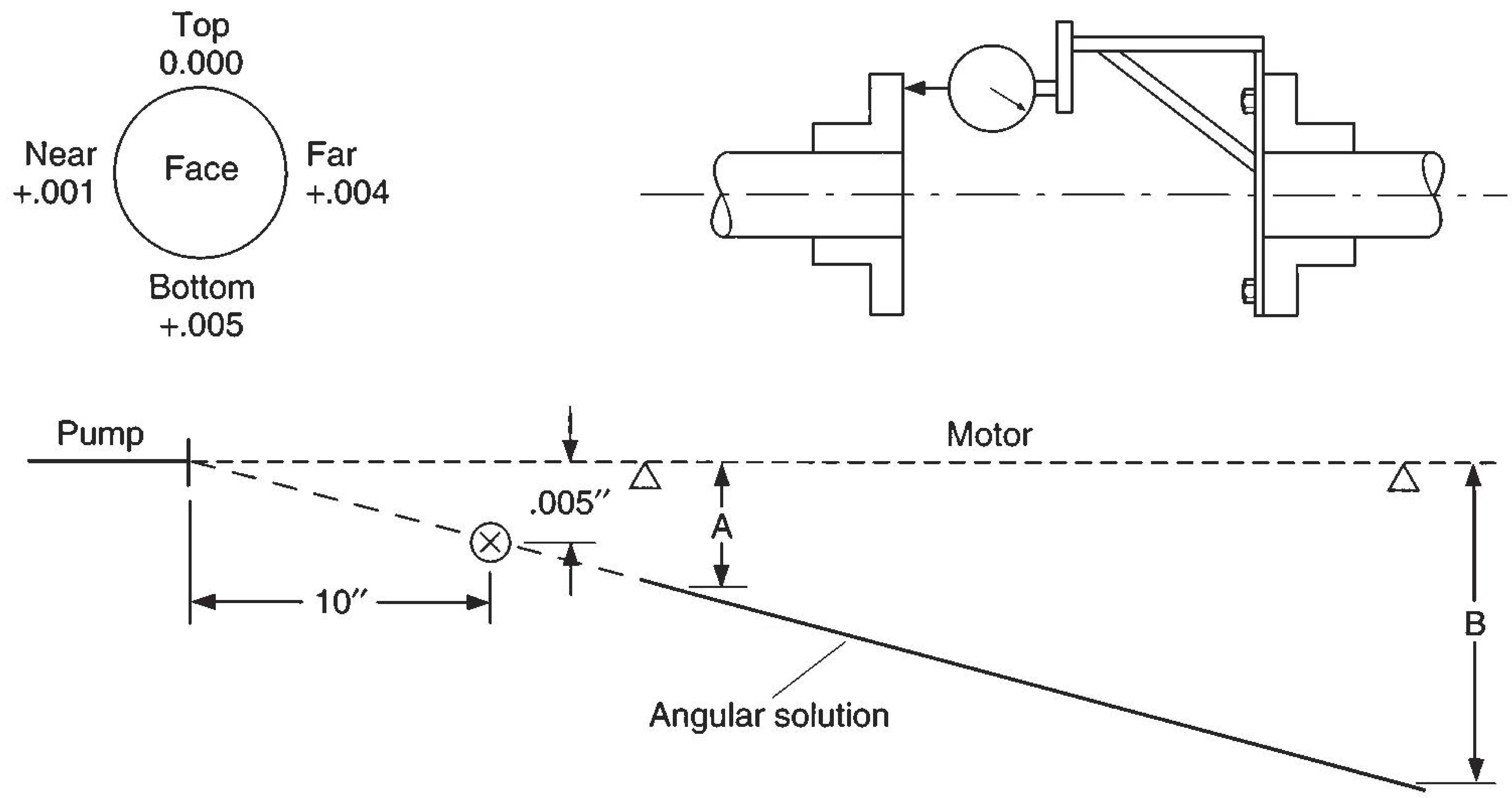

FIGURE 3.30

Reading on the face at a 10-in. diameter and getting +0.005 at the bottom, we know that the motor shaft is off 0.005 in. for every 10 in. of length. The plus reading at the bottom means that the indicator was compressed. This can only happen when the motor shaft is low compared with the pump center line extension. This can be shown graphically; use the pump flange center as a pivot point. Extend the 10 in. (diameter of hub flange) along the graph (pump center line). Using a vertical scale of one small division on the graph equals 0.001 in., plot the 0.005 in. as shown in the example. See Fig. 3.30. Draw a line from the center of the pump flange face through the 0.005-in. point and extend it past the motor feet.

Note: If the vertical scale chosen is too big, it may realize some minor shimming errors.

The (face) angular vertical misalignment could now be corrected. From the graph, the solution is to add 0.0075-in. shims to the front foot (*A*) and add 0.020-in. shims to the back foot (*B*). Since it is easier to make one shim change instead of two, solve the parallel offset (rim) before shimming. Then add the two results together and make one move.

Parallel-Offset (Rim) Solution. Now, with the indicator bracket attached to the motor hub, reading off the pump hub outside diameter, rotate the unit in 90° increments and take readings. See Fig. 3.31.

Bottom reading is then corrected for indicator sag. Indicator sag in the example was determined to be −0.005. The −0.005 was subtracted from the +0.010 indicator reading to give an actual +0.015 reading, or +0.010 − (−0.005) = +0.015.

Since this is a total indicator reading (T.I.R.), it is two times the actual shaft-to-shaft relation. Thus +0.015/2, or +0.0075, is used to show where the motor center line extension is relative to the pump shaft center line at the pump hub. With a plus reading at the bottom, it indicates that the motor shaft is high compared with the pump. Using a scale of one small division on the graph equal 0.001 in., plot this point as shown in the example. The parallel-offset (rim) misalignment alone could be corrected by removing 0.0075 in. of shims from under both front and back feet.

Total Vertical Alignment Solution. By drawing a line parallel to the angular (face) solution and through the parallel-offset (rim) point, the total solution can be read off the graph at *C* and *D.* In this example *C* = 0 and *D* = 0.0125 in. Add 0.0125-in. shims to back foot. See Fig. 3.31.

If there are any thermal growth considerations, they should be added or subtracted to the results before shim change. In the example, there are none.

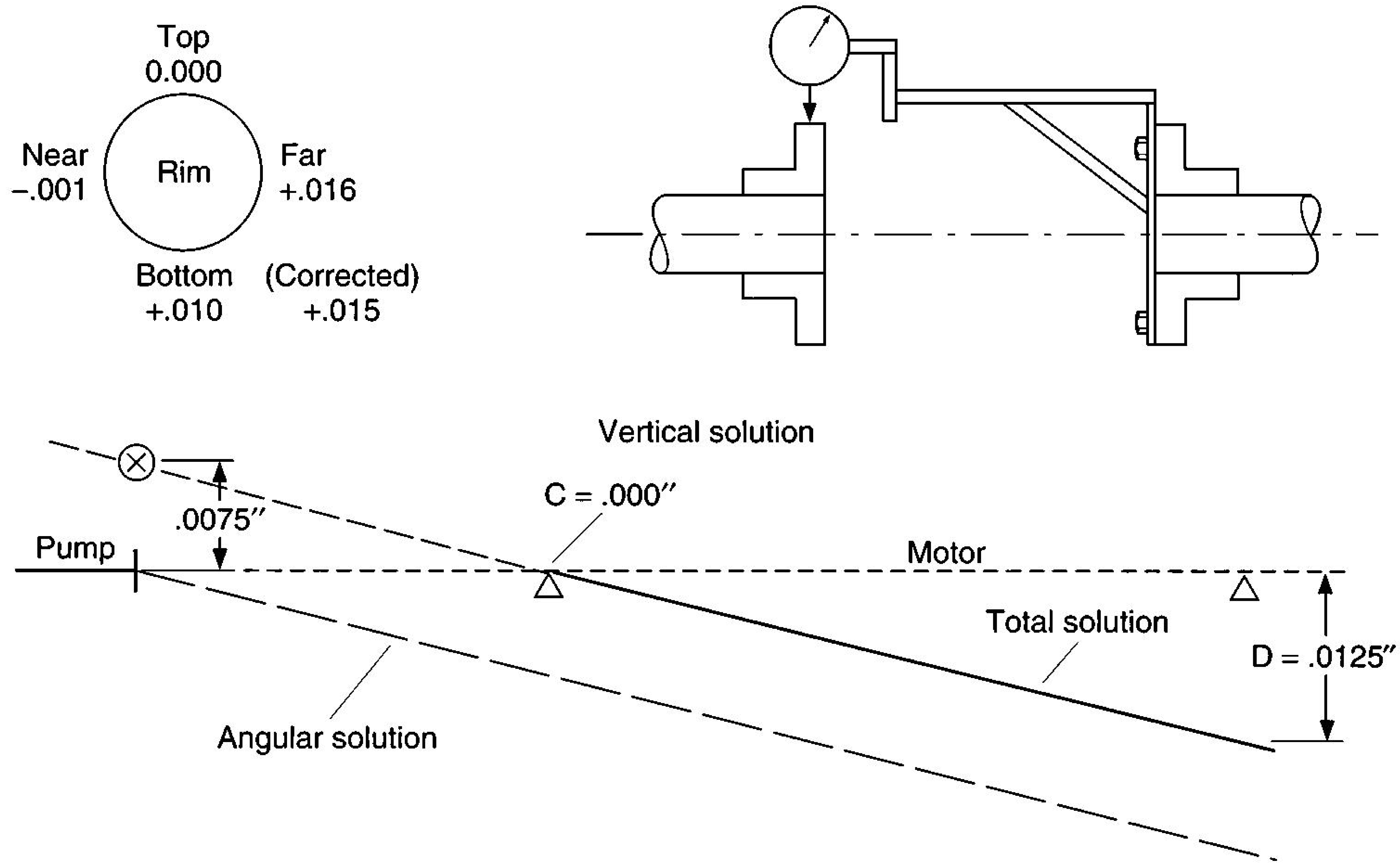

FIGURE 3.31

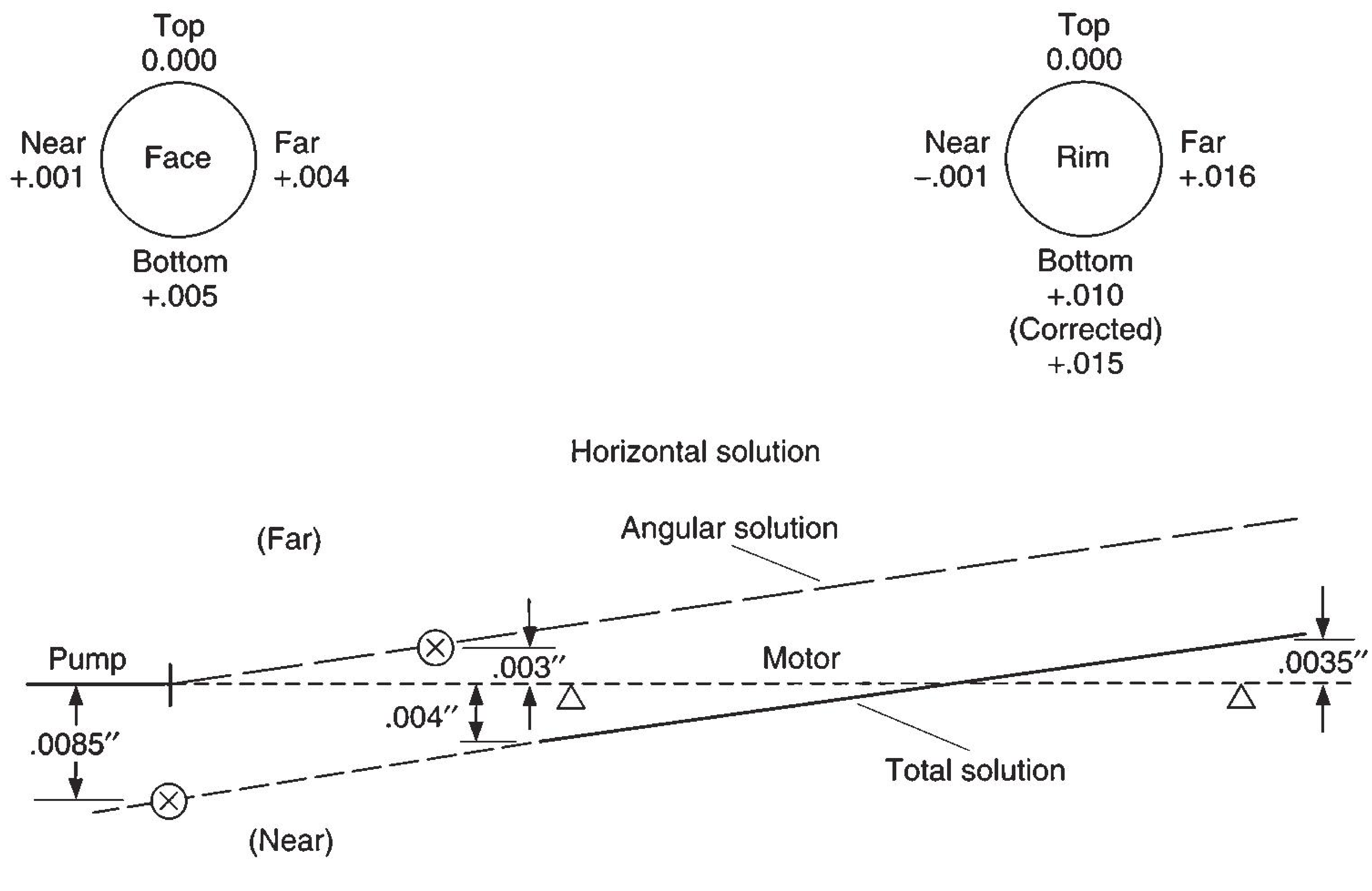

FIGURE 3.32

Total Horizontal Alignment Solution. For the horizontal (side-to-side) results, the same procedure is used. Algebraically subtract the side-to-side readings. Indicator sag can be ignored because it cancels out. Plot these readings, and the results can be read off the graph. See Fig. 3.32.

The solution for this example is: At the front motor feet, push the motor away from you by 0.004 in., and at the back motor feet, pull the motor toward you by 0.0035 in. The alignment chart in Fig. 3.28 may be helpful.

Across-the-Flex-Element

When the distance between the disk packs is long, and where it is not practical to try to span the distance with indicator bracketry, the across-the-flex-element method can be used.

To explain the across-the-flex-element alignment procedure, a motor to a right-angle gear example will be used. First, correct the vertical misalignment by shimming, and then correct the horizontal misalignment by sliding the equipment from side-to-side. With proficiency, these two steps can be done together.

Before starting the alignment, determine which piece of equipment is easiest to move. This is not to eliminate the option of moving both units if a problem occurs. The gearbox, in this example, will be fixed. The motor will be moved into alignment with the gearbox.

Vertical Alignment Solution. Now, on a sheet of graph paper, lay out the equipment being aligned. See Fig. 3.33.

You should always use a scale that is convenient to the size of the graph paper. The horizontal scale on the graph used here is one small division equals 1 in. The distances that are critical are

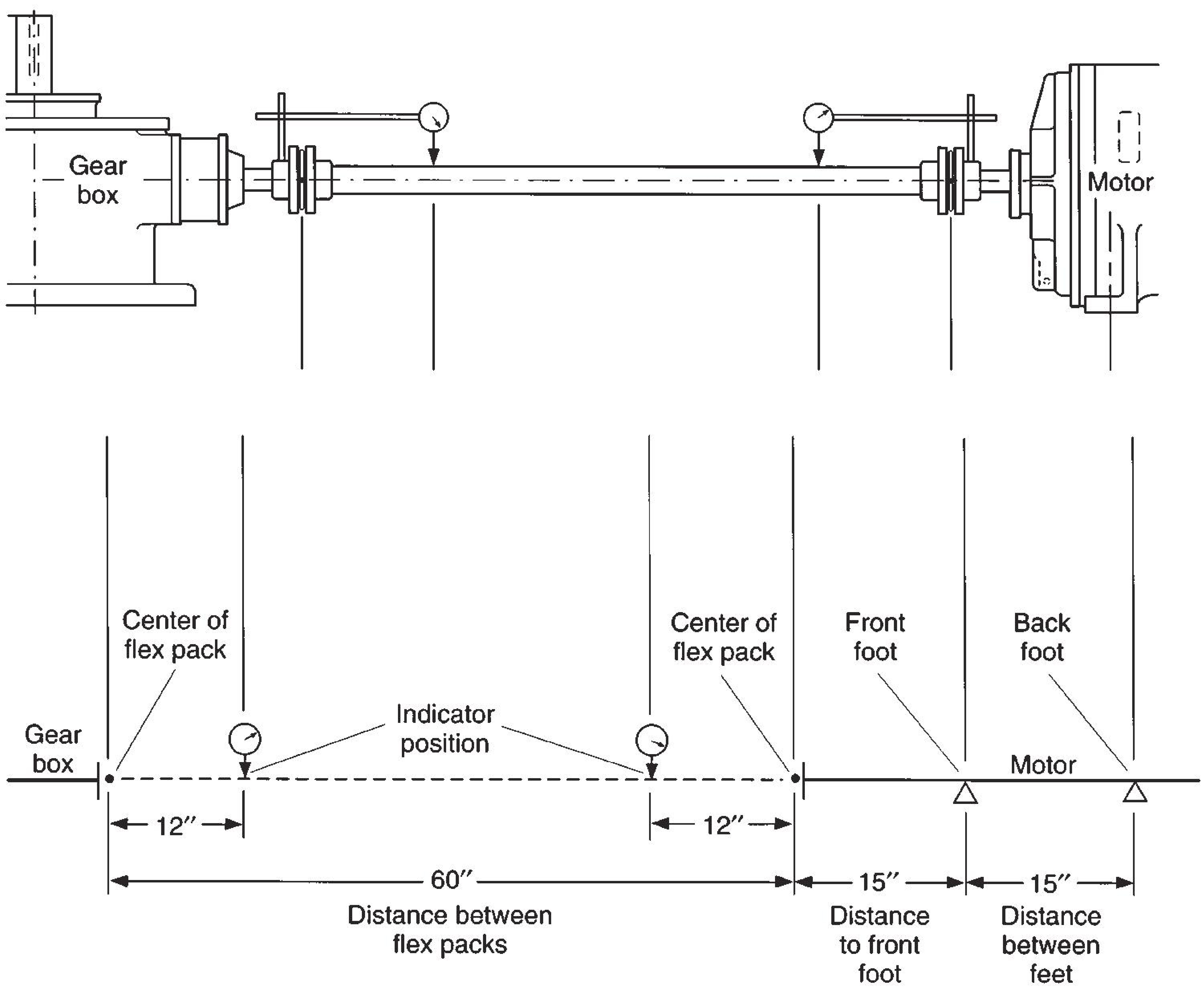

FIGURE 3.33 Across-the-flex alignment.

1. Distance from the center line of one flex pack to the center line of the other flex pack. In the example, it is 60 in.
2. Distance from the center line of the motor flex pack to the center of front motor foot. In this example, it is 15 in.
3. Distance from the center of the motor front feet to the center of the motor back feet. In this example, it is 15 in.
4. Distance from the flex pack to the dial indicator on center member. In this example, the distance is 12 in.

The alignment can be done only with the coupling totally installed. Find a spot to mount the dial indicator bracket. The shaft behind the hub itself is good. A chain-clamp alignment bracket usually fits in well.

With the indicator bracket attached to the gearbox hub, reading out on the center member a convenient distance (in the example 12 in. was used), rotate the unit in 90° increments and take readings. See Fig. 3.34.

Bottom reading is then corrected for indicator sag. The indicator sag in the example was determined to be −0.004 in. The −0.004 was subtracted from the +0.016 indicator reading to give an actual reading of +0.020, or +0.016 − (−0.004) = +0.020.

Since this is a T.I.R., it is two times the actual center member center line location relative to the pump shaft extension of +0.020/2 = +0.010 (what we are trying to do here is to determine the angle the center member makes with respect of the gearbox shaft).

A plus reading at the bottom indicates that the center member tips down as it extends away from the gearbox. Using a scale of one small division on the graph equals 0.002 in., plot the 0.010 as shown in the example. See Fig. 3.33.

Now with the indicator bracket attached to the motor hub reading out on the center member, rotate the unit in 90° increments and take readings. See Fig. 3.35.

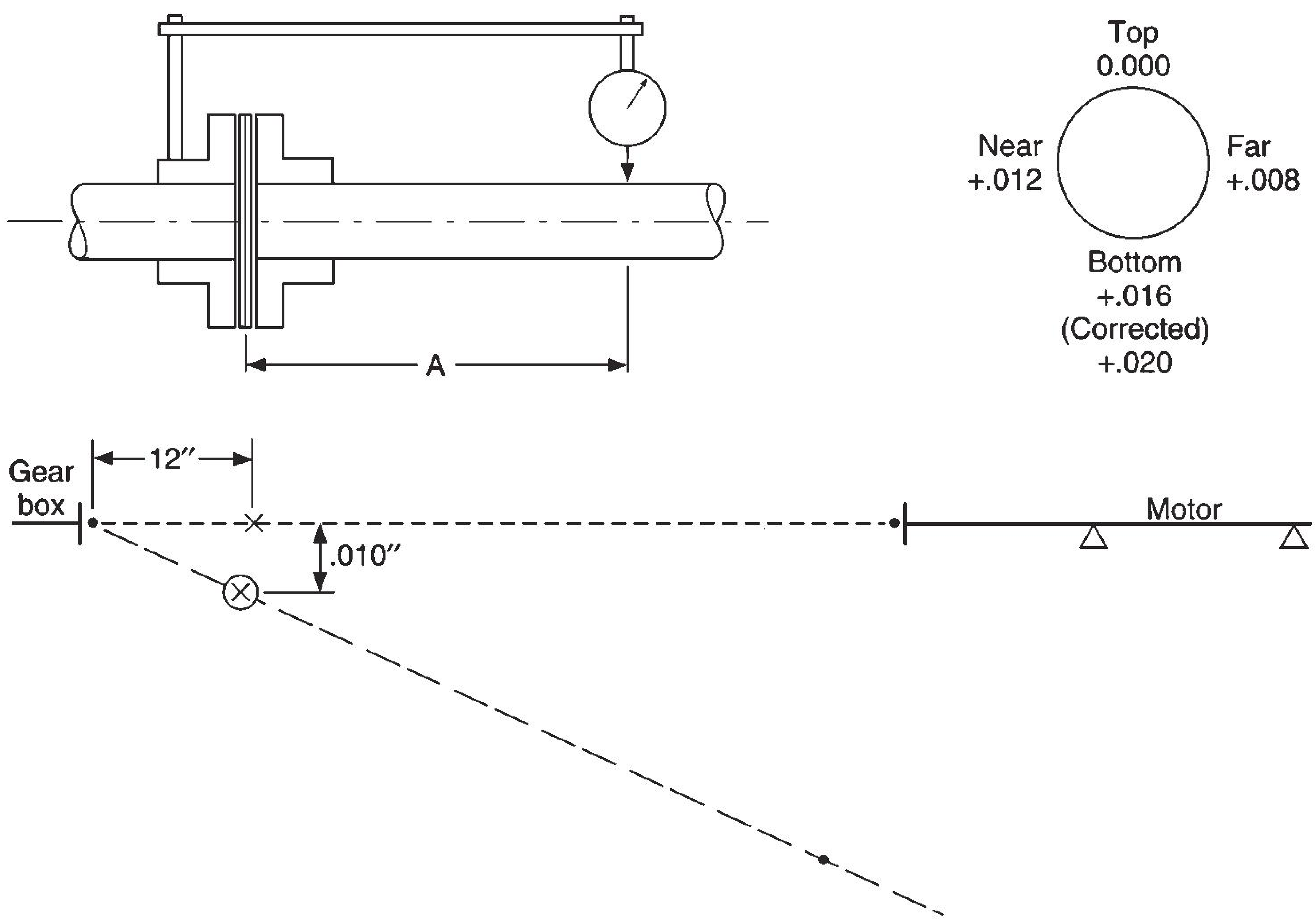

FIGURE 3.34

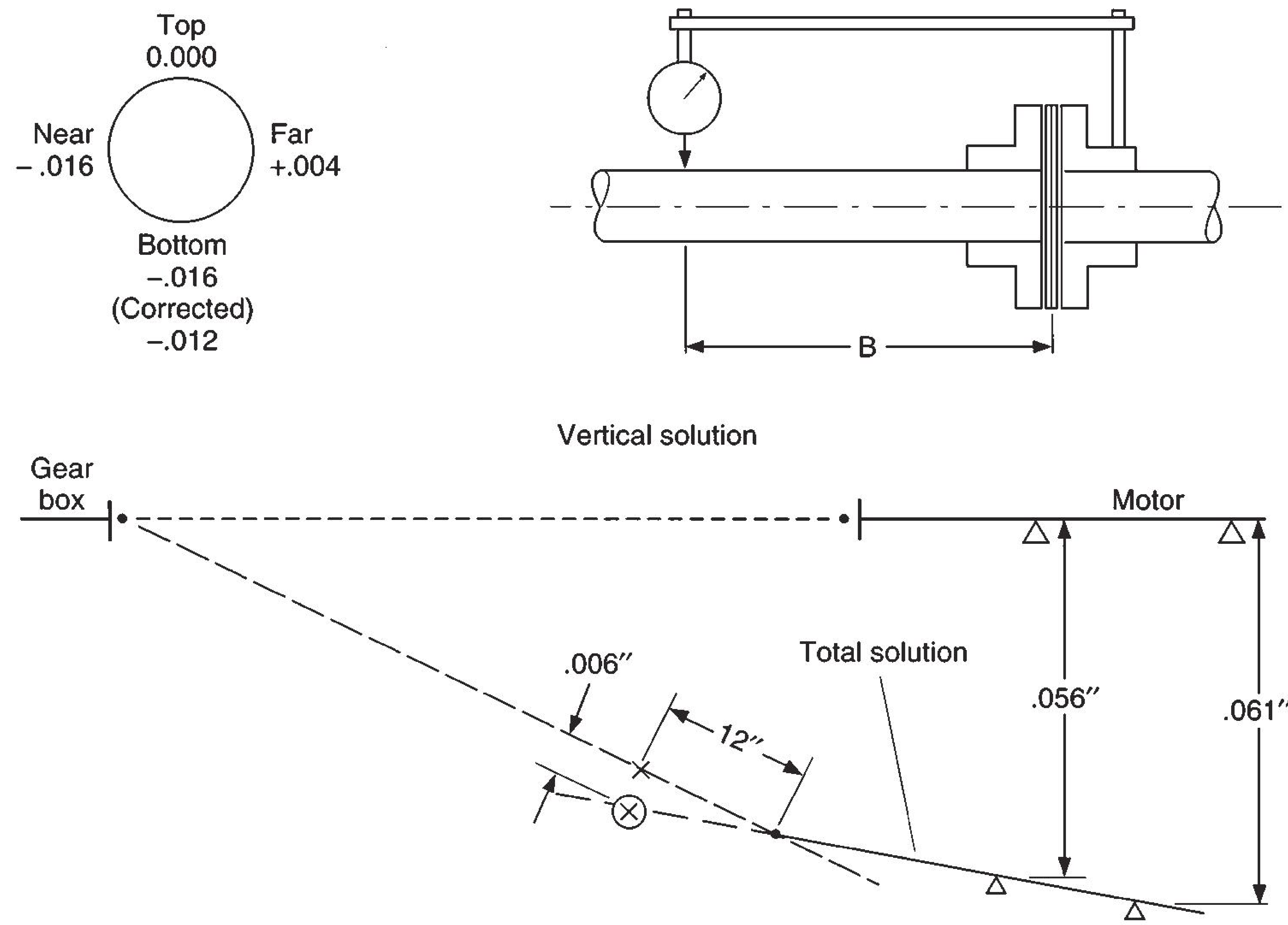

FIGURE 3.35

Bottom reading is corrected for indicator sag: −0.016 − (−0.004) = −0.012. This is a T.I.R., so the actual figure is +0.006 (what we are trying to do here is determine the angle the center member makes with respect to the motor shaft).

The minus reading on the bottom indicates that the center member tips up as it extends away from the motor. Using a scale of one small division on the graph equals 0.002 in., plot the 0.006 in. as shown in the example.

The motor shaft can now be drawn in because two points along it have been defined: (1) center of the flex element and (2) the point just plotted 0.006 in. below the center member. The shimming requirements can now be read off the plot where the motor shaft intersects the planes of the motor feet.

Note: If the vertical scale chosen is too big, it may realize some minor shimming errors.

In this example, the motor should be shimmed up 0.056 in. under the front feet and shimmed up 0.061 in. under the back feet.

Horizontal Alignment Solution. For the horizontal (side-to-side) results, the same procedure is used. Algebraically subtract the side-to-side readings. Indicator sag can be ignored because it cancels out. Plot these readings, and the results can be read off the graph. See Fig. 3.36.

The solution for this example is: At the front motor feet, do not move the motor, and at the back motor feet, pull the motor toward you by 0.010 in. See Fig. 3.36. The alignment chart in Fig. 3.28 may be helpful.

The varying coupling configurations and the varying conditions of the drives make it impossible to establish fixed rules of alignment at installation. Normally, the coupling manufacturer's recommendations should be followed.

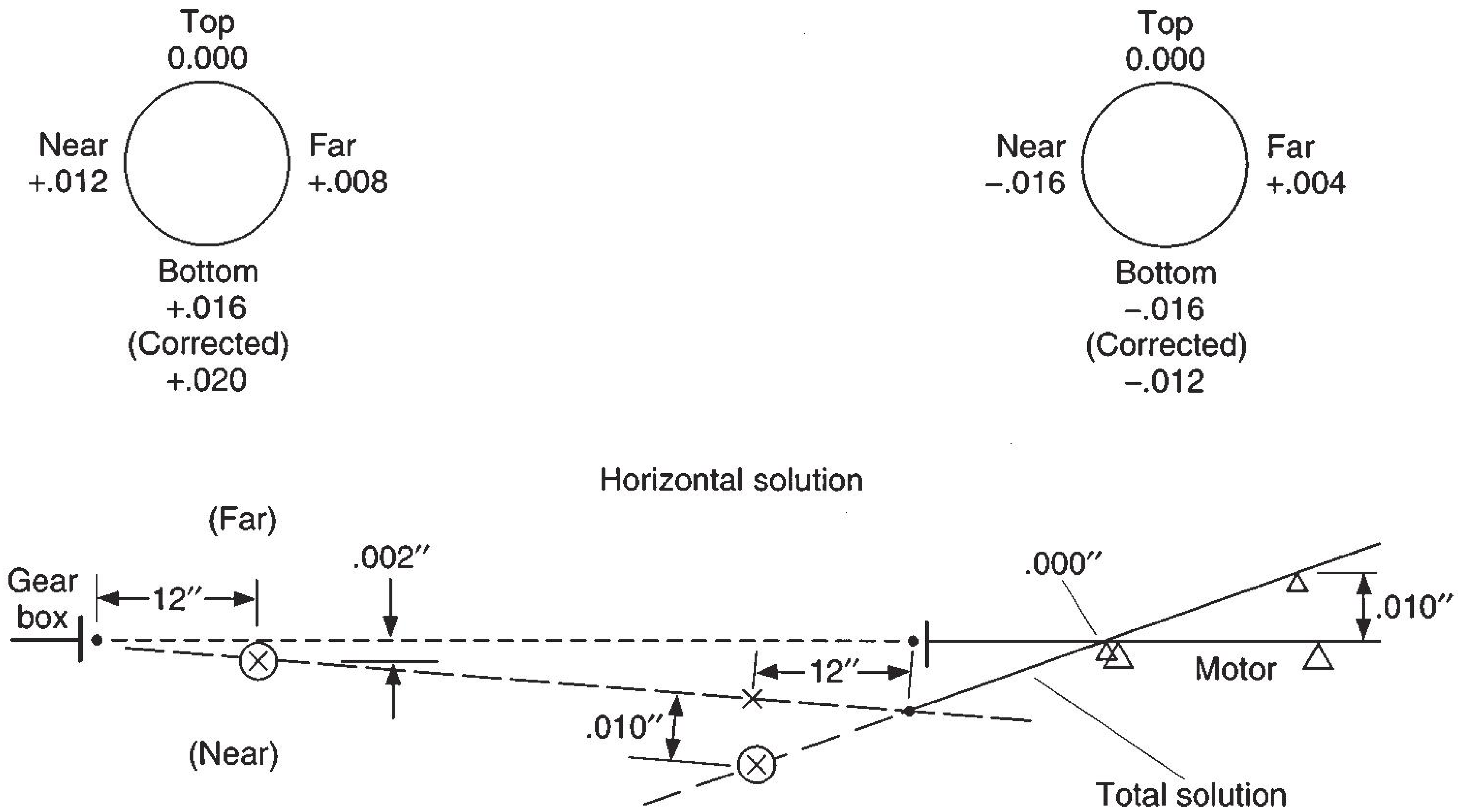

FIGURE 3.36

It must be remembered that all the preceding comments have been relative to cold equipment alignment. Most equipment will shift to some extent as it comes up to operating temperatures. If the movements are known to be slight, no adjustments are necessary. If the movements are unknown or suspected of being significant, a hot check of the alignment should be made. If corrections are necessary, the previously outlined procedures should be used.

Long-span couplings are particularly difficult to align. There are known cases where the spans were in excess of 30 ft. The narrow width of the flanged hubs precludes the use of a straightedge. It is again recommended that dial indicators be used and that they be rigidly mounted. The procedure outlined for angular misalignment applies. When measurements are taken and corrections made at both end joints, the system is in alignment.

Three bearing drives require the use of a single engagement coupling. In that they are normally required to support a heavy radial load, these couplings are capable of accepting only angular misalignment. Normal double-flexing couplings cannot be used. Typical examples are engine to single-bearing generators, belt drives, and compressors driven by a single-bearing motor. The procedures outline for angular misalignment under "Face/Rim" apply.

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CHAPTER 4
CHAINS FOR POWER TRANSMISSION

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This section on precision chain drives places before the plant engineer general information and instructions for their installation, lubrication, and maintenance. It will aid him in advising his maintenance departments on how to obtain the best results from chain drives.

Chain drives consist of an endless series of chain links which mesh with toothed wheels, called *sprockets*. The sprockets are keyed to the shafts of the driving and driven mechanisms.

A roller chain has two kinds of links—roller links and pin links—alternately assembled throughout the chain length. A roller link consists of two sets of hollow rollers and bushings, the bushings being press-fitted into the apertures in the roller link plates, the rollers being free to rotate on the outside of the bushing. The pin link has two pins press-fitted into the apertures of the pin-link plates.

When the chain is assembled, the two pins of the pin links fit within the cylindrical bushings of the two adjacent roller links. The pins oscillate inside the bushings, while the rollers turn on the outside of the bushings. This latter action eliminates rubbing of the rollers on the sprocket teeth.

Roller chain is identified by three principal dimensions: pitch, width, and roller diameter (Fig. 4.1).

The term *silent chain* has been generally adopted to describe the inverted-tooth-link type of chain. This type of precision chain is a series of toothed links alternately laced on pins or a combination of joint components in a manner permitting joint articulation between adjoining pitches (Fig. 4.2).

The ends of the toothed links engage the faces of alternate sprocket teeth. The center sections of the links are recessed to provide clearance so that the links straddle one tooth and engage the adjacent teeth (Fig. 4.2).

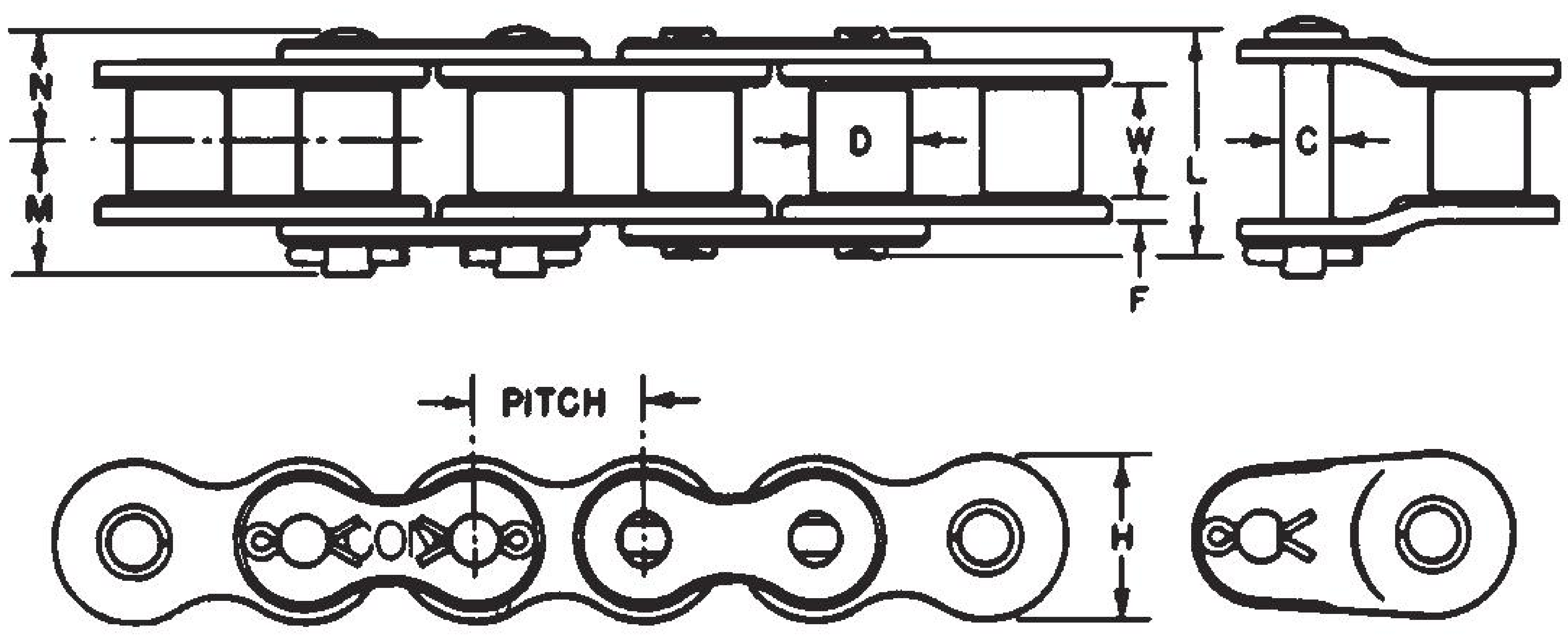

FIGURE 4.1 Dimensions for roller-chain identification.

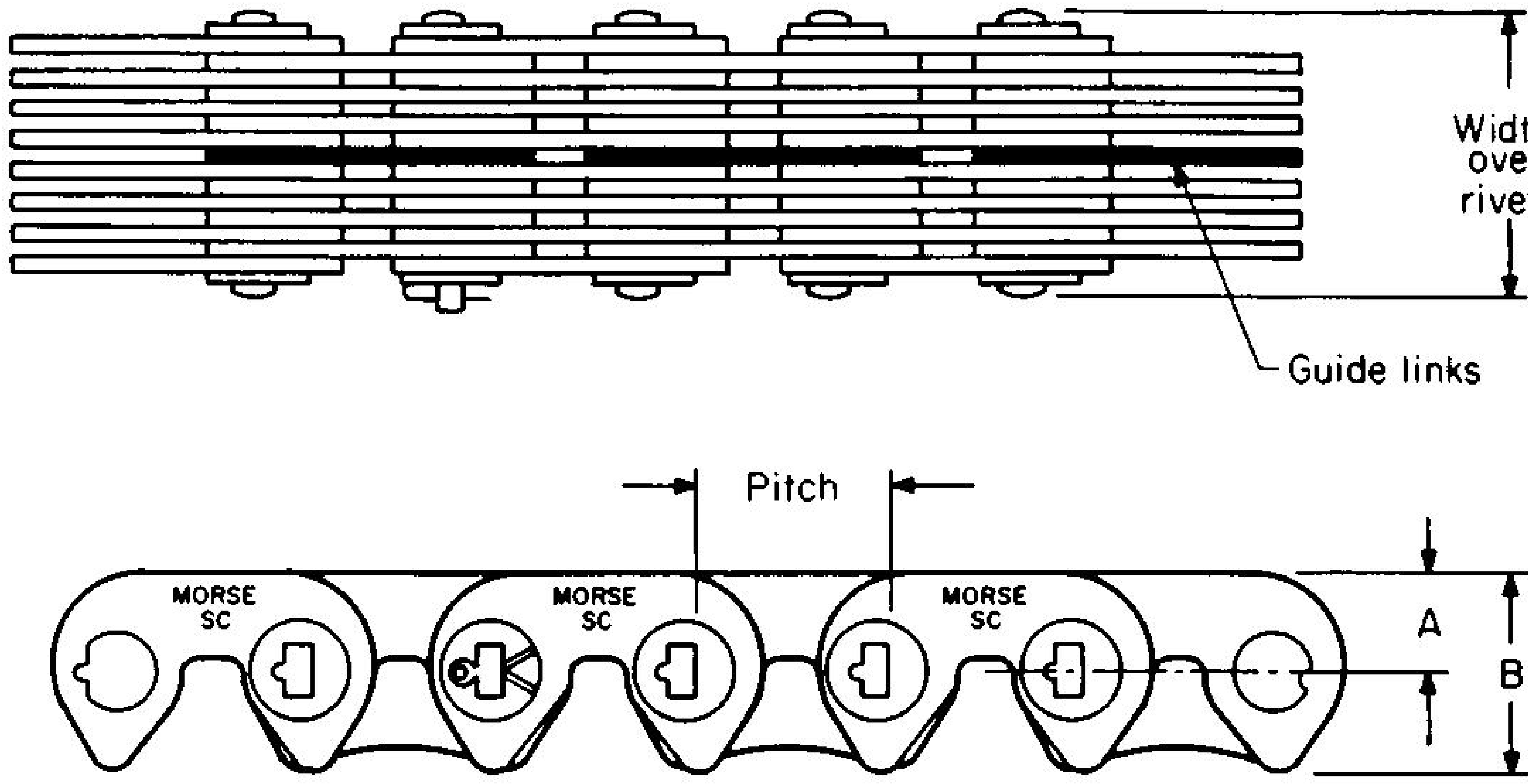

FIGURE 4.2 Silent-chain construction.

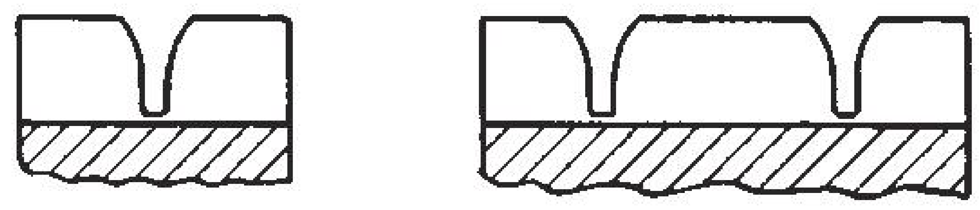

FIGURE 4.3 Silent-chain sprocket guide grooves.

The chain is retained on the sprockets by means of guide links (not recessed) assembled in the chain. The guide links track in a groove cut in the sprocket teeth (Fig. 4.3).

The use of precision chain drives has steadily increased during the past quarter century through their constantly expanding field of application. This is owing primarily to the following advantages:

1. *Drive efficiency:* This is normally in excess of 98 percent.
2. *Uniform driven speed:* Roller and silent chain drives are positive; the principle of teeth, not tension, results in no loss in rotative speeds through slippage or creep.
3. *Low bearing loads:* Slack side tension is not required.
4. *Larger ratios:* Less wrap on driver sprocket is required, which permits a higher speed ratio in given area than can be obtained from belt drives.
5. *More power per inch of width:* Strength of steel permits greater loads for any given diameter and speed.
6. *Relatively unrestricted center distances:* Chain can be made endless to any length within limits.
7. *Ease of installation:* Center distances and alignment do not require close tolerances.
8. *Standardization:* Industry standardization of chain and sprockets means that replacements are available from many sources.
9. *Repair on the job:* Repair links are available for quick emergency replacement of worn or damaged links.
10. *Drive multiple shafts:* Chain is one of the most convenient methods of driving several shafts from one power source.
11. *Long drive life:* Wear is reduced through distribution of load over a number of sprocket teeth. Normal chain wear is a slow process and therefore requires infrequent adjustment.
12. *No deterioration:* Adequately lubricated chains do not deteriorate with age, nor are they adversely affected by sun, oil, and grease.

No one type of chain is ideal for all kinds of service. Certain chain drives are most efficient at very low speeds, others at intermediate speeds, and still others are capable of fairly high-speed operation.

Chain speed, quietness of operation, service life, freedom from maintenance, and the relative first cost will vary within limits according to the combination of chain and sprockets selected. Too much emphasis should not be placed on first cost. Low cost at the expense of other requirements is false economy. Usually it is a matter of compromise to arrive at the best possible combination of specifications to fulfill the requirements of any one drive. Therefore, it is well to evaluate the relative importance of various drive requirements.

Most chain manufacturers' catalogs contain adequate design information on the method of selecting a chain drive. Manufacturers will offer design suggestions for particular applications if they are furnished adequate information.

Basic data needed for the correct design of chain drives:

1. Horsepower and rpm or torque and rpm to be transmitted and whether rpm is exact or approximate
2. Center distance fixed or adjustable and, if adjustable, the amount of adjustment to be provided
3. Type of driver (electric motor, gasoline engine, diesel engine, torque convertor, jack shaft, etc.)
4. Type of driven unit (machine tool, hoist, conveyor, agricultural equipment, construction equipment, timing and motion control, lift hoist, tension member, etc.)
5. Service-continuous or intermittent-average number of hours per day
6. Type of load-smooth and uniform, load reversals, moderate shock or heavy shock
7. If speed is variable, the maximum, minimum, and usual speeds; the horsepower or torque to be transmitted at each speed; and the approximate percentage of operating time at each speed
8. Shaft diameters and lengths, keyways and setscrew dimensions
9. Available space dimensions
10. Approximate position of the drive (horizontal, vertical, etc.) and the direction of rotation
11. Operating conditions (wet, dusty, etc.)
12. Lubrication-whether drive will be encased or otherwise adequately lubricated

Horsepower-rating tables appearing in chain manufacturers' catalogs are based on an average life expectancy of approximately 15,000 service hours under optimal drive conditions and a service factor of 1.

SERVICE FACTORS

Horsepower ratings for roller chains must be modified according to the type of load induced by the driver and driven equipment. It is impossible to give service factors by which rated capacities must be multiplied without knowing the type of driver and driven equipment. However, the most prevalent conditions and their accompanying service factors are given in Table 4.1. Conditions which may require a modifying or service factor to be applied to the horsepower ratings are given in the following list.

Favorable service conditions which will contribute to chain life:

1. Drive for intermittent or for standby service
2. Less than maximum service life required

TABLE 4.1 Service Factors

Type of load	Internal-combustion-engine hydraulic drive	Electric motor or turbine	Internal-combustion-engine mechanical drive
Smooth	1.0	1.0	1.2
Moderate shock	1.2	1.3	1.4
Heavy shock	1.4	1.5	1.7

3. Slow speeds, smooth steady load
4. Low ratios permitting larger number of teeth in the sprockets
5. Long centers, on adjustable center distance drives
6. Exceptionally good lubrication

Unfavorable service conditions which restrict chain life:

1. Small sprocket having fewer teeth than recommended by chain manufacturers
2. Unusually large sprockets
3. Inpulse, load reversals, or shock loading
4. Three or more sprockets in the drive
5. Long centers, on fixed center distance drives
6. Poor lubrication
7. Dirty or dusty conditions

Impulse or shock loading should not be confused with high starting or momentary overloads. Because of the high factors of safety with respect to tensile strength, high starting loads or peak loads of short duration do not necessarily require an increase in horsepower rating.

Service factors as given in Table 4.1 should be used in connection with the horsepower ratings, the load being multiplied by the factor to obtain the required chain capacity.

Normal chain wear is caused by the flexing of the chain joints in both roller and silent chains. Wear in the chain joints is usually the limiting factor in the life of a chain. Such wear results in chain elongation, or in other words, the chain pitch is increased. This increase in pitch permits the chain to ride out on the sprocket teeth, which are usually designed to permit moderate pitch elongation. When excessive pitch elongation occurs, the chain must be replaced before it overrides the sprocket teeth.

SPECIAL INVERTED TOOTH CHAINS

HV, a typical version of an inverted-tooth chain designed expressly for high-horsepower and high-speed application where service is severe, utilizes a unique tooth form on the sprocket and a modified chain-link profile, materially reducing the effect of chordal action and linear pulsation while substantially increasing chain endurance strength.

The chain assembly is similar to silent chain, with inverted tooth links laced in alternate sections across the width of the chain, and assembled with two pins having the same sectional geometry, one called the *pin* and the other the *rocker.* This forms the joint which articulates between the joining links.

Chordal action is a serious limiting factor in roller-chain performance. It may be described as the vibratory motion caused by the rise and fall of the chain as it goes over a small sprocket. Figure 4.4 shows schematically a roller chain entering a sprocket (*a*); the line of approach is not tangent to the pitch circle. The chain makes contact below the tangency line, is then lifted to the tangent line (*b*), and then is dropped again (*c*) as sprocket rotation continues. Because of its fixed-pitch length, the pitch line of the link cuts across the chord between two pitch points on the sprocket and remains in this position relative to the sprocket until the chain disengages. This chordal action seriously detracts from chain performance and life.

1. There is a very definite surge of force in the chain caused by the acceleration and deceleration of the chain as it makes this chordal rise and fall.
2. When the chain enters the sprocket, the tooth gap into which the joint is to fall is rising while the chain strand is falling. Therefore, at contact, there is a definite impact. This impact is very much aggravated by any increase in velocity.

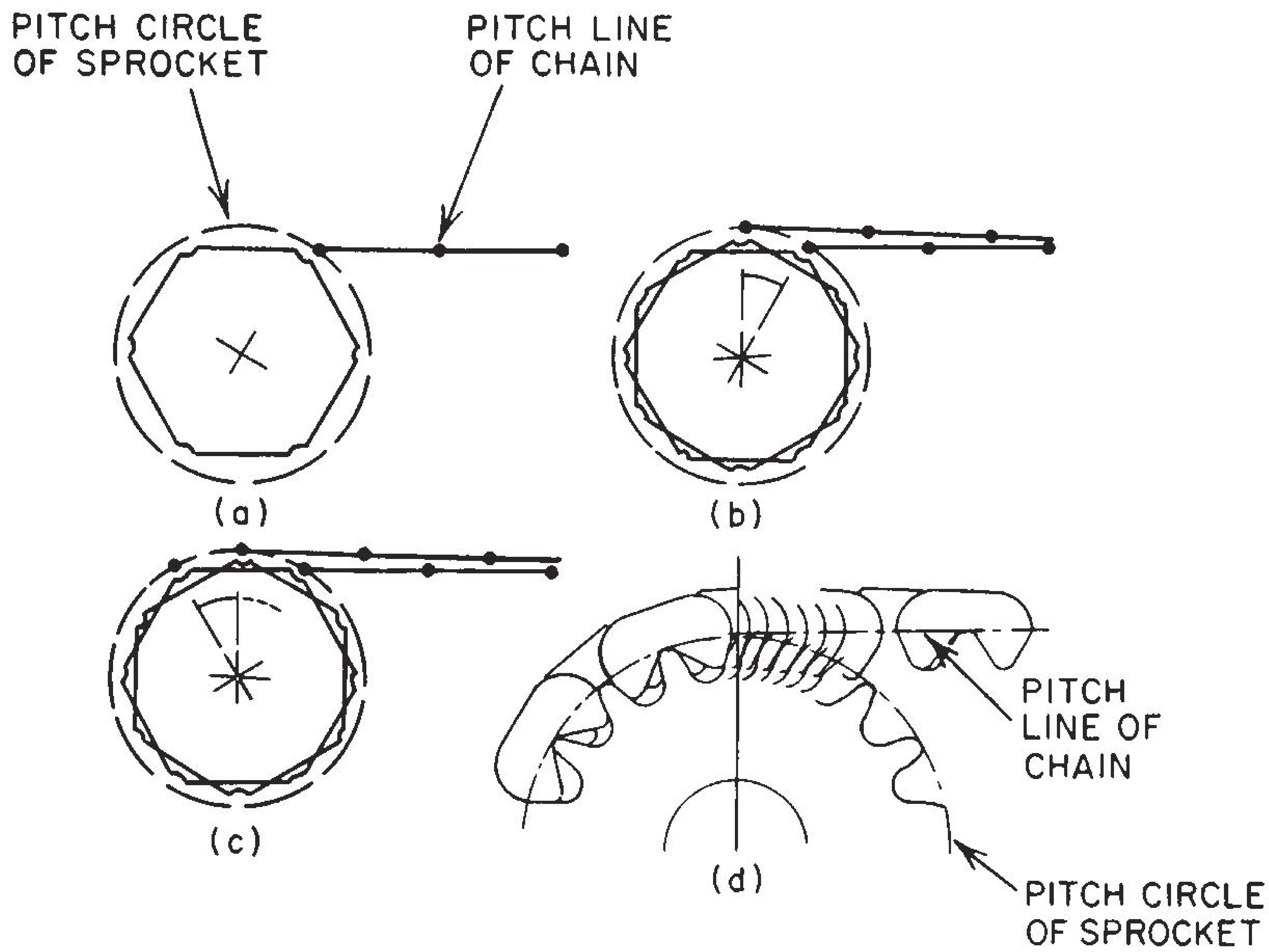

FIGURE 4.4 Chordal action.

Chordal action, therefore, not only produces pulsations in the chain and generates noise and vibration but also, because of all these things, considerably curtails the power-transmitting capacity and speed range of a roller chain.

The HV drive is designed to minimize chordal action. Smooth engagement with the sprocket minimizes shock loading and stresses in the links as well as noise, vibration, and heating. Figure 4.4*d* shows how the chain meets the sprocket. It enters approximately tangent to the pitch circle and maintains this position as it travels around the sprocket. This is made possible by two design features: (1) pitch elongation produced by the compensating joint action and (2) mating contours of the sprocket's involute-tooth form and the chain links.

The compensating joint is so designed that as flexure takes place the pitch of the chain actually elongates. The joint consists of a pin and rocker of identical cross section—the curved surfaces in contact with each other being tilted in such a manner that the contact point is below the pitch line of the chain. As the joint articulates, the contact point moves upward and the pitch of the chain elongates; the amount of pitch rise is near to that required for the chain to wrap the sprocket along the pitch circle. This is known as *chordal compensation.* The combination of involute-tooth sprocket design and the compensating joint ensures approximate tangential engagement of the chain into the sprocket for smooth and quiet operation.

INSTALLATION OF CHAIN DRIVES

A chain drive is essentially a flexible medium, and its installation is less difficult than many other forms of power transmission. However, care during installation will more than repay the time involved. Improper or careless installation will destroy the precision of any finely designed engineering system.

The shafts must be well supported by suitable and rigidly mounted bearings. Shafting, bearings, and foundations should be suitable to maintain the initial static alignment. Shaft displacement will destroy alignment and so shorten chain life. All shafts should be horizontal and parallel with each other.

Sprockets must be aligned axially on the shafts and secured against axial movement.

Proper chain tension is essential. Too tight a chain will cause excessive bearing loads. Too loose a chain will result in noisy operation and chain pulsations which will cause abnormal chain and sprocket wear.

Contact between the drive and surrounding objects must not be permitted. Ample clearance should be provided to allow for chain pulsations and for possible end float of the shafts. If loose material such as coal, dust, and gravel is present, sufficient clearance is essential to prevent accumulation around the drive.

Installation Procedure

Align each shaft with a machinist's level applied directly to the shafts. (Shafting with silent chain or multiple-width roller chain sprockets may be aligned by applying the level across the sprocket teeth.) Check shafts for parallelism with a feeler bar (Fig. 4.6). After adjusting for parallelism, recheck the shaft levels. Repeat these adjustments until both level and alignment are satisfactory.

Mount sprockets on shafting, and align by checking with a straightedge along the finished sides of the sprockets (Fig. 4.7). A taut wire may be used if the center distance is too long for a straightedge. If a shaft is subject to end float, block it in its running position before aligning the sprockets. Secure the sprockets against axial movement by tightening setscrews.

Before installing the chain, recheck the preceding adjustments and correct any that may have been disturbed.

Wrap the chain around the sprockets, bringing the free ends together on one sprocket. To accomplish this, shorten shaft centers sufficiently. Connect the free ends by use of the connecting link or pins provided.

Readjust shaft centers to check chain tension. Chains should be installed fairly tight with only a small amount of slack. In the case of vertical drives, the chain should be kept snug and provision for adjustment of chain may be necessary.

New chains will loosen slightly owing to the seating of the joints as the chain is cycled over sprockets under load. After the first several weeks of operation, it is advisable to adjust the centers,

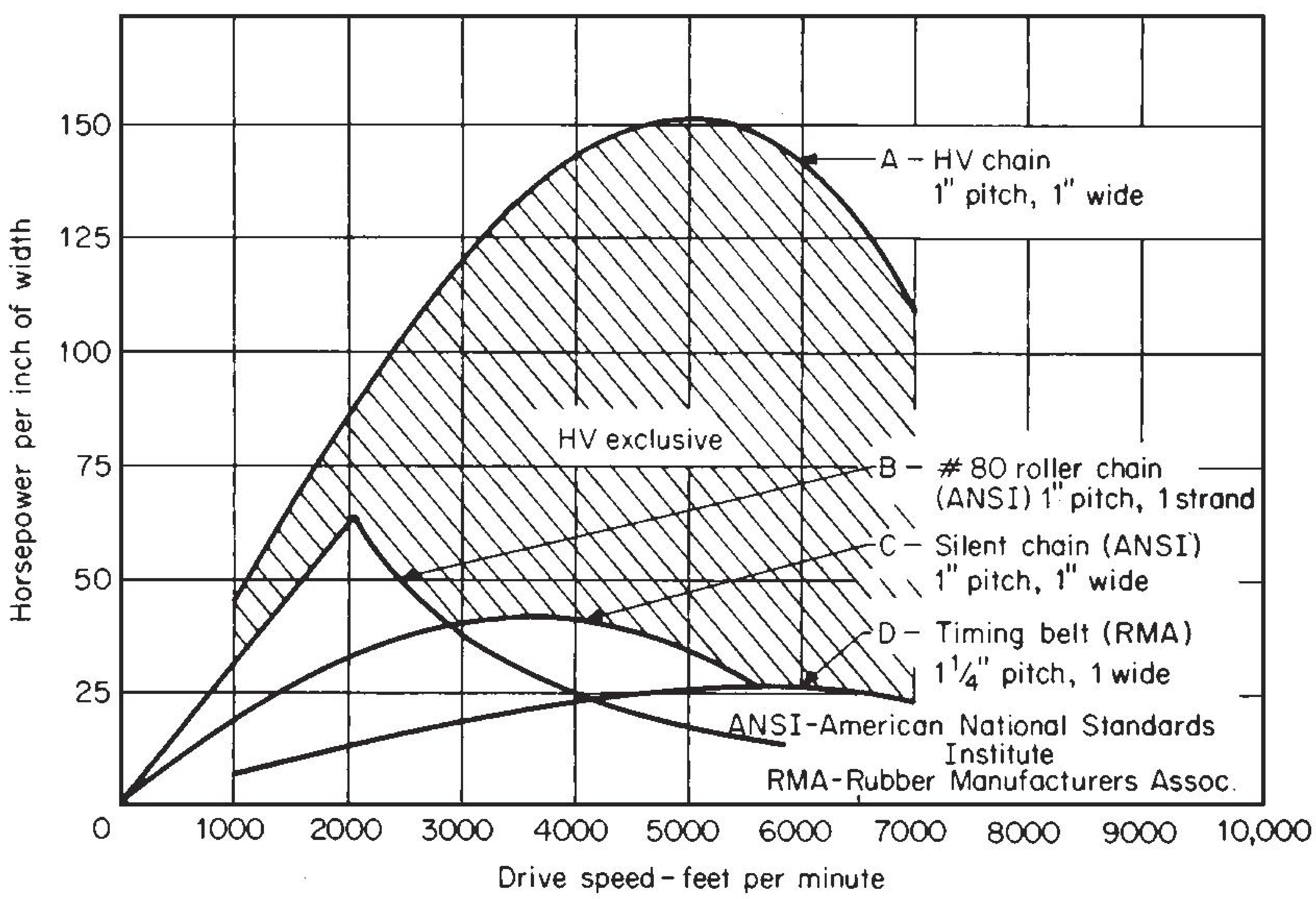

FIGURE 4.5 Horsepower capacity—comparable drives.

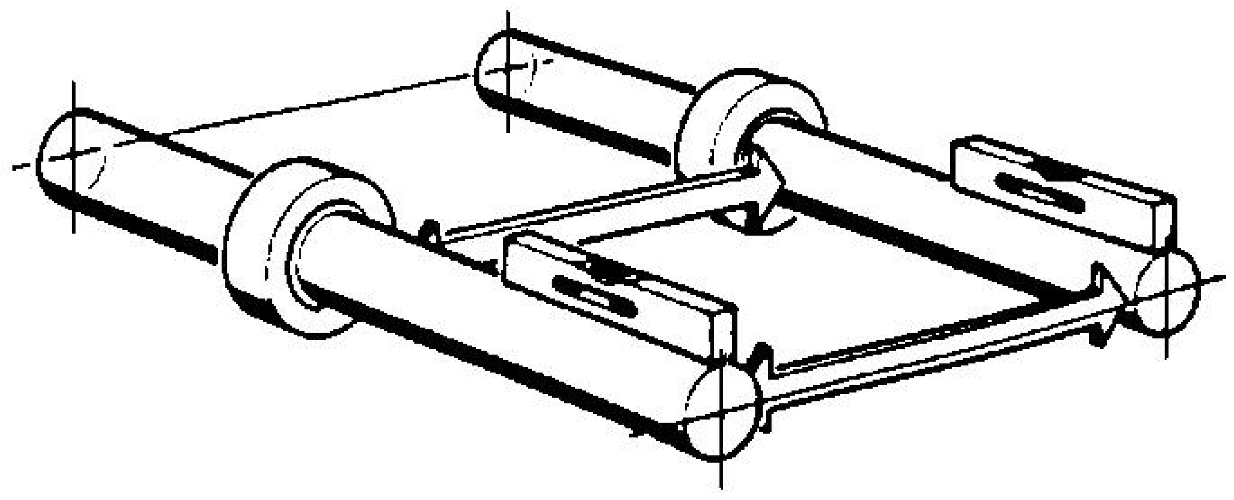

FIGURE 4.6 Shaft alignment.

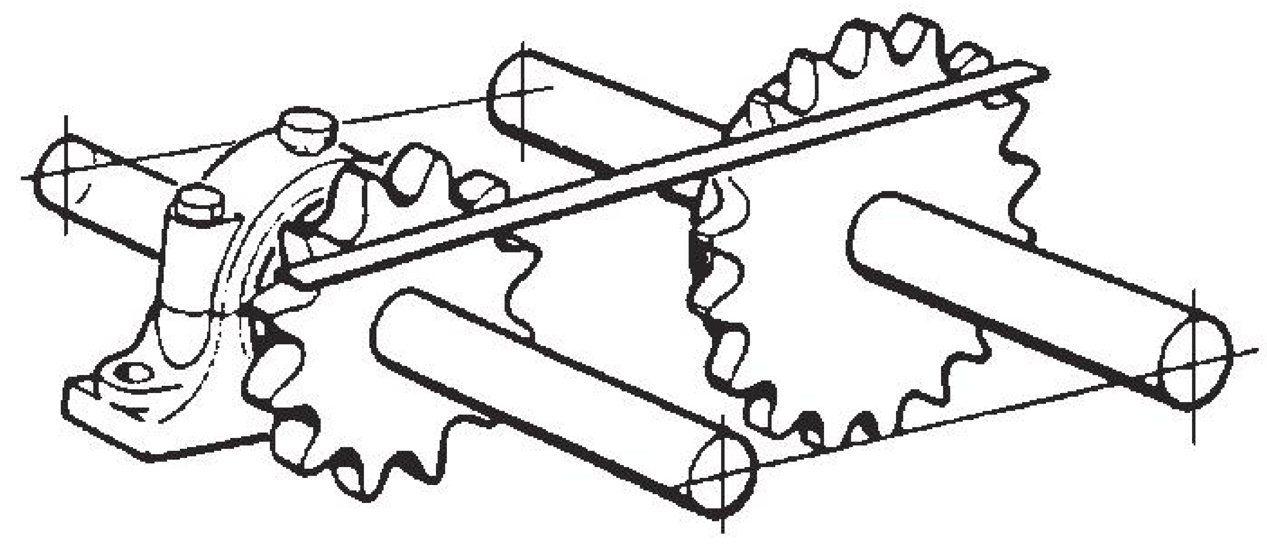

FIGURE 4.7 Sprocket alignment.

if needed, particularly on long center drives. After this initial elongation, with proper care and lubrication, precision chain drives will give long service without undue elongation or wear.

LUBRICATION OF PRECISION CHAIN DRIVES

With the increased speed and horsepower capabilities of modern chain drives, the role of lubrication has increased. The precision roller chain is actually a series of connected journal bearings, and it is essential that lubrication minimize the metal-to-metal contact of the pin-bushing joints of the chain. Many lubrication factors affect chain life.

Heat

Proper chain-drive lubrication will serve to increase the drive life by dissipating frictional heat generated in the joint area. This heat, which varies according to the chain speed, horsepower transmitted, center distance, ratio, drive size, amount and viscosity of lubricant, etc., will range from surrounding temperature to 60 to 70°F above the ambient temperature. Normal chain-drive temperatures should not exceed 180°F.

Improper Lubrication

A lubrication-starved chain drive will show a brownish (rusty) coloration around the joints and in the roller-bushing areas when the link is disassembled and the pin is inspected. The normal highly polished surface of the pin will have deteriorated to a roughened, grooved, or galled surface, which can eventually destroy the hardened surfaces of the chain parts and increase wear until the drive is completely destroyed.

Windage

A chain drive can be running through a sump of good lubricant and still destroy itself from lack of lubrication if the speed exceeds 2500 ft/min (fpm). The chain is actually blowing the lubricant out of its path. In high-horsepower, high-speed drives, it is necessary to use pressurized streams to ensure proper lubrication of the articulating components and to dissipate the heat generated. This lubricant should be sprayed onto the inside of the chain as it enters the sprockets to allow the centrifugal force to carry the lubricant through the joints.

Contamination

Lubricants should be protected from dirt and moisture. A filtering system should be utilized to remove wear particles and abrasive particles to minimize wear on the drive chain.

Oil Viscosity

A good grade of lubricant should be used between the chain parts to maximize the wear life of a chain drive. Lubricants containing antifoam, antirust, or film-strength-enhancing additives may be useful. It is essential that the lubricant reach the sideplate wearing surfaces and pin bushing areas. Therefore, normally heavy oils and greases are not recommended. The lubricant should be free-flowing at the prevailing temperature (Tables 4.2 and 4.3).

The type of lubrication will be dictated by the speed of the chain and the amount of power transmitted. The choice of lubrication method will be determined by the drive itself.

TABLE 4.2 Recommended Lubricants and Temperatures for Roller Chain

Temp, °F	Recommended lubricant
0–20	SAE 10
20–40	SAE 20
40–100	SAE 30
100–120	SAE 40
120–140	SAE 50

TABLE 4.3 Maximum Roller-Chain Speeds, ft/min (fpm)

	Chain No.										
Lubrication	35	40	50	60	80	100	120	140	160	200	240
Type A manual and drip	370	300	250	220	170	150	130	115	100	85	75
Type B bath	2800	2300	2000	1800	1500	1300	1200	1100	1000	900	800
Type C pumped	Suitable up to the maximum chain speed										

METHODS OF LUBRICATION

Type A: Manual or Drip Lubrication

When manual lubrication is used, oil is applied periodically with a brush or spout can, preferably once every 8 hours of operation. Volume and frequency should be sufficient to prevent discolorization of the lubricant in the chain joints.

When drip lubrication is used, oil drops are directed between the link plate edges by a drip lubricator. Volume and frequency should be sufficient to prevent discolorization of the lubricant in the chain joints. Precaution must be taken against misdirection of the drops by windage. Drops on the center of the chain will not effectively lubricate the joint areas. The lubricant should be directed at the inside of the pin and roller sideplate surfaces.

Type B: Bath or Disk Lubrication

With bath lubrication, the lower strand of the chain runs through a sump of oil in the drive housing. The dynamic oil level should be at the pitch line of the chain at its lowest operating point. With disk lubrication, the chain operates above oil level. The disk picks up oil from the sump and deposits it

on the chain, usually by means of a trough. The diameter of the disk should be sufficient to produce rim speeds between 600 and 8000 ft/min (fpm).

Type C: Oil-Stream Lubrication

The lubrication is usually supplied in a continuous stream to each chain drive. Oil should be applied inside the chain loop, evenly across the chain width, and directed, preferably, at the slack strand.

MAINTENANCE OF PRECISION CHAIN DRIVES

As in the case of any precision-built mechanism, proper maintenance contributes toward long, satisfactory service life.

Before discussing maintenance procedure, it must be assumed that the drive components have been properly selected for the installation, the chain and sprockets have been properly installed, and adequate lubrication has been provided.

1. Every chain drive should be checked periodically for alignment. Misalignment is conclusively indicated when the sides of sprocket teeth or inside surfaces of the chain-link plates show wear. Immediate steps should be taken to realign the drive when these defects are evident.

2. Chain should be checked for excessive slack. If the chain is running close to the tips of the teeth of the larger sprocket, the chain should be replaced. This can be checked visually while the drive is running or by lifting the chain away from the large sprocket, making sure the chain is in mesh with the sprocket teeth, as indicated by arrows in the drawing (Fig. 4.8). Excess clearance is conclusive evidence that the chain has elongated in pitch, and no amount of tension adjustment will keep it properly meshed with the sprocket teeth. Continued operation will cause the chain to jump teeth, destroying the chain and/or sprocket teeth.

3. Do not install a new chain on sprockets that are badly worn. Worn sprockets should be replaced to ensure proper chain fit on the sprockets, thus eliminating the possibility of premature wear of the replacement chain. The life of a worn sprocket may be extended by reversing it on the shaft to bring a new set of working tooth surfaces into use. If this is done, be careful to check alignment and make sure the sprocket runs true in its new position.

4. New drives should be inspected frequently for any possible interference with the chain. Naturally, if a chain is rubbing or striking against any obstruction, it will necessitate premature replacement.

5. Packing foreign material between the sprocket teeth will occasionally cause the chain to ride high on the sprocket teeth, exert undue stresses and accelerate wear in the chain, and cause abnormal wear of the sprocket teeth.

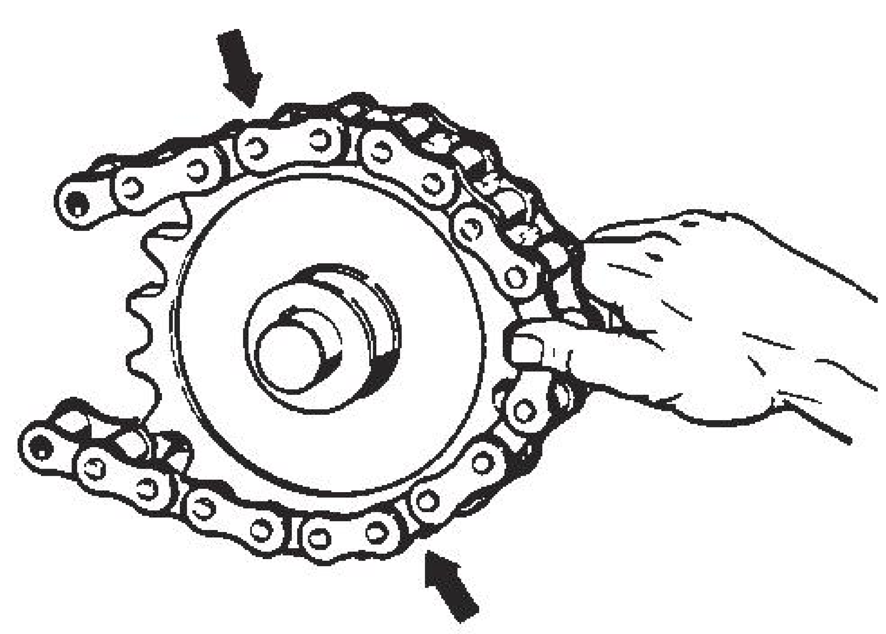

FIGURE 4.8 Elongated chain.

6. For all types of lubrication, check the quality and grade of the lubricant. For *manual* lubrication, make sure that the lubrication schedule is being followed, and that the oil is being properly applied. For *drip* lubrication, inspect the filling of the oiler cups and the rate of feed; check that the feed pipes are not clogged. Check to ensure that lubricant is applied to link plate edges so that it will penetrate to the pins. For *bath* or *disk* systems, inspect the oil level and check that there is no sludge. Drain, flush, and refill the system at least once a year. For *force-feed* systems, inspect the oil level in the reservoir and check the pump drive and delivery pressure; check that there is no clogging of the piping or nozzles. Drain, flush, and refill the reservoir at least once a year.

7. If roller chains have not been lubricated properly, the joints will have a brownish (rusty) color and the pins of the connecting link of the chain, when removed, will be discolored (light or dark brown). Also, the pins will be roughened, grooved, or galled. Properly lubricated chains will not show the brownish color at the joints, and the connecting link pins will be brightly polished with a very high luster.

8. Even under the best operating conditions, periodic cleaning of the chain is good economy. Gummed lubricant and the products of normal wear cause abnormally rapid pin and bushing wear. A chain exposed to dusty surroundings requires more frequent cleanings.

Clean a chain as follows:

- ***a.*** Remove the chain from the sprockets.
- ***b.*** Wash the chain in kerosene. If the chain is badly gummed, soak it for several hours in the cleaning fluid and then rewash it in fresh fluid.
- ***c.*** After draining off the cleaning fluid, soak the chain in oil to restore the internal lubrication.
- ***d.*** Hang the chain over a rod to drain off the excess lubricant.
- ***e.*** Inspect the chain for wear or corrosion. While the chain is off the sprockets, clean the sprockets with kerosene and inspect them for wear or corrosion.

9. Unless properly protected, the components of a chain drive will deteriorate during long periods of idleness. If a chain is to be stored, remove it from the sprockets and coat it with a heavy oil or light grease. Then wrap it in heavy, grease-resistant paper. Store the chain where it will be protected from moisture and mechanical injury. The sprockets may be left in place on the shafts. Cover each with grease, and protect them from mechanical injury. Before placing the drive in service again, thoroughly clean the chain and sprockets to remove the protective grease; then relubricate the chain.

CHAPTER 5
CRANES: OVERHEAD AND GANTRY

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GENERAL

Overhead and gantry cranes represent major investment in equipment. Reliable functioning of such equipment is generally vital to operations performed in the areas served by the cranes. Proper installation, operation, inspection, and maintenance of the crane are necessary to ensure performance and to avoid premature breakdowns or accidents which might injure persons working on, under, or near the crane. This chapter is intended to outline recommended procedures. It is by no means all-inclusive.

Manufacturer's instructions should be carefully read, retained, and followed. Attention also must be given to applicable federal standards, OSHA regulations, and state and local codes which include mandatory rules relating to crane inspection and maintenance. (Fig. 5.1.)

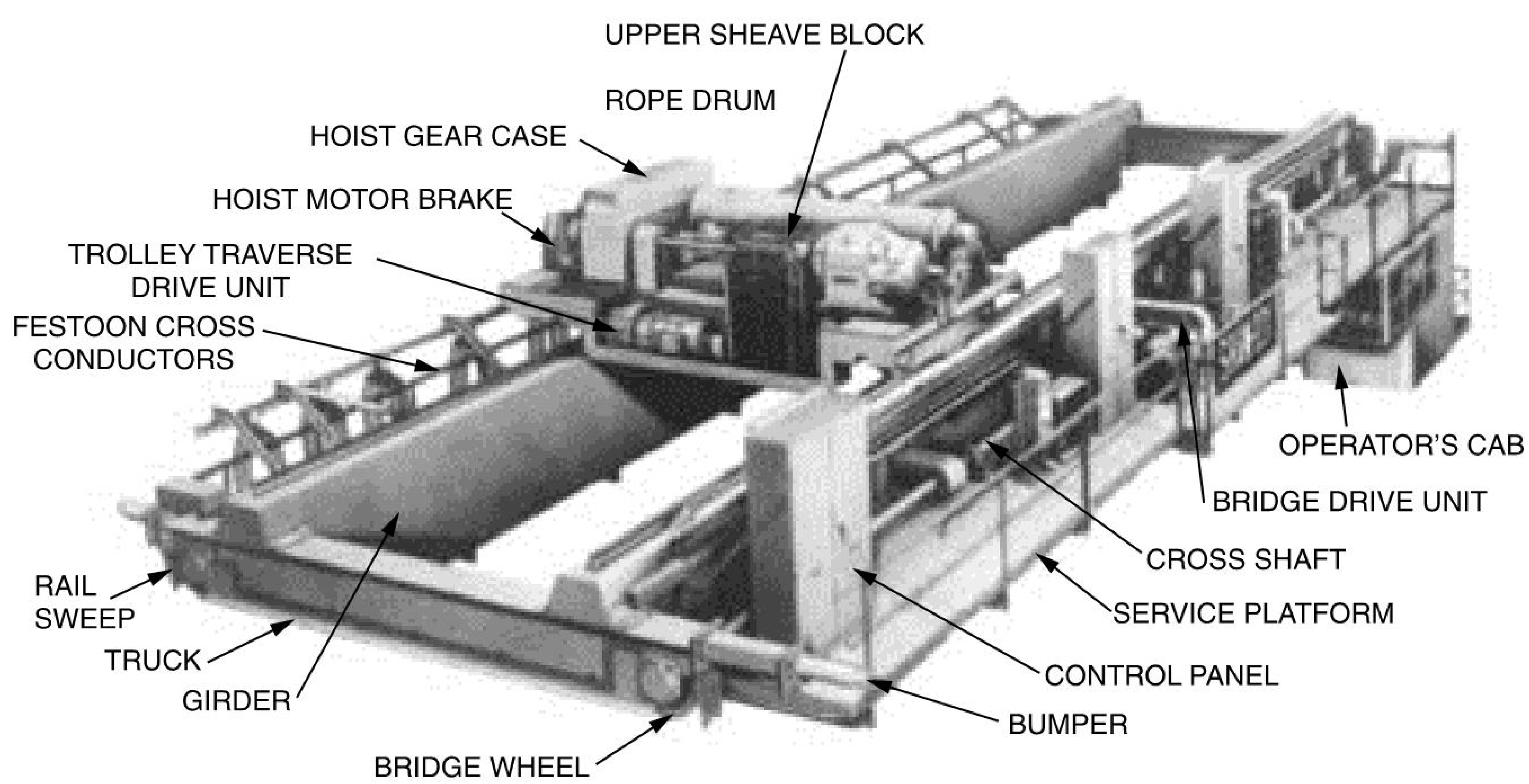

FIGURE 5.1 Typical heavy-duty overhead crane.

CRANE INSTALLATION

Good maintenance begins with a good installation. Prior to, during, and following erection of the crane, the following precautions should be observed:

1. Crane runway rails should be straight and accurately aligned to the correct span the entire length of the runway.
2. Be sure the crane is assembled in accordance with the match marks and instructions provided by the manufacturer.
3. It is of utmost importance that girders be square with the end trucks, that the trucks be parallel to each other, and that the bolts furnished with the crane be used for making the connection between the girders and end trucks. These bolts are usually of the ground-body type fitted into reamed holes.
4. Check all connecting bolts for tightness, and be sure lock washers or other locking means have been used as furnished.
5. Check for and remove any loose articles, such as bolts, hammers, or wrenches, which may have been left on top of the trolley or girders or on the platform.
6. Grease bearings on the crane as required. Check and add oil to gear housings as required.
7. Grease the hoisting cables.
8. Be sure the hoisting cable is reeved properly.
9. Check for any oil or grease spillage that may have occurred during erection, and wipe all oil spots dry.

Before the crane is placed in service:

1. Check operation of controllers for intended movement of each of the crane motions. In checking the hoisting unit, it is particularly important to note that with a three-phase electrical supply it is possible to have "reverse phasing," causing the hook to lower when the "up" push button, master switch, or controller is actuated. When this condition exists, the automatic limit switch will be inoperative, and hoist operation will be dangerous. To correct this condition, reverse one phase by interchanging any two of the supply wire leads. Do not rewire push buttons or master switches to effect this correction.
2. Check adjustment and operation of all brakes.
3. Check hoist-limit stops, and adjust for proper operation. The trip setting of hoist-limit switches should be determined by tests with the empty hook traveling in increasing speeds up to the maximum speed. The actuating mechanism of the limit switch must be located so that it will trip the switch, under all conditions, in sufficient time to prevent contact of the hook or hook block with any part of the trolley.
4. Check operation of other limit stops or locking or safety devices which may be installed on the crane or runway.
5. Operate the crane slowly through all motions, over the entire length of runway, entire length of bridge, and entire length of lift, checking for proper performance throughout.
6. New cranes and those in which load-sustaining parts have been altered, replaced, or repaired should be subjected to a load test confirming the load rating of the crane. The load rating should be not more than 80 percent of the maximum load sustained during the test, and test loads should not be more than 125 percent of the rated load, unless otherwise recommended by the manufacturer.

CRANE INSPECTION

The frequency of inspection and degree of maintenance required for cranes vary with the service to which the cranes are subjected, heavily used cranes requiring more attention than standby or lightly

used cranes. Close attention should be given the crane in the first few days and weeks of operation, following which routine inspection procedures should be instituted.

Daily to monthly inspections are recommended to include

1. Operation of all limit switches, without load on hook (the crane motion should be inched or run into the limit position at slow speed for these checks)
2. All functional operating mechanisms for misadjustment, damage, or wear
3. Air and hydraulic systems components for deterioration or leakage
4. Hooks for deformations, cracks, and wear
5. Hoisting ropes for broken wires, abrasions, kinks, or evidence of not spooling properly on drum

Monthly to yearly inspections should include checks on

1. Loose connections, bolts, nuts, rivets, keys, etc.
2. Cracked, worn, deformed, or corroded members, including rails or beam flanges on which the crane operates
3. Cracked, worn, or distorted mechanical parts such as shafts, bearings, pins, wheels, rollers, gears, pinions, and locking or clamping devices
4. Excessive wear on brake parts, pawls, pins, levers, ratchets, linings, etc.
5. Rope drums and sheaves for excessive wear or cracks
6. Electric or other types of motor for performance and wear of commutator, slip rings, brushes, etc.
7. Chain and sprockets for excessive wear or stretch
8. Crane hooks for cracks, by magnetic particle, dye penetrant, or other reliable crack-detection method; and hook-attaching means including hook nut, locking pin, etc., for security of hook attachment to lower block
9. Load-limiting or other safety devices installed on the crane
10. Electrical devices, controls, and wiring for signs of deterioration or wear; electrical contactor points for excessive pitting

Standby cranes should be inspected at least semiannually, and more frequently in adverse environment.

Written, dated, and signed inspection reports and records need to be maintained, particularly on critical items such as crane hooks, hoisting ropes, sheaves, drums, and brakes. Figure 5.2 shows the first sheet from the Overhead Crane Inspection and Maintenance Checklist published by and available from the Crane Manufacturers Association of America, Inc., 8720 Red Oak Blvd., Suite 201, Charlotte, NC 28217. Use of this or similar checklist is recommended. Cranes with identified safety hazards should be removed from service until repairs are completed unless other appropriate precautions are taken to eliminate possibility of an accident or injury to personnel.

CRANE MAINTENANCE

A preventive-maintenance program should be established based on the manufacturer's or a qualified person's recommendations. Service schedules and dated detailed records should be maintained.

Since the original equipment manufacturer is usually in the better position to provide replacement parts and ensure their safety, interchangeability, and suitability for the application, it is recommended that such parts be obtained from the original equipment manufacturer.

A good preventive-maintenance program identifies parts requiring replacement sufficiently in advance of actual need to permit ordering of parts after approaching need is identified; however, for cranes in regular service, it is generally advisable to carry on hand a reasonable minimum inventory of repair parts. The needed inventory will vary with type and age of crane, service to which it is

CRANE MANUFACTURERS ASSOCIATION OF AMERICA
CRANE INSPECTION SCHEDULE AND MAINTENANCE REPORT

Customer: ______________________ Date: __________

Capacity: __________ Span: __________ Type: __________
Mfr. Ser. No.: __________ Cust. Idnt. No.: __________

	Component & Location	Inspection Interval			Condition						Corrective Notes
Location	Component	Weekly	Monthly	Semi-An'l.	OK	Adjust	Repair	Replace	Lubricate	Clean	Describe, Initial, and Date When Corrected
Bridge	Motor			○							
	Brake & Hydraulics	○									
	Control Panels			○							
	Control Operation		○								
	Resistors			○							
	Lights		○								
	Trolley Conductors			○							
	Runway Collectors		○								
	Reducer		○								
	Couplings		○								
	Line Shaft Bearings		○								
	Wheels		○								
	Wheel Gearing		○								
	Wheel Bearings		○								
	Girder Connections			○							
	Align. & Tracking			○							
	Trol. Rails & Stops			○							
	Guards & Covers		○								
	Bumpers			○							
	Rail Sweeps			○							
Cab	Master Switches	○									
	Mainline Disconnect	○									
	Warning Device	○									
	Fire Extinguisher		○								

FIGURE 5.2 This inspection report is being employed by many users of cranes. (Partial checklist shown.)

subjected, repair history, and general availability of parts. The crane manufacturer can provide lists of recommended spares.

Typical recommended spare-parts lists are apt to include

Brake solenoids, coils, disks, linings

Hoist-limit switches

Contactors
Contact kits
Timing relays
Push-button stations or parts
Crane wheels and guide rollers
Motor couplings and brushes
Current collectors or collector shoes
Bearings
Load hooks, nuts, and thrust bearings
Hoisting ropes
Load brake parts

On-hand availability of parts such as these can often spell the difference between a long, costly wait for repair and efficient replacement at a convenient time in advance of actual breakdown.

When ordering parts for a crane, observation of the following points can save time and expense:

1. Identify the crane by manufacturer's serial number.
2. Refer to the parts manual furnished by the manufacturer, and identify parts by the numbers given.
3. For cranes with auxiliary hoists, specify whether parts are for main or auxiliary hoist; if for a bucket crane, if parts are for the holding-line or closing-line mechanism.
4. If the crane carries more than one trolley or hoist, specify which trolley or hoist the parts relate to.
5. When ordering brake parts, specify which brake, whether hoist (main or auxiliary), trolley, or bridge.
6. If parts are for electrical equipment or other equipment not shown on the parts lists, describe the part and identify the serial number of the unit for which it is required.

Before adjustments or repairs are started, several precautions should be taken:

1. The crane to be repaired should be located where it will cause least interference with other operations in the area.
2. All controllers should be placed in the off position.
3. Main and emergency switches should be locked in the open position.
4. Warning signs should be placed on the crane and on the floor beneath the crane or on the crane hook if near the floor.
5. If other cranes are operating on the same runway, rail stops or other means should be provided to prevent collision with the idle crane, or a signal man may be employed to warn off approaching cranes.

Following completion of adjustments and repairs, all guards and safety devices must be reinstalled and maintenance equipment removed before the crane is returned to service.

Adjustments and repairs to cranes should be done by designated and qualified personnel as soon as possible after identification of need, and before further use of the crane if a safety hazard is involved. Adjustments should be maintained for optimal crane performance and to ensure the safe functioning of all systems and components.

Hook deformation is usually a sign of tip loading of the hook or overloading of the crane. If overloading is suspected, other load-bearing parts of the crane should be checked for possible damage due to overloading. Hooks with cracks, or having a throat opening in excess of normal, or with twist from the plane of the unbent hook should be considered for replacement (see ANSI and other applicable standards). Hooks showing wear in the saddle of the hook, which indicates reduction of strength of the hook, also should be considered for replacement.

Any load-bearing parts which are cracked, bent, or excessively worn must be repaired or replaced.

Pitted or burned electrical contacts should be corrected only by replacement and only in sets.

All control stations should be kept clean with function labels intact. Missing or illegible warning labels must be replaced promptly.

Lubrication should be applied regularly to all moving parts for which lubrication is specified and/or indicated by lubrication fittings. Follow the manufacturer's recommendations as to frequency and types of lubricants to be used. Avoid overgreasing of bearings and overfilling of gear cases. Unless the crane is equipped with automatic lubricators, the same preliminary precautions should be taken for lubricating the crane as for making repairs.

Hoisting ropes require special attention. On cranes in continuous service, ropes should be inspected visually daily, and a thorough inspection should be made at least once a month, with written record made as to rope condition and possible need for replacement. All inspections should be made by an appointed or authorized person. Any form of rope deterioration which could result in appreciable loss of original rope strength should be carefully noted. Conditions such as the following require a determination as to whether continued use constitutes a safety hazard.

1. Kinked, crushed, cut, or unstranded sections
2. Broken, worn, or corroded outside wires
3. Reduction of rope diameter due to loss of core support, corrosion, or wear
4. Damaged end connections or damaged rope wires at the connections

Ropes which have been out of service for long periods of time should be checked carefully before service is resumed.

Replacement rope should be the same size, grade, and construction as the original rope furnished by the crane manufacturer, unless otherwise recommended by the crane manufacturer.

Rules governing replacement of ropes are not precise and hence require judgment by an appointed person as to whether the remaining strength and life in the rope are sufficient to permit continued use. This judgment should take into account the service to which the crane is subjected, as well as the observed condition of the rope. For example, many users consider the following to be conditions which produce serious question as to safety of rope for continued use:

1. Crushed, kinked, birdcaged, or otherwise distorted rope
2. Twelve or more randomly distributed broken wires in one rope lay or four or more broken wires in one strand of one rope lay
3. Wear on outside individual wires exceeding one-third of the original wire diameter
4. Evidence of heat damage
5. Reductions in nominal rope diameters of more than

$1/64$ in. for ropes of $5/16$ in. diameter and smaller

$1/32$ in. for ropes of $3/8$ to and including $1/2$ in.

$3/64$ in. for ropes $9/16$ to and including $3/4$ in.

$1/16$ in. for ropes of $7/8$ to and including $1\,1/8$ in.

$3/32$ in. for ropes of $1\,1/4$ to and including $1\,1/2$ in.

Wire rope should be stored and handled in a manner which avoids damage or deterioration. Unreeling or uncoiling rope requires care to avoid damaging the rope or introducing twist.

Before a rope is cut, seizings must be places on each side of the cut location to prevent unraveling of the rope when cut. Apply seizings as follows:

Preformed rope: one seizing each side of cut

Nonpreformed rope $7/8$ in. or smaller: two seizings each side of cut

Nonpreformed rope 1 in. or larger: three seizings each side of cut

Hoisting ropes should be maintained in well-lubricated condition to reduce internal friction and prevent corrosion. See crane and/or rope manufacturer's recommendations.

GOVERNMENTAL REGULATIONS

The reader is cautioned to recognize that a great many of the recommendations presented in this chapter are now mandatory under OSHA regulations which became effective in 1972 and are also likely to be mandatory under various state safety codes. It should be further noted that OSHA regulations require cranes to conform to the National Electrical Code (ANSI C1). These various regulations are subject to change. OSHA and other regulations applicable to your location should therefore be carefully checked for current requirements. Such codes generally deal with the crane equipment and its operation as well as with maintenance.

For further information on crane maintenance, the reader is referred to the following standards, codes, and regulations:

American National Safety Standards, published by The American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York, NY 10017:

ANSI B30.2.0: Overhead and Gantry Cranes (Top Running Bridge, Multiple Girder)

ANSI B30.9: Slings

ANSI B30.11: Monorails and Underhung Cranes

ANSI B30.16: Overhead Hoists

ANSI B30.17: Overhead and Gantry Cranes (Top Running Bridge, Single Girder Underhung Hoist)

CMAA Crane Specifications, published by The Crane Manufacturers Association of America, Inc., 8720 Red Oak Blvd., Suite 201, Charlotte, NC 28217

HST Performance Standards for Hoists, Published by ASME, 345 East 47th St., New York, NY 10017

OSHA regulations as published in the *Federal Register.* (*Note:* Portions of ANSI standards, such as ANSI B30.2.0, have been adopted as OSHA regulations.)

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CHAPTER 6
CHAIN HOISTS

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Chain hoists, both manually and power operated, are a widely used type of hoisting equipment. Their simplicity, dependability, and relatively low cost have made them standard equipment in manufacturing plants, foundries, mills, refineries, repair shops, garages, and practically every phase of the construction field.

This chapter describes the various types of chain hoists, explains their relative advantages and usual applications, and provides information on preventive maintenance, inspection, and upkeep.

TYPES OF CHAIN HOISTS

Manually Lever-Operated Chain Hoists (Pullers)

These lightweight, portable tools can be used for pulling horizontally, vertically, or at any angle. A reversible ratchet mechanism, located in the lever, permits short-stroke operation for both tensioning or relaxing. An automatic friction-type load brake, sometimes known as a *releasable ratchet,* is often used to control the load and permit accurate positioning. Other types of lever-operated units utilize a ratchet and pawl-type of brake, which involves ratcheting in both directions, for load control.

Lever-operated hoists are commonly available in both link and roller chain types, in capacities from ½ thru 15 tons. See Fig. 6.1.

Hand-Chain Manually Operated Chain Hoists

These hoists are most frequently used for overhead lifting applications where the use of a powered hoist may not be practical. These would include many maintenance- or construction-type applications where a power source may not be readily available or the portability, load-spotting accuracy, or close-quarter capability of a hand-chain operated hoist may be required.

There are a number of types and classes of hand-chain manually operated chain hoists available, ranging from heavy-duty, high-speed ball bearing spur-geared units, designed for heavy-duty industrial service, to much lighter-weight units with less efficient gearing and power trains designed for more infrequent service.

Heavy-duty, high-speed, spur-geared units, incorporating low-friction bearings and a Weston self-energizing disk-type load brake, typically have very high mechanical efficiency ratings, permit-

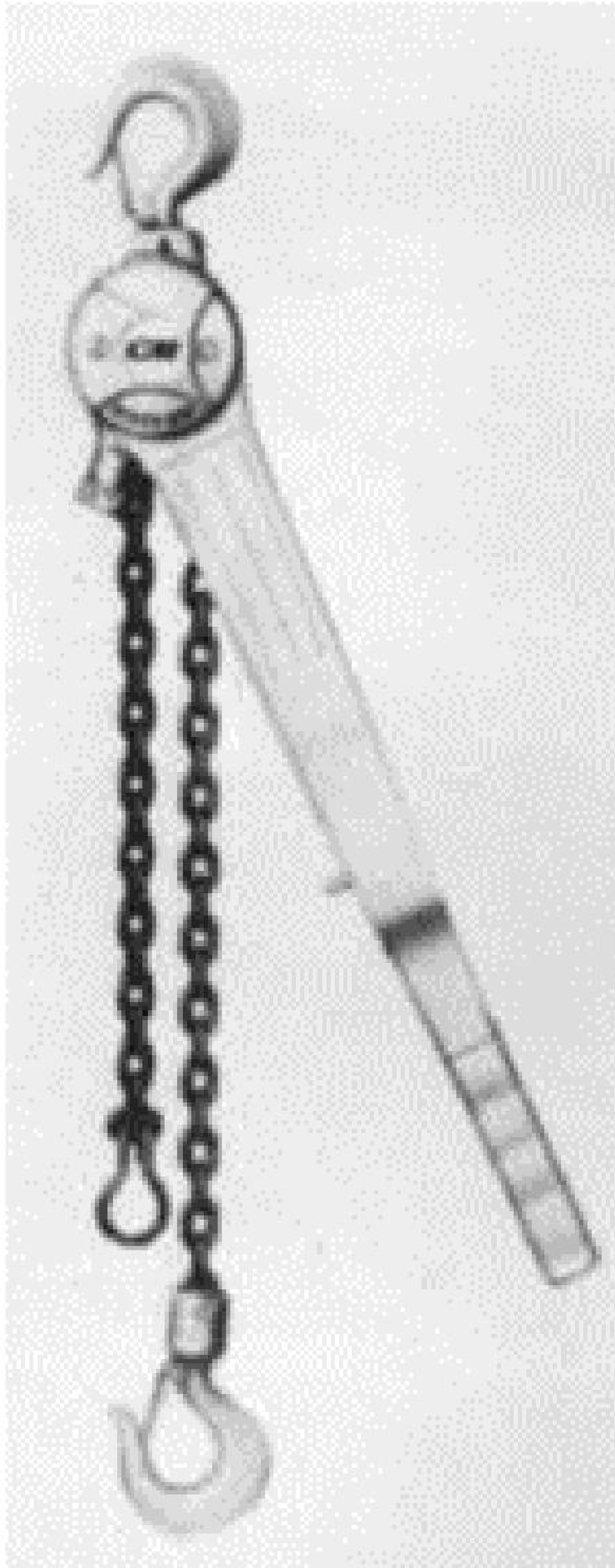


FIGURE 6.1 Puller or ratchet-level hoist with link chain.

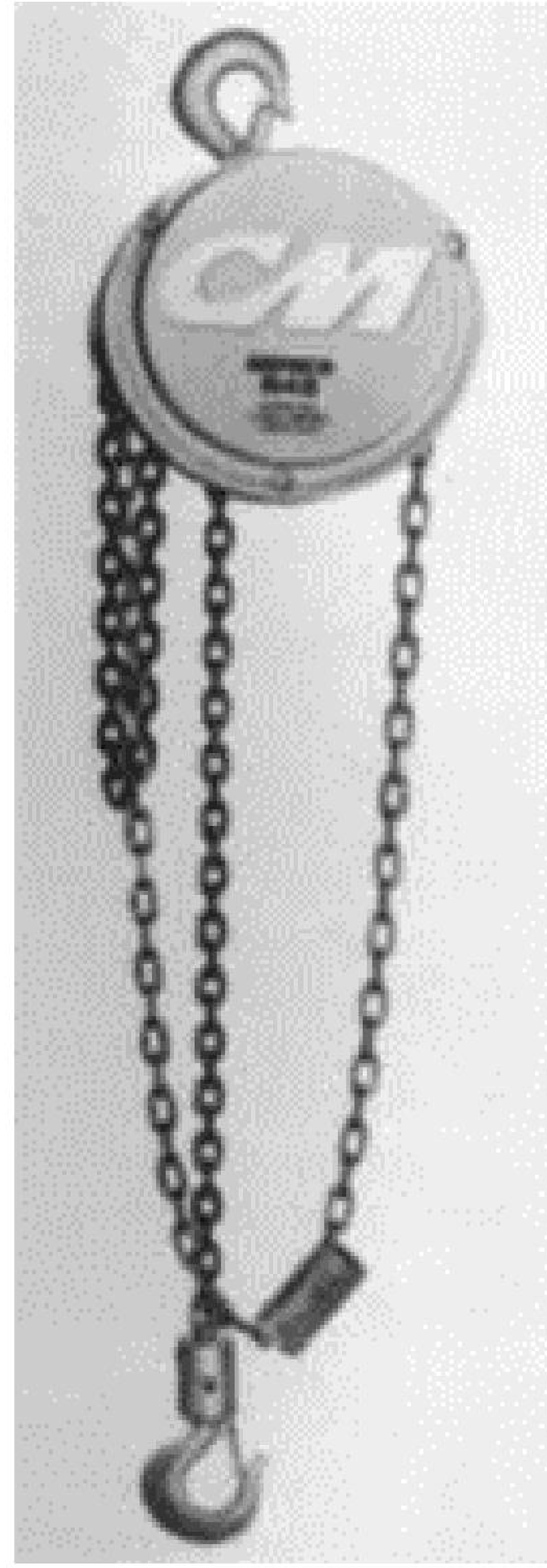


FIGURE 6.2 Spur-geared hoist.

ting operation at high speeds in the heavy-duty applications often encountered in industry. Separate hand and load chains operate over pocket wheels which are connected by a gear train and load brake. The brake is disengaged during hoisting by a one-way ratchet mechanism. To lower the load, the hand chain must be pulled continuously in the reverse direction to overcome the holding force of the brake.

Modern hand-chain manually operated chain hoists utilize more compact designs and lightweight alloys to achieve much greater portability through a significant weight reduction as compared with earlier hoist models. See Figs. 6.2 to 6.4. Lower-cost, lighter-duty, spur-geared hand-chain manually operated chain hoists often incorporate gears which are not accurately machined, unlike those employed in high-speed units, and bushings are frequently used in place of low-friction bearings. While these lighter-duty units are somewhat lower in cost, they also operate at mechanical efficiencies 30 to 40 percent lower than those required for high-speed industrial service. Further, these lighter-duty units will typically not withstand the more severe service encountered in rigorous industrial-type applications.

Worm gearing is sometimes used in hoisting applications to provide an inexpensive power train incorporating a high ratio in a relatively small space. These types of power trains are approximately 60 percent less efficient than high-speed spur gearing, however. This is especially true if a high enough gear ratio is utilized to make the power train self-locking. See Fig. 6.5.

A differential hoist is generally the simplest and least expensive chain hoist. It has an efficiency rating approximately 30 percent of that of a high-speed spur-geared unit. A dual-pocketed upper sheave and grooved single lower sheave are connected by an endless reeved chain for lifting and operating. The mechanical advantage is gained by the two upper sheaves differing by one link pocket

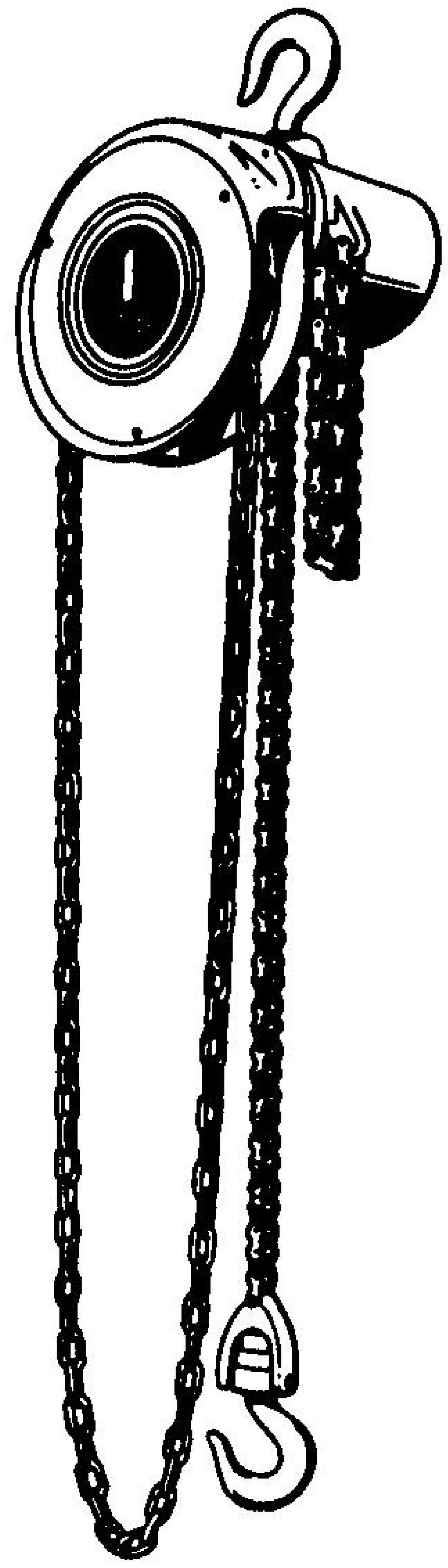


FIGURE 6.3 Spur-geared single-reeved hoist with roller chain.

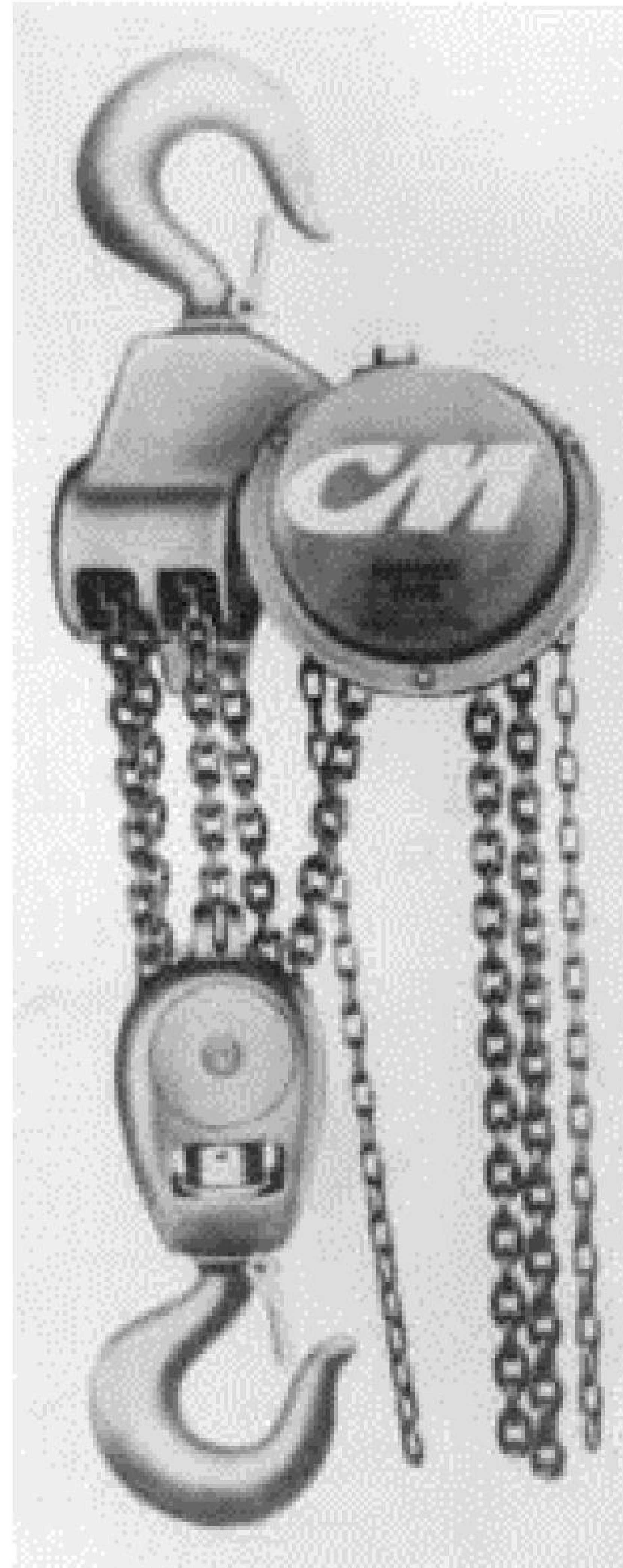


FIGURE 6.4 Multiple-reeved spur-geared hoist with link chain.

so that more chain passes over the larger wheel than the smaller one to produce a net raising or lowering of the load. The difference in diameter of these two sheaves results in such a small turning moment on this combination that normal friction holds the load in the suspended position at any point, and this serves as the hoist braking means. An effort must be applied to either raise or lower the load hook. See Fig. 6.6.

There are a number of variations to the hook-suspended hand-chain manually operated chain hoist, which are sometimes encountered, as follows:

Low-headroom army-type trolley hoists are spur-geared units with an integral trolley to provide greatly reduced headroom for close-quarter operation. Figure 6.7 illustrates a low-headroom hoist with plain trolley. Hand-geared trolleys are also available.

Twin-hook hoists are spur-geared hoists with two hooks which are operated simultaneously by one hand chain and are adapted to the handling of bulky objects which require two-point suspension. A twin hoist will lift a total load equal to the capacity marked on the hoist. The load often can be all on one hook or the other or divided any way between the two hooks. Typical extensions range from 3 to 16 ft. See Fig. 6.8.

Extended-handwheel hoists have a handwheel extending from a spur-geared hoist and are designed for service which requires that the hand chain and operator be at a distance from the load being raised. Typical extensions range from 3 to 16 ft. See Fig. 6.9.

Often, chain hoists are used to lift a load, the weight of which may be totally unknown. Because of this fact, many chain hoists are available with overload limiting devices designed to prevent the lifting of dangerous overloads.

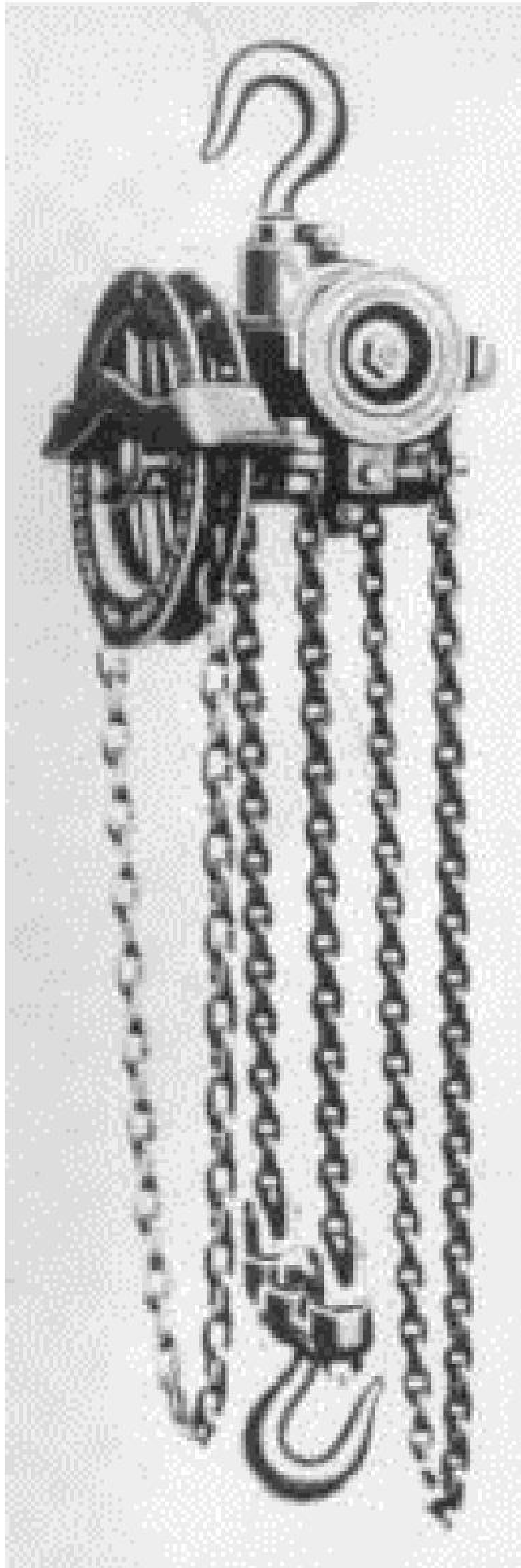


FIGURE 6.5 Screw-geared hoist.

FIGURE 6.6 Differential hoist.

Powered Chain Hoists (See Figs. 6.10 to 6.12)

Powered chain hoists are typically used for repetitive higher-speed lifting, as often encountered in production applications. Most powered hoists utilized electrical power, although quite a number of chain hoists are air-powered. Both types of powered units are equipped with either push-button or pendant rope controls. Both link-type chain and roller chain are utilized in powered hoists. Link chain has the advantage of being flexible in all directions, whereas roller chain is flexible in only one plane. Also, powered hoists are normally equipped with travel limit switches to restrict upper and lower extremes of travel, thus preventing the load hook from jamming against the bottom of the hoist or the chain from running out of the hoist.

Electric chain hoists are available for use with most types of current. Many small-capacity models are equipped with single-phase 115-V motors, which can be plugged into a standard receptacle which receives a standard three-prong plug. Some manufacturers offer three-phase dual-voltage single-speed models as well as three-phase single-voltage two-speed models. Most single-speed dual-voltage units are reconnectible for operation on either 230 or 460 V, 60-Hz power. Two-speed units are generally built to operate on one specific three-phase voltage (i.e., 230 V, 460 V, etc.). Also, other voltages such as 575 V, three-phase, and 230 V, single-phase, as well as a large number of 50-Hz export voltages, are also readily accommodated.

Powered hoists are generally available with a wide variety of suspensions and accessories. These units may be equipped with swivel or rigid hook, lug, plain trolley, hand-geared trolley, or powered trolley suspensions. Chain containers are often used to collect the unloaded loose end of the load chain

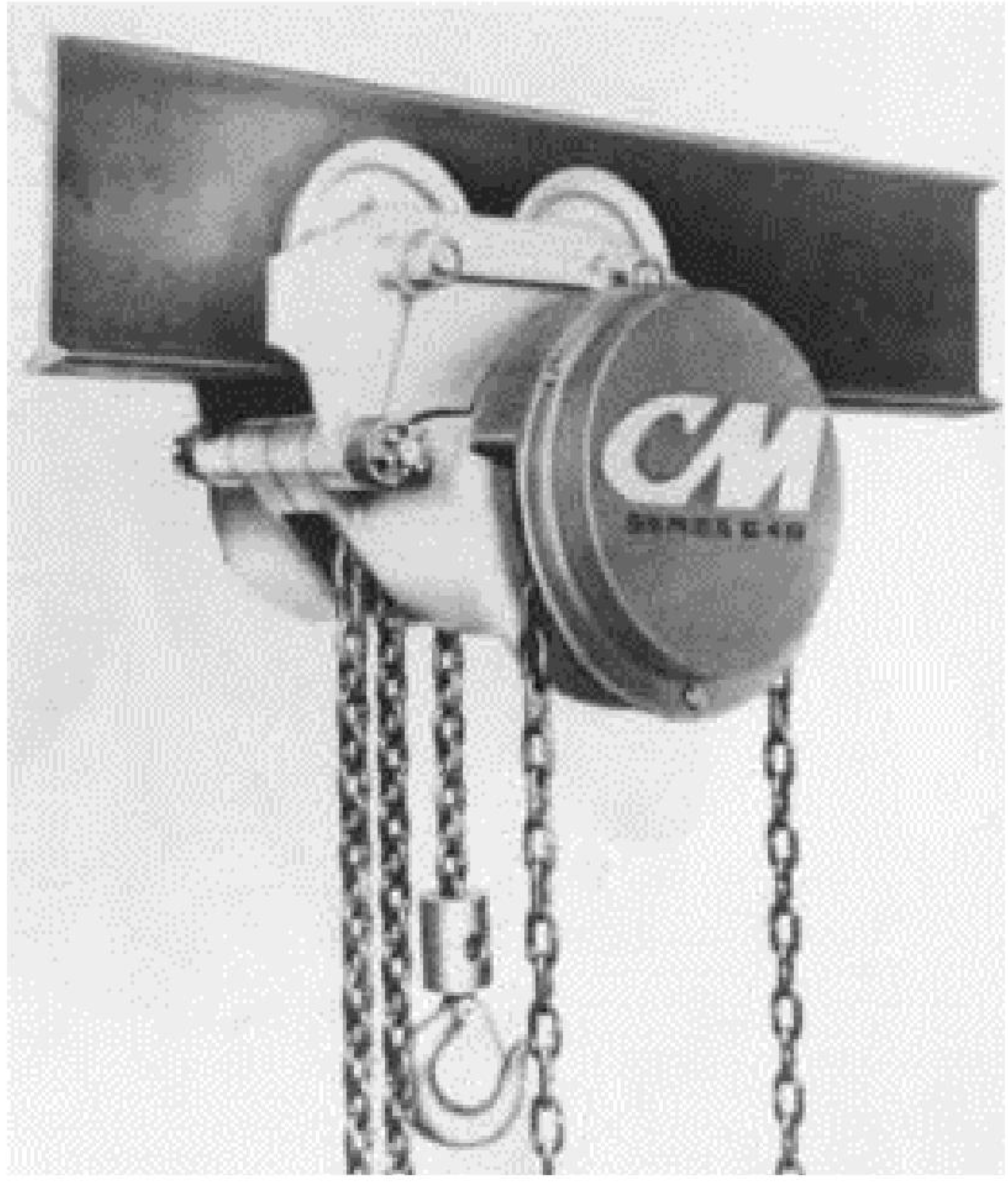


FIGURE 6.7 Low-headroom trolley hoist with link chain.

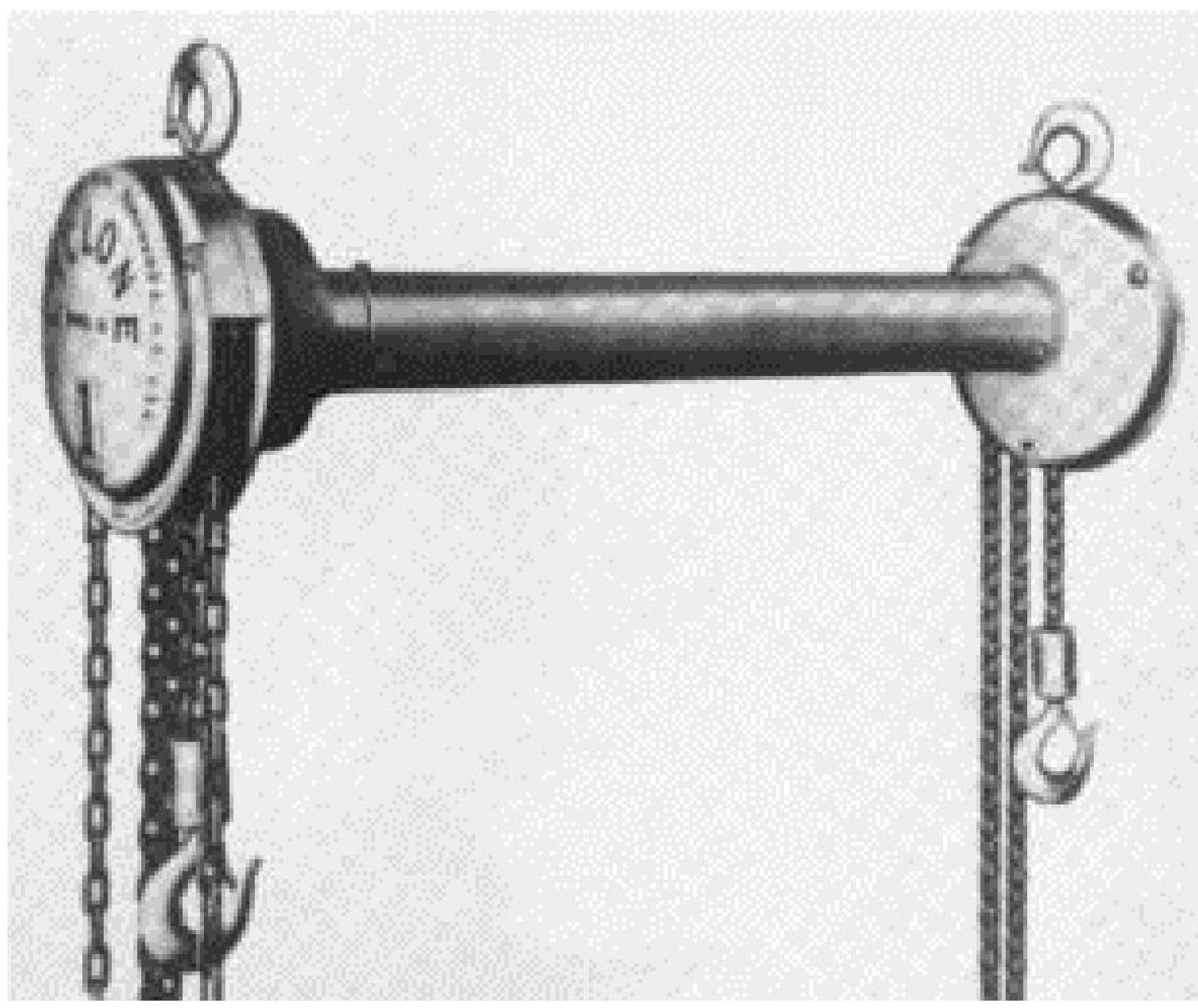


FIGURE 6.8 Twin-hook hoist with link chain.

at the hoist. A wide variety of power-distribution systems are also available to allow for travel of powered hoists on monorail or bridge crane applications. Often, powered hoist controls are integrated with trolley travel, bridge travel, mainline power disconnect systems, powered accessories, etc.

Powered chain hoists, in capacities from ⅛ to 15 tons, are widely used throughout industry because of their convenience, relatively low cost, and durability.

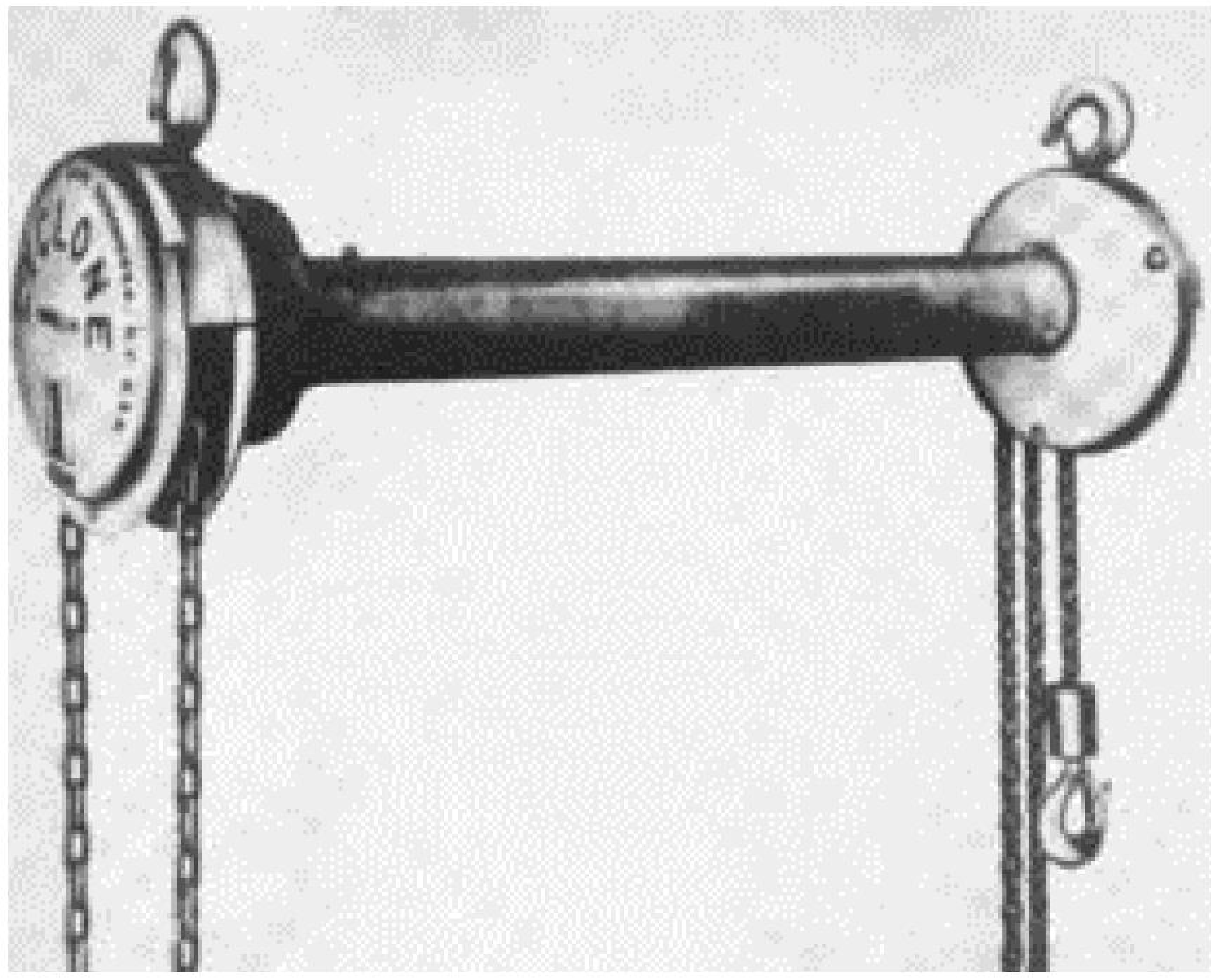


FIGURE 6.9 Extended-handwheel hoist with link chain.

FIGURE 6.10 Lightweight electric chain hoist with push-button control and low headroom trolley.

FIGURE 6.11 Lightweight electric chain hoist with push-button control and hook suspension.

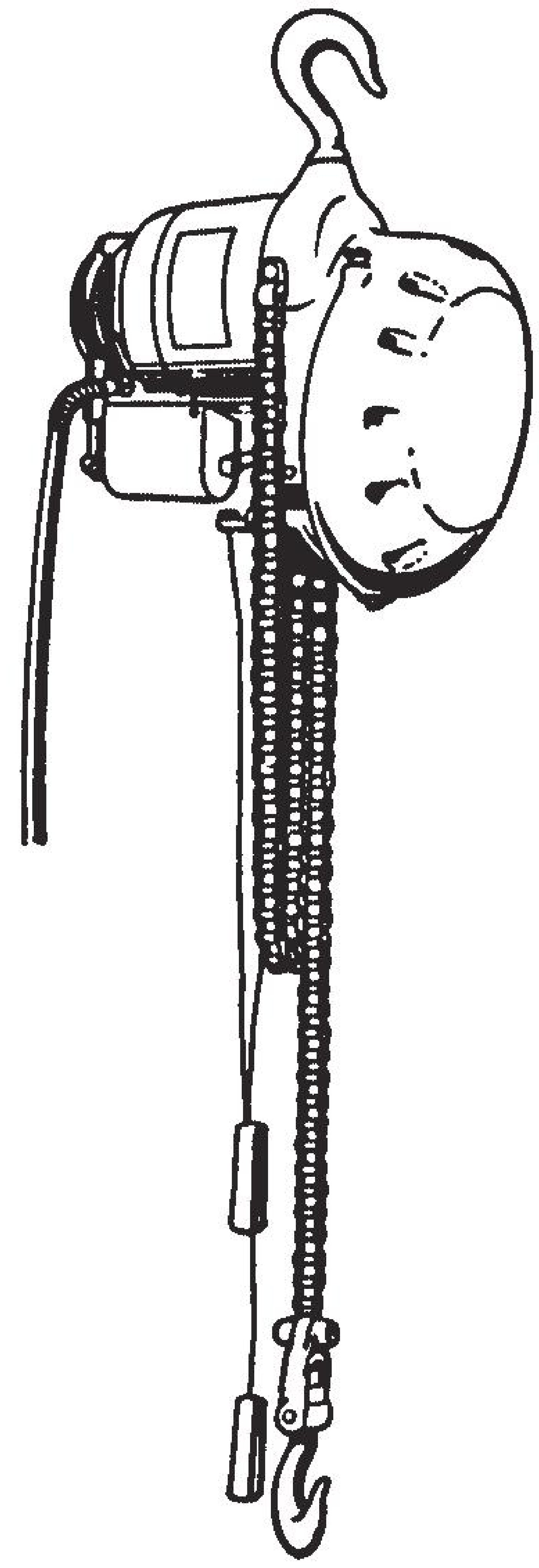
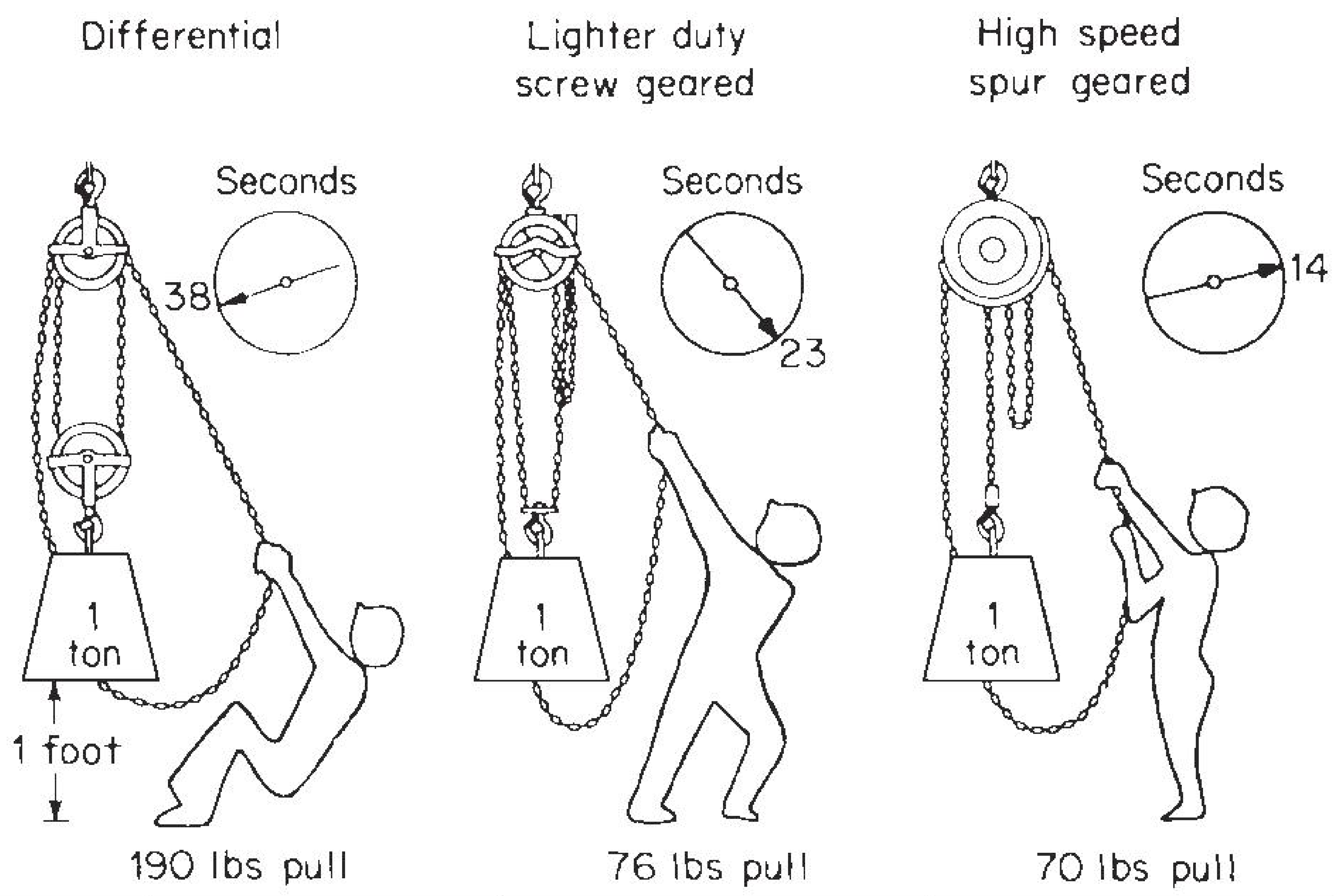


FIGURE 6.12 Electric chain hoist with roller chain and pendant rope control.

SELECTION OF CHAIN HOISTS

In selecting either manually operated or powered hoists, certain considerations are basic and common to both types. Figure 6.13 provides a graphic comparison of the types of manual hoists used today for overhead lifting, each offering a varied range of mechanical efficiency and price and filling the need of a particular condition. Figure 6.14 illustrates the important performance and physical characteristics of both manually operated and powered hoists which must be considered in selecting a unit for a given use and specific installation.

Intended use, safety, labor savings, portability, initial cost, and upkeep are all important factors to be considered in properly selecting a hoist for an application. With high labor costs for both operation and maintenance, low initial cost may not be the true measure of hoist value for a given application.



Mechanical efficiency	35%	60%	95%
Cost comparison	30%	75%	100%
Standard capacities	1/2 to 1 ton	1/4 to 10 tons	1/4 to 40 tons

FIGURE 6.13 Comparison of hand hoists.

Hoist capacity, in terms of the heaviest load to be lifted, is of prime importance. Headroom, height of lift, location and height of hand chain or push-button/pendant controls, lifting speed on powered hoists, type of suspension, travel speed on powered trolleys, hoist and trolley clearances, etc., are all factors affecting decisions on specific installations. Figure 6.14 is a self-explanatory diagram and checklist which will be found useful in selecting either a manually operated or powered hoist.

Unusual atmospheric conditions, whether indoors or outdoors, may require sealed enclosures, weatherproof covers, or special protective coatings on the housings, chain, and other fittings. Under normal atmospheric conditions encountered in typical indoor applications, standard hoists are generally satisfactory.

PREVENTIVE MAINTENANCE

For all aspects of chain-hoist preventive maintenance, we suggest that the recommendations contained in ANSI/ASME Standards B30.16 and B30.21 be followed in detail. These ANSI/ASME standards represent an industry consensus published by the American Society of Mechanical Engineers, 22 Law Drive, P.O. Box 2300, Fairfield, NJ 07007-2300, which is nationally recognized and in wide use today.

DESIGN/PERFORMANCE

For additional detailed information on various types of chain hoists, the following standards are referenced:

ANSI/ASME HST 1, Electric Chain Hoists

ANSI/ASME HST 2, Overhead Manual Hoists

ANSI/ASME HST 3, Manual Lever Chain Hoists

ANSI/ASME HST 5, Air Chain Hoists

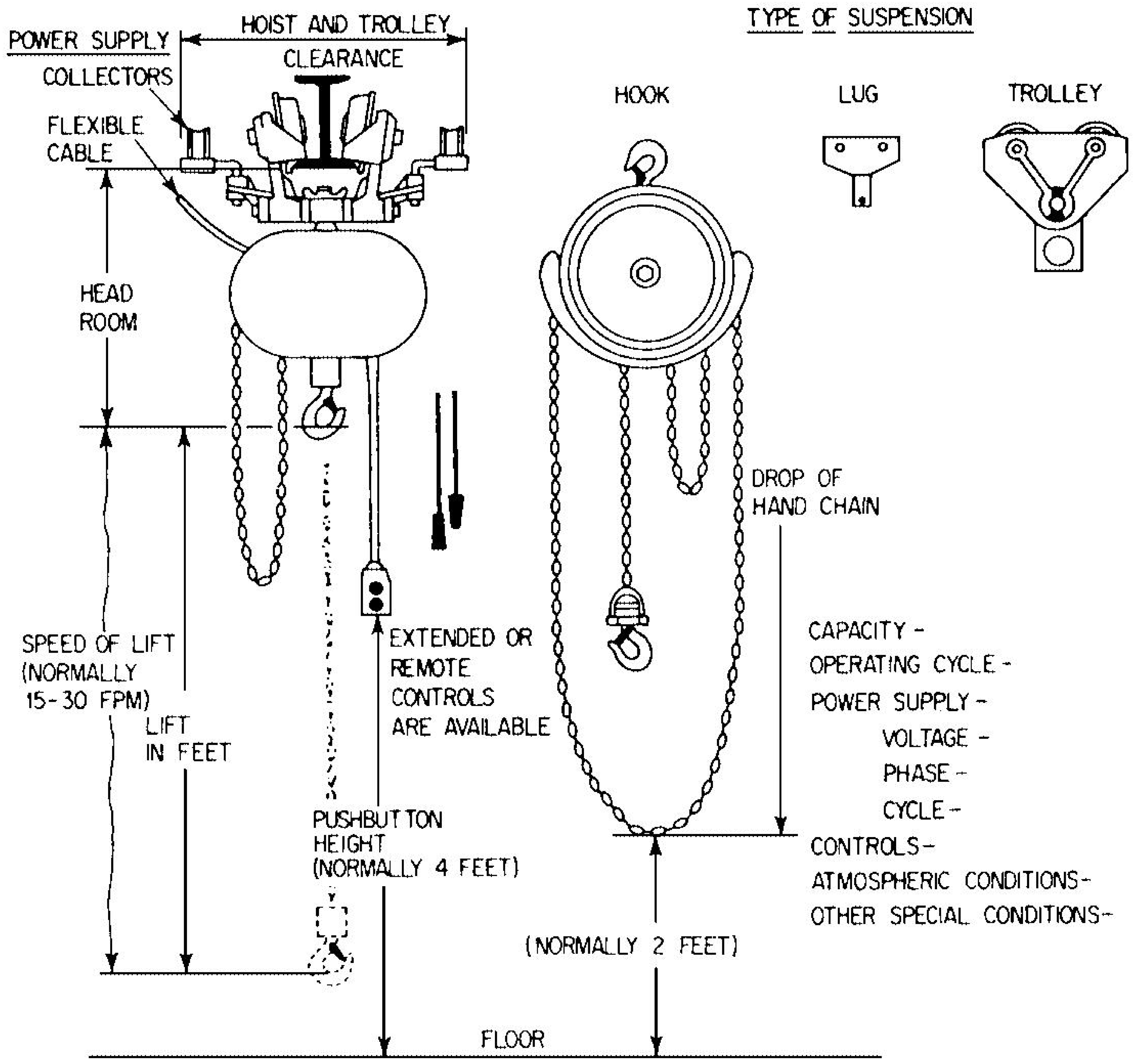


FIGURE 6.14 Hoist installation check diagram.

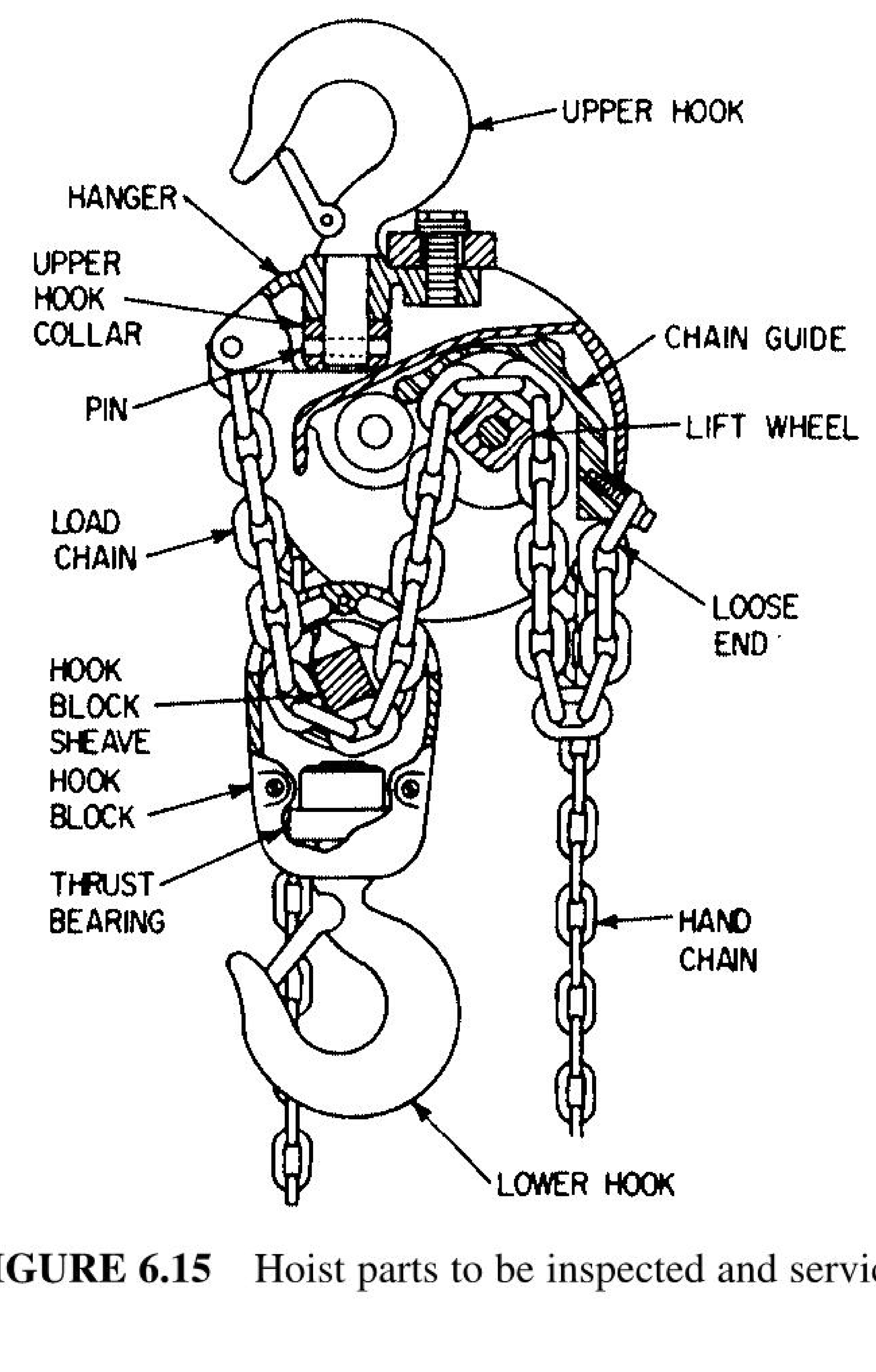


FIGURE 6.15 Hoist parts to be inspected and serviced.

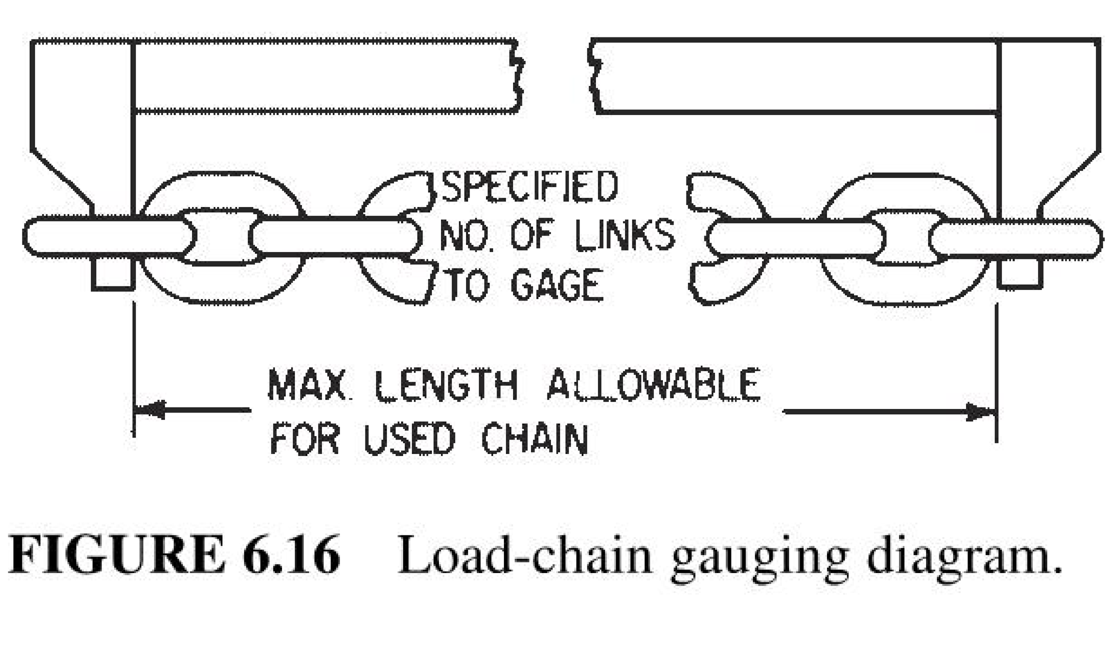


FIGURE 6.16 Load-chain gauging diagram.

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CHAPTER 7
BELT DRIVES

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GENERAL

Properly designed, correctly installed, and adequately tensioned belt drives give dependable power transmission with a minimum of maintenance. This chapter contains basic physical dimensions of V-belts and sheaves, synchronous belts and sprockets, synchronous application guidelines, installation and maintenance suggestions, and a troubleshooting guide. For actual design of belt drives, manufacturers' design manuals should be consulted.

V-BELT TYPES AND NOMINAL DIMENSIONS

Most V-belt drives used in industrial applications fall into two categories: heavy duty (industrial) and light duty (fractional horsepower). There are primarily two types of industrial belts: the classic cross sections (A, B, C, and D), which have been used for decades, and the *narrow* cross sections (3V, 5V, and 8V), which are relatively new. Most of the sections are available in banded (wrapped) and molded notch (cog) constructions. The banded belt has a fabric cover which completely encloses the exterior of the belt. The molded notch belt is manufactured with notches on the inside circumference of the belt (Figs. 7.1 and 7.2). The notches allow the belt to be more flexible in bending than the banded belt and

FIGURE 7.1 Typical narrow cross-sectional V-belt, banded construction.

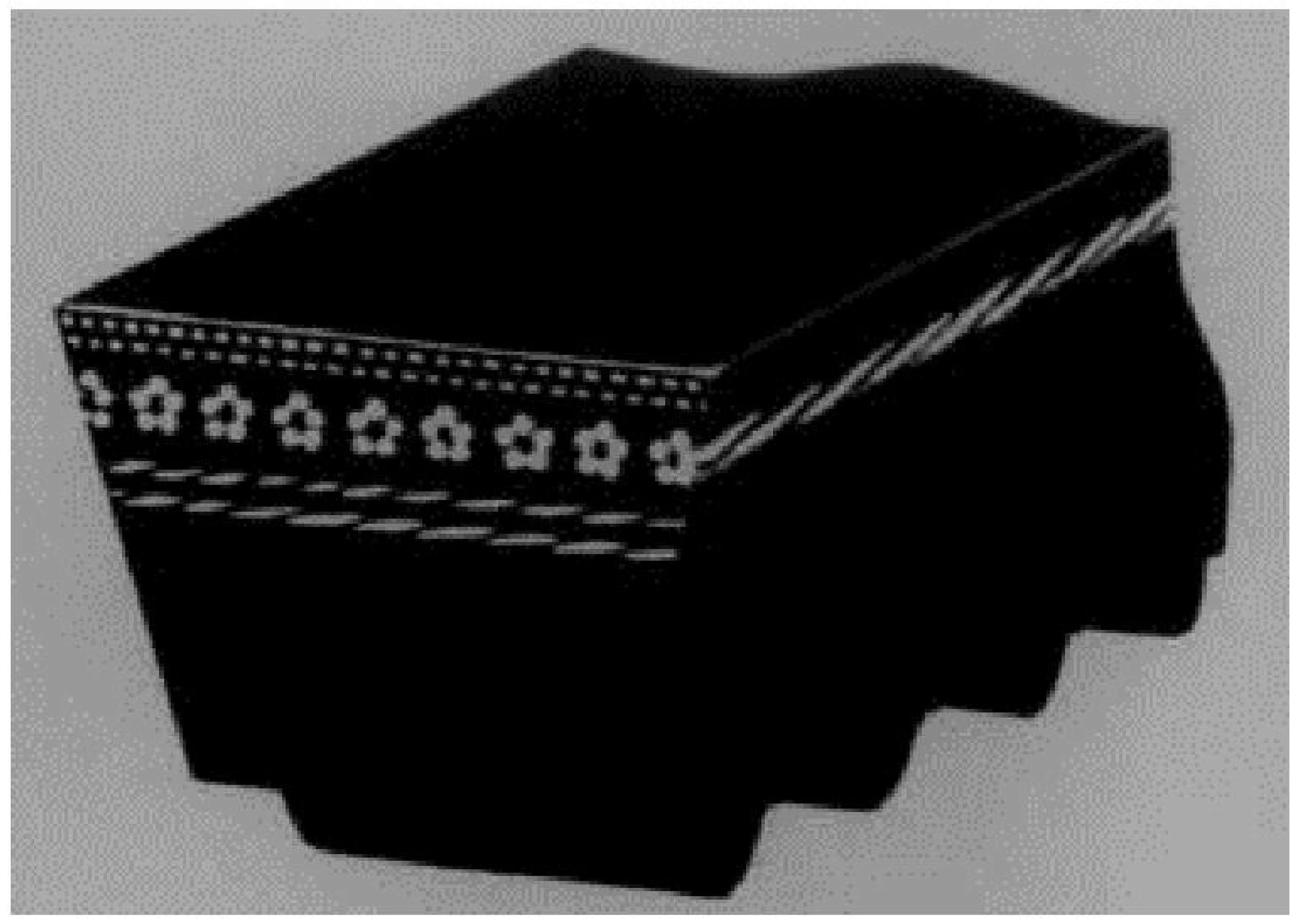

FIGURE 7.2 Typical classic cross-sectional V-belt molded notch construction.

may therefore be used on drives where smaller sheaves are required. (Molded notch construction belts are usually designated with an *X* after the section letter. A 3V molded notch belt would be designated 3VX.)

Both the classic and narrow section belts are available as joined belts. A *joined belt* consists of two or more individual belts fastened together with a layer of fabric across the tops of the belts. Joined belts help to prevent belt turnover and maintain belt stability in shock-loaded applications and on drives with long center distances. The horsepower rating of a joined belt is equivalent to that of an equal number of individual belts.

Fractional-horsepower belts are used most often on drives transmitting less than 1 horsepower. Consequently, they are generally not used in applications requiring multiple belts and are not length-matched out of stock. Length matching will be discussed in the next section. Fractional-horsepower belts are available in the following sections: 2L, 3L, 4L, and 5L.

Any V-belt is completely specified by its cross section and length. Nominal dimensions are shown in Figs. 7.3 to 7.5 as an aid in identifying belt cross sections. Other V-belts that are available include the V-ribbed belts (for use in high-speed applications and small-diameter sheaves) and the wide-range variable-speed belts (for use with special sheaves where the belt drive is used for speed variation between the driver and driven shafts). Refer to manufacturers' design manuals for details on these belts.

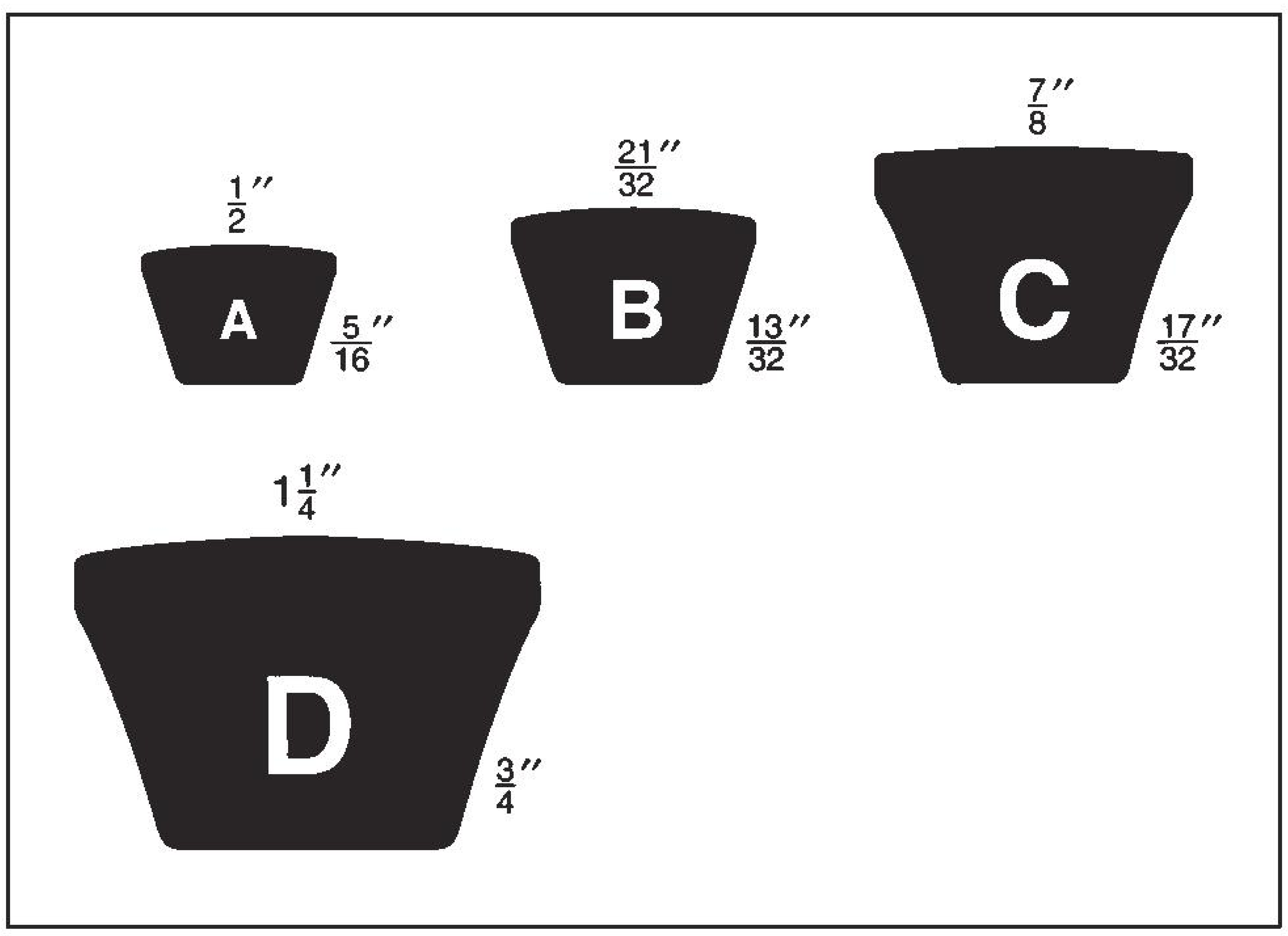

FIGURE 7.3 Classic section belts.

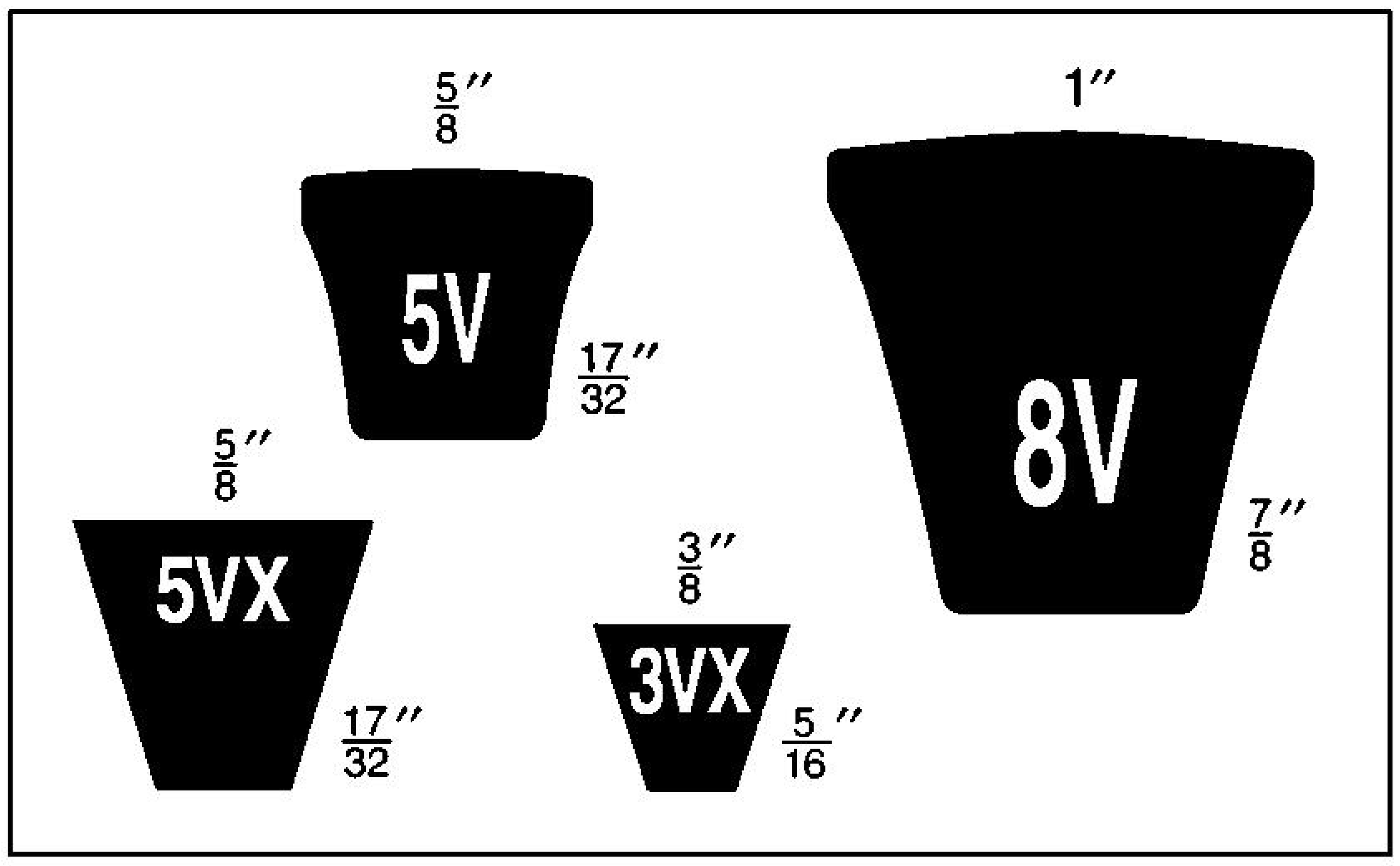

FIGURE 7.4 Narrow section belts.

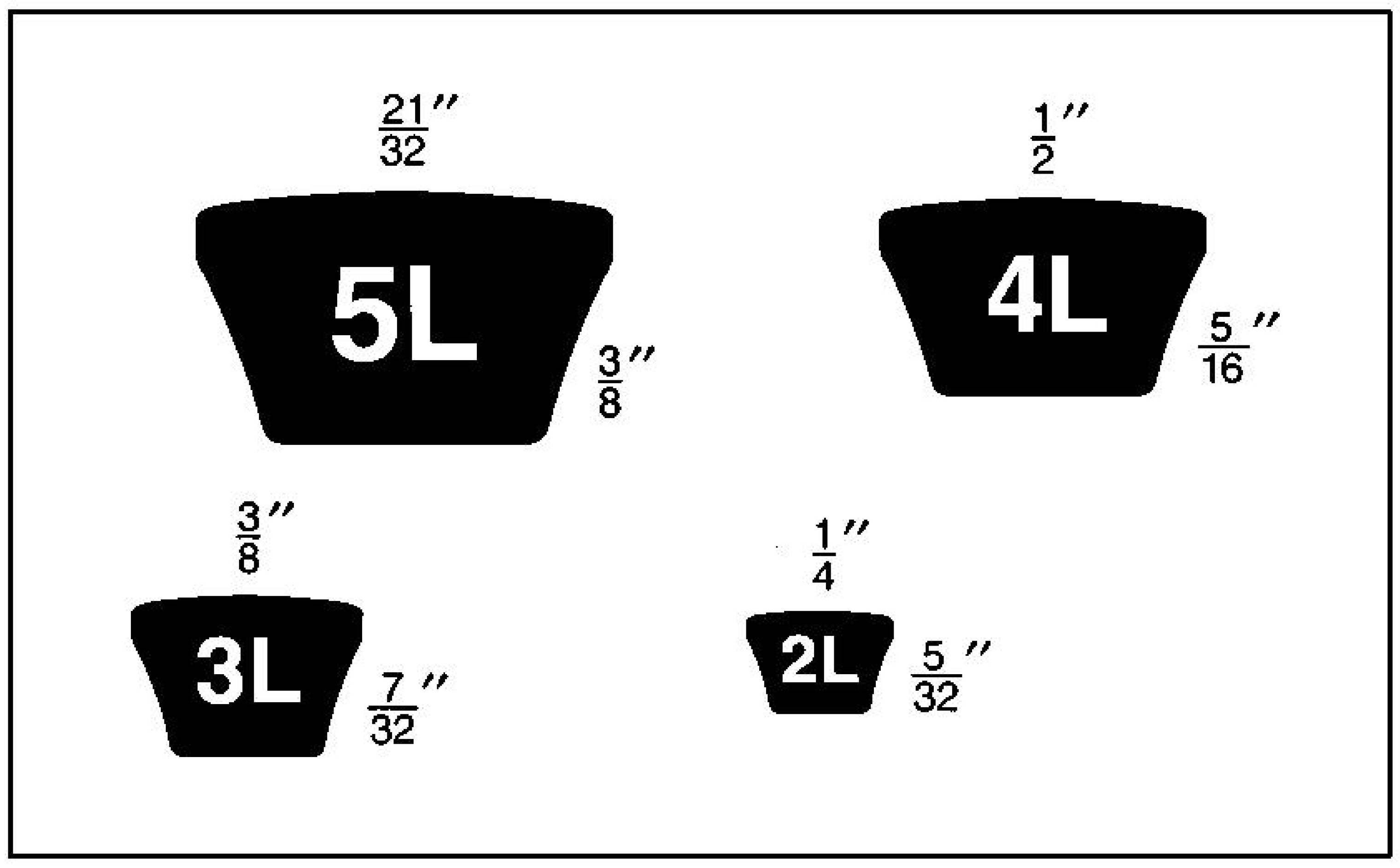

FIGURE 7.5 Fractional-horsepower belts.

V-Belt Length

There are three distinctly different ways of measuring V-belt length: outside circumference (OC), datum length (DL), and effective length (EL). The *outside circumference* is measured by wrapping a tape measure around the outside surface of the belt. This method is useful for obtaining nominal dimensions but does not give a truly accurate belt-length measurement. *Datum length* is a recent designation adopted by all belt manufacturers in order to retain standard belt and sheave designations while more accurately reflecting the changes that have occurred in belt pitch length and pitch-line location within the belt (*pitch length* is the length of the neutral axis of the belt).

Effective length is measured on a length-inspection machine. The machine consists of two parallel shafts on movable centers with a scale to accurately measure the center distance (see Fig. 7.6). Inspection sheaves of equal diameter and grooved in accordance with industry standards are mounted to these shafts. A belt is mounted on these sheaves and tensioned to a specified force. The belt is

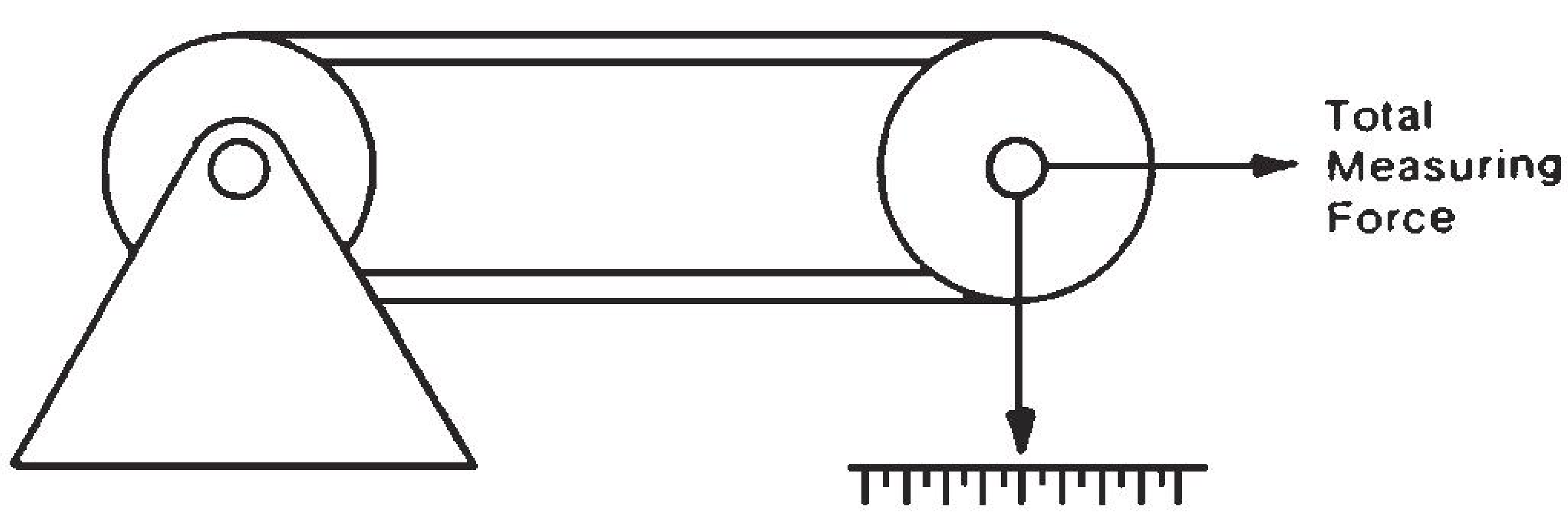

FIGURE 7.6 Schematic of a V-belt measuring fixture.

rotated through at least three complete revolutions to ensure that the tension is equalized around the belt and that the belt is seated in the grooves. The *effective length* is defined as the measured center distance plus the outside circumference of one of the inspection sheaves. This measurement method accounts for the modulus (stretchability) and dimensional variations among belts with the same cross section. All manufacturers use this designation. To appreciate why this method of measurement is important, consider the following examples:

1. Two belts are both identified as a B105. They both have the same outside circumference; however, one belt has a slightly narrower top width than the other. When the drive is tensioned, the belt with the narrower top width will fit deeper into the sheave groove than the wider top width belt and will therefore have a longer effective length.
2. Two belts are both identified as a B105. They both have the same outside circumference and physical dimensions; however, one is manufactured with a higher-modulus tensile cord than the other. When the drive is tensioned, the belt with the lower-modulus cord will elongate more, resulting in a longer effective length.

The effective length designation ensures that regardless of belt modulus and dimensional variations, two identically labeled belts will exhibit the same length (within a tolerance) on a tensioned drive. Accurate belt-length measurements are important in applications which require multiple belts. In these applications, significant differences in belt length will result in unequal load distribution between the belts and subsequent premature belt failure. Belts that are used in these applications must be length-matched. Belt matching is a system in which deviations from ideal effective length are recorded as match numbers. For a given belt-length range, belts to be used on a multiple-belt drive must all fall within a certain match-number range (length tolerance). Most manufacturers have moved away from this system and now produce belts within the matching tolerances, thus making belt matching by match numbers unnecessary. The standard matching tolerances are listed in Table 7.1.

V-Belt Sheaves

Standard V-belt sheaves are manufactured for all the listed belt sections. V-belt sheaves have exact rather than nominal dimensions. In 1988, the industry adopted the datum system as the standard for specifying classic (A, B, C, and D section) V- belt sheaves. All classic sheaves are now identified by datum diameter rather than by pitch diameter. The physical dimensions of the sheaves have not changed. An old 10.0-in. pitch-diameter sheave is directly replaced by a 10.0-in. datum-diameter sheave. Datum diameters and datum lengths should be used to calculate center distances and belt datum length. The often critical speed-ratio and horsepower-rating calculations are now more accurately based on the modern pitch diameter. Industrial sheave dimensions are shown in Fig. 7.7 and Tables 7.2 and 7.3. Deep-groove sheaves are used to increase belt stability in applications where extreme vibration, belt twist, or extreme misalignment are encountered. Specifications for these sheaves are also shown in Tables 7.2 and 7.3. (Joined belts cannot be used with deep-groove sheaves.) Fractional-horsepower sheave dimensions are shown in Fig. 7.8 and Table 7.4.

TABLE 7.1 Standard Matching Tolerances for Industrial V-Belts

A, B, C, D		3V, 5V, 8V	
Standard length designation	Matching tolerance, in.	Standard length designation	Matching tolerance, in.
26–60	0.15	250–670	0.15
68–144	0.30	710–1500	0.30
158–240	0.45	1600–2500	0.45
270–360	0.60	2650–3750	0.60
390–480	0.75	4000–5000	0.75
540–660	0.90		

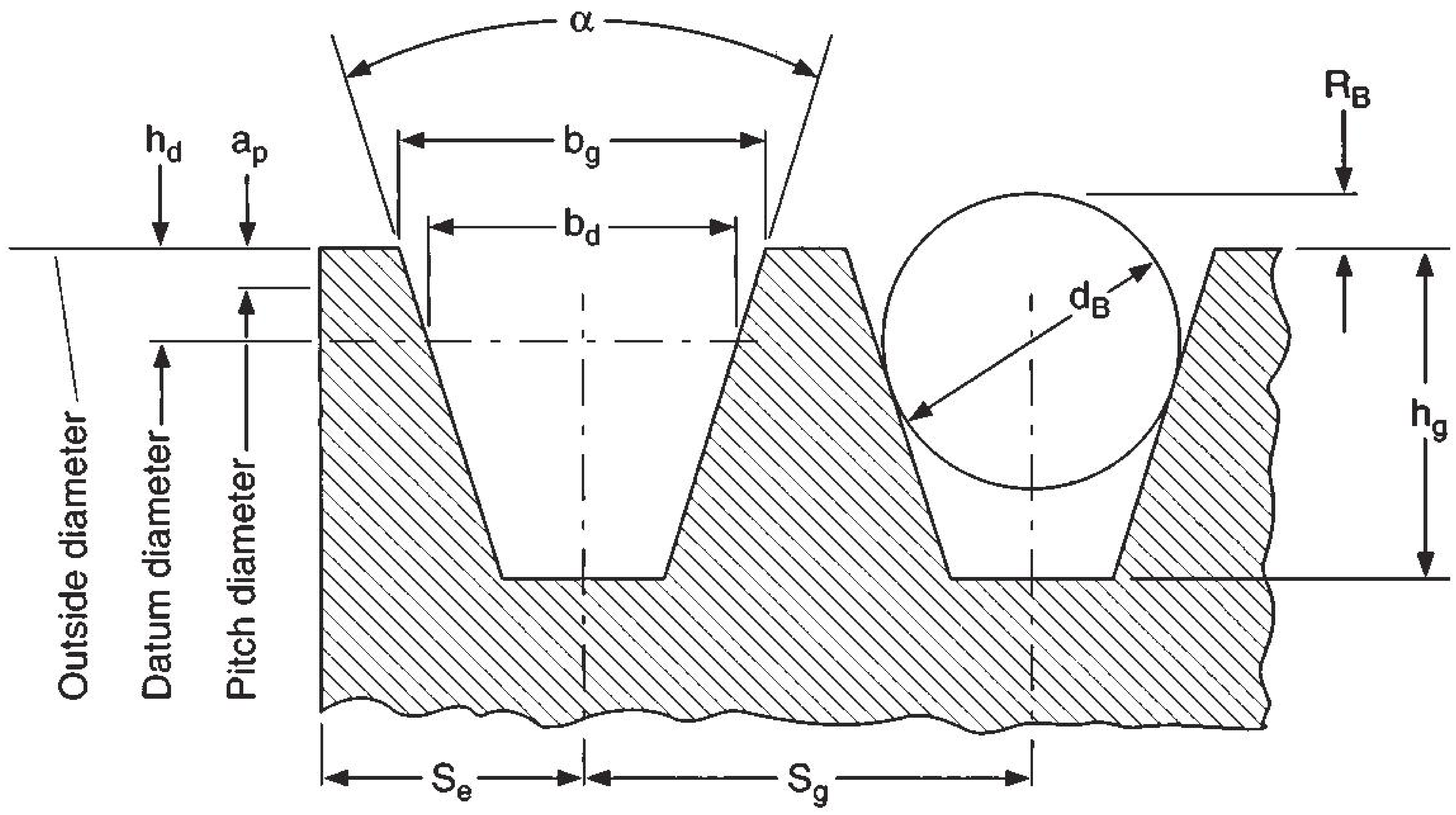

FIGURE 7.7 Groove dimensions for standard industrial V-belt sheaves.

Minimum Sheave Diameters

When a V-belt bends around a sheave, compressive forces develop in the bottom of the belt, and tension forces develop in the top of the belt. The magnitude of these forces is a function of the diameter of the sheave and the cross section of the belt and has a significant impact on belt life. These forces increase with smaller diameters and larger cross sections. Therefore, minimum recommended diameters were developed for each belt cross section. They are listed in Table 7.5. Using sheaves that are below the recommended minimum will always result in reduced belt life.

Belt drives are often used in applications where the driver is an electric motor. The motor bearings are specified assuming some maximum overhung load. As the motor shaft sheave diameter decreases, the overhung load increases. Therefore, the National Electrical Manufacturers Association (NEMA) developed a standard which specifies minimum sheave diameters for all the common motor sizes. Using sheaves that are below these minimums could result in shortened motor bearing life. These minimums are listed in Table 7.6.

SYNCHRONOUS BELTS

Synchronous belts are toothed belts in which power is transmitted through positive engagement between belt tooth and pulley or sprocket groove rather than by the wedging friction of V-belts. The positive drive characteristics of these belts provide exact synchronization between driver and driven shafts and also increase power transmission efficiency. Other advantages of synchronous belts over other modes of power transmission include a wider load/speed range, lower maintenance, increased wear resistance, and a smaller amount of required takeup.

There are three types of synchronous belts: trapezoidal, curvilinear, and modified curvilinear. See Fig. 7.9. The most common belt in use today is the trapezoidal belt. It is also the only type for which industry standards have been developed. Therefore, all belt and pulley dimensions in this chapter will refer to trapezoidal belts. Details on other synchronous belts and sprockets may be obtained from individual manufacturers' catalogs.

The curvilinear tooth form was developed in order to provide increased capacity over trapezoidal belts. Innovations in tooth profile design and increased tooth depth resulted in better stress distribution in the belt teeth and increased resistance to tooth ratcheting. The curvilinear belt consequently has a higher horsepower capacity than does a comparable trapezoidal belt. The third type of synchronous belt is the modified curvilinear. The modified curvilinear profile is a refinement of the curvilinear profile in which the tooth shape, depth, and materials were optimized to maximize

TABLE 7.2 Groove Dimensions for Standard and Deep-Groove Narrow-Section Sheaves, in.

		Standard groove dimensions								Design factors	
Cross section	Standard groove effective diameter	Groove angle ±0.25 degrees	b_g ±0.005	b_e ref.	h_g min.	R_B min.	d_B ±0.0005	S_g ±0.015	S_e	Minimum recommended effective diameter	$2a_p$ $2h_e$
	Up through 3.49	36				0.181					
	Over 3.49 to and										
3V	including 6.00	38	0.350	0.350	0.340	0.183	0.3438	0.406	0.344	3V 2.65	0
3VX	Over 6.00 to and					0.186			+0.094	3VX 2.20	
	including 12.00	40							−0.031		
	Over 12.00	42				0.188					
	Up through 9.99	38				0.329					
5V	Over 9.99 to and								0.500	5V 7.10	0
5VX	including 16.00	40	0.600	0.600	0.590	0.332	0.5938	0.688	+0.125	5VX 4.40	
									−0.047		
	Over 16.00	42				0.336					
	Up through 15.99	38				0.575					
	Over 15.99 to and								0.750	12.50	0
8V	including 22.40	40	1.000	1.000	1.990	0.580	1.000	1.125	+0.250		
									−0.062		
	Over 22.40	42				0.585					

		Deep groove dimensions								Design factors	
Cross section	Deep groove effective diameter	Groove angle ±0.25 degrees	b_g ±0.005	b_e ref.	h_g min.	R_B min.	d_B ±0.0005	S_g ±0.015	S_e	Minimum recommended effective diameter	$2a_p$ $2h_e$
	Up through 3.49	36	0.421			0.070					
	Over 3.49 to and										
3V	including 6.00	38	0.425	0.350	0.449	0.073	0.3438	0.500	0.375	3V 2.65	0.218
3VX	Over 6.00 to and	40	0.429			0.076			+0.094	3VX 2.20	
	including 12.00								−0.031		
	Over 12.00	42	0.434			0.078					
	Up through 9.99	38	0.710			0.168					
5V	Over 9.99 to and								0.562	5V 7.10	0.320
5VX	including 16.00	40	0.716	0.600	0.750	0.172	0.5938	0.812	+0.125	5VX 4.40	
									−0.047		
	Over 16.00	42	0.723			0.175					
	Up through 15.99	38	1.180			0.312					
	Over 15.99 to and								0.844	12.50	0.524
8V	including 22.40	40	1.191	1.000	1.262	0.316	1.0000	1.312	+0.250		
									−0.062		
	Over 22.40	42	1.201			0.321					

TABLE 7.3 Groove Dimensions for Standard and Deep-Groove, Classical-Section Sheaves, in.

		a. Standard groove dimensions									Design factors	
Cross section	Datum diameter range	α Groove angle ±0.33	b_d ref.	b_g	h_g min.	$2h_d$ ref.	R_B min.	d_B ±0.0005	S_g ±0.025	S_e	Minimum recommended datum diameter	$2a_p$
	Up through 5.4	34		0.494			0.148	0.4375		+0.090	A 3.0	
A, AX			0.418	±0.005	0.460	0.250			0.625	0.375		0
	Over 5.4	38		0.504			0.149	(7/16)		−0.062	AX 2.2	
	Up through 7.0	34		0.637			0.189	0.5625		+0.120	B 54	
B, BX			0.530	±0.006	0.550	0.350			0.750	0.500		0
	Over 7.0	38		0.650			0.190	(9/16)		−0.065	BX 4.0	
	Up through 7.4[1]	34		0.612		0.634	0.230				A 3.6(1)	
A, AX belt[4]				±0.006		(3)						0.37
	Over 7.4	38		0.625		0.602	0.226			+0.120	AX 2.8	
			(2)					0.5625				
			0.508		0.612			(9/16)	0.750	0.500		
B, BX belt[4]	Up through 7.4[1]	34		0.612		0.333	0.230			−0.065	B 5.7(1)	
				±0.006		(3)						−0.01
	Over 7.4	38		0.625		0.334	0.226				BX 4.3	
	Up through 7.99	34		0.879			0.274					
	Over 7.99	36		0.887 ±0.007			0.276	0.7812		+0.160	C 9.0	
C, CX	to and		0.757		0.750	0.400			1.000	0.688		0
	including 12.0							(25/32)		−0.070	CX 6.8	
	Over 12.0	38		0.895			0.277					
	Up through 12.99	34		1.259			0.410					
	Over 12.99	36		1.271 ±0.008			0.410	1.1250		+0.220		
D	to and		1.076		1.020	0.600			1.438	0.875	13.0	0
	including 17.0							(1 1/8)		−0.080		
	Over 17.0	38		1.283			0.411					

(1) Diameters shown for combination grooves are outside diameters. A specific datum diameter does not exist for either A or B belts in combination grooves.
(2) The b_d value shown for combination grooves is the "constant width" point but does not represent a datum width for either A or B belts ($2h_d$ = 0.340 reference).
(3) $2h_d$ values for combination groove are calculated based on b_d for A and B grooves.
(4) A, AX & B, BX Combin.

TABLE 7.3 Groove Dimensions for Standard and Deep-Groove, Classical-Section Sheaves, in. (*Continued*)

		b. Deep groove dimensions										Design factors	
Cross section	Datum diameter range[1]	α Groove angle ±0.33	b_d ref.		b_g	h_g min.	$2h_d$ ref.	R_B min.	d_B ±0.0005	S_g ±0.025	S_e	Minimum recommended datum diameter	$2a_p$
	Up through 7.0	34		0.747			0.007	0.5625		+0.120	B 5.4		
B, BX			0.530		±0.006	0.730	0.710			0.875	0.562		0.36
	Over 7.0	38		0.774				0.008	(9/16)		−0.065	BX 4.0	
	Up through 7.99	34		1.066				−0.035					
	Over 7.99			1.085	±0.007			−0.032	0.7812		+0.160	C 9.0	
C, CX	to and		0.757			1.055	1.010			1.250	0.812		0.61
	including 12.0	36							(25/32)		−0.070	CX 6.8	
	Over 12.0	38		1.105				−0.031					
	Up through 12.99	34		1.513				−0.010					
D	Over 12.99			1.541	±0.008			−0.009	1.1250		+0.220		
	to and		1.076			1.435	1.430			1.750	1.062	13.0	0.83
	including 17.0	36							(1 1/8)		−0.080		
	Over 17.0	38		1.569				−0.008					

(1) The A/AX, B/BX combination groove should be used when deep grooves are required for A or AX belts.

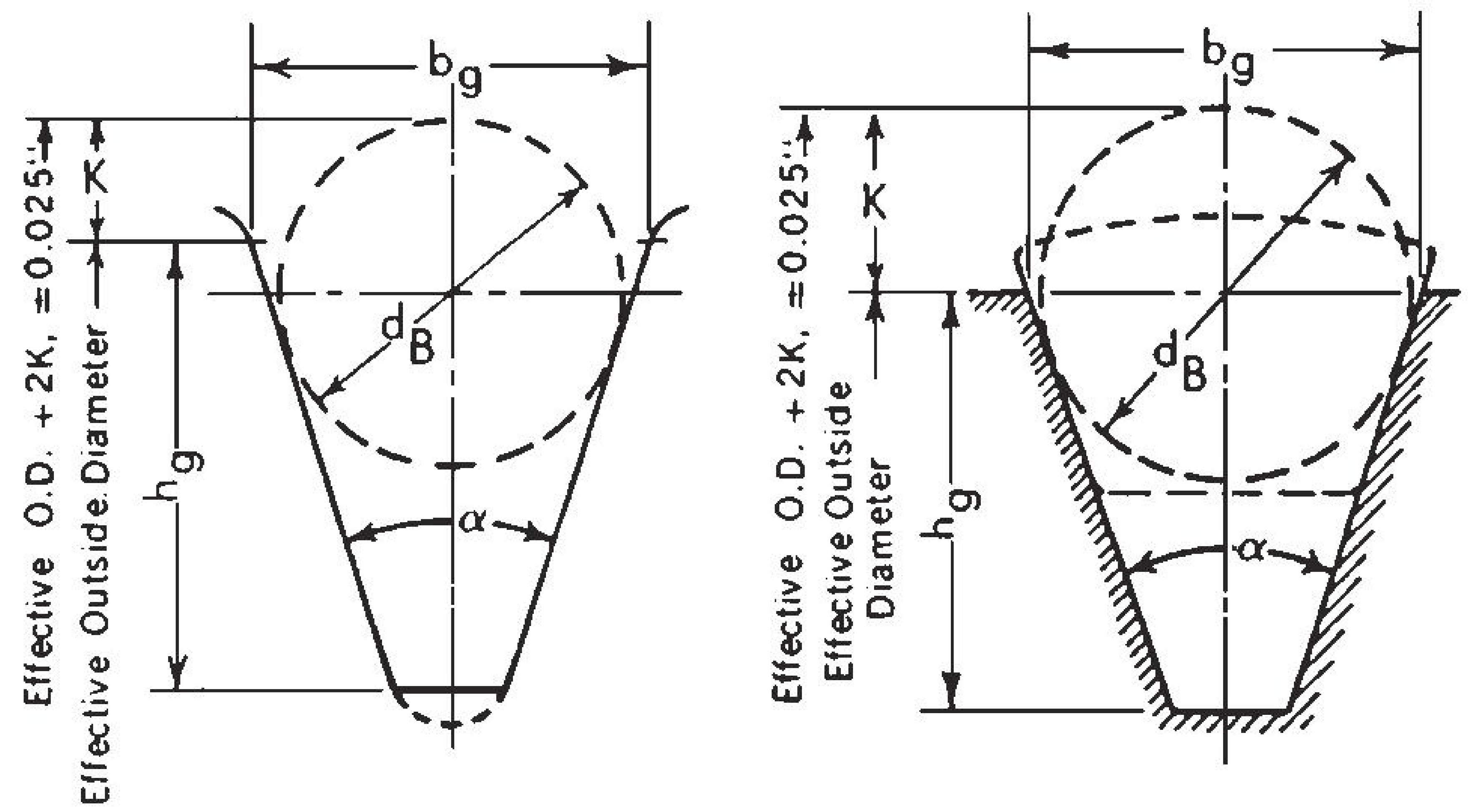

FIGURE 7.8 Groove dimensions for standard light-duty V-belts sheaves.

TABLE 7.4 Groove Dimensions for Standard Light-Duty Sheaves

Belt section	Effective outside diameter range, in.	α Groove angle, ±0°20' degrees	d_g Ball diameter, ±0.0005 in.	$2K$ in.	b_g Reference, in.	h_g Minimum, in.	$2a$ in.
	Less than 1.50	32		0.176			
2L	1.50 to 1.99	34	0.2188	0.182	0.240	0.250	0.04
	2.00 to 2.50	36		0.188			
	Over 2.50	38		0.194			
	Less than 2.20	32		0.177			
3L	2.20 to 3.19	34	0.3125	0.191	0.364	0.406	0.06
	3.20 to 4.20	36		0.203			
	Over 4.20	38		0.215			
	Less than 2.65	30		0.299			
4L	2.65 to 3.24	32	0.4375	0.316	0.490	0.490	0.10
	3.25 to 5.65	34		0.331			
	Over 5.65	38		0.358			
	Less than 3.95	30		0.385			
5L	3.95 to 4.94	32	0.5625	0.406	0.630	0.580	0.16
	4.95 to 7.35	34		0.426			
	Over 7.35	38		0.461			

load/life capacity and other drive characteristics. With these higher-capacity drive systems, the pulleys are generally referred to as *sprockets.*

A synchronous belt is identified by the profile type, tooth pitch (distance between adjacent teeth), pitch length, and top width. Trapezoidal belts are designated in the following manner: 240H150. The first set of digits designates pitch length in tenths of an inch, for this example, 24.0 in. The letter indicates the belt pitch, in this case, H or ½ in. The last set of digits specifies nominal belt width in hundredths of an inch, for this example, 1.50 in. The tooth profile of standard trapezoidal belts is shown in Fig. 7.10. Trapezoidal belts are available in a single-sided configuration and, for serpentine applications requiring shaft rotations in opposite directions, in a double-sided configuration. The double-sided configuration is not available for all pitch sizes. Nominal tooth dimensions for the standard belt sections are shown in Table 7.7. The available belt widths and pitch lengths are shown in Tables 7.8 and 7.9, respectively.

TABLE 7.5 Minimum Recommended Sheave Diameters for Various Cross Sections

Cross section	Minimum sheave diameter
3V	2.65
*3VX	2.2
5V	7.1
*5VX	4.4
8V	12.5
A	3.0
*AX	2.2
B	5.4
*BX	4.0
C	9.0
*CX	6.8
D	13.0
2L	0.8
3L	1.5
4L	2.5
5L	3.5

*These are recommendations. There are no industry standards for these cross sections.

SYNCHRONOUS PULLEYS

Synchronous pulleys are identified by the number of grooves, pitch, and the nominal pulley face width in hundredths of an inch. For trapezoidal pulleys (Figs. 7.11 to 7.13), a 36L100 refers to a 36 groove, L pitch, 1.00-in. nominal face width pulley. The generating tool rack form described in Table 7.10 and Fig. 7.11 defines the pulley groove profile. Standard pulley diameters and face widths are given in Tables 7.11 and 7.12.

SYNCHRONOUS APPLICATION GUIDELINES

The synchronous method of power transmission is significantly different than that of V-belts. Therefore, there are unique aspects of synchronous belt application that should be considered.

1. Synchronous-belt horsepower ratings are based on having at least six teeth in mesh with each loaded sprocket. The belt must be derated for any situation in which this cannot be achieved.
2. Do not mix belt and sprocket tooth profiles. The result will always be decreased life and performance.
3. Use synchronous belts to increase drive efficiency. A well-maintained V-belt drive is 93 to 97 percent efficient. A well-maintained synchronous belt drive is 98 to 99 percent efficient.
4. The precision of shaft synchronization varies between different types of synchronous belts. Contact individual manufacturers for specifics.
5. Since synchronous belts are not wedged in grooves as V-belts are, the belt tends to track across the sprocket. Therefore, at least one sprocket in a two-sprocket system must be flanged.
6. Alignment is more critical to satisfactory synchronous-belt performance than it is to satisfactory V-belt performance. Do not use synchronous belts where there is severe inherent misalignment.

TABLE 7.6 Electric Motor Frames and Minimum Diameters

Frame no.	Shaft diameter, in.	Horsepower at synchronous speed, rpm 3600 (3450)*	1800 (1750)*	1200 (1160)*	900 (870)*	Narrow-section V belts, minimum outside diameter, in.	Classical section V-belts, minimum datum diameter, in.
143T	0.875	1½	1	¾	½	2.2	2.2
145T	0.875	2–3	1½–2	1	¾	2.4	2.4
182T		3	3	1½	1	2.4	2.4
182T	1.125	5	—	—	—	2.4	2.6
184T		—	—	2	1½	2.4	2.4
184T	1.125	5	—	—	—	2.4	2.6
184T		7½	5	—	—	3.0	3.0
213T	1.375	7½–10	7½	3	2	3.0	3.0
215T		10	—	5	3	3.0	3.0
215T	1.375	15	10	—	—	3.8	3.8
254T		15	—	7½	5	3.8	3.8
254T	1.625	20	15	—	—	4.4	4.4
256T		20–25	—	10	7½	4.4	4.4
256T	1.625	—	20	—	—	4.4	4.6
284T		—	—	15	10	4.4	4.6
284T	1.875	—	25	—	—	4.4	5.0
286T	1.875	—	30	20	15	5.2	5.4
324T	2.125	—	40	25	20	6.0	6.0
326T	2.125	—	50	30	25	6.8	6.8
364T		—	—	40	30	6.8	6.8
364T	2.375	—	60	—	—	7.4	7.4
365T		—	—	50	40	8.2	8.2
365T	2.375	—	75	—	—	8.6	9.0
404T		—	—	60	—	8.0	9.0
404T	2.875	—	—	—	50	8.4	9.0
404T		—	100	—	—	8.6	10.0
405T		—	—	75	60	10.0	10.0
405T	2.875	—	100	—	—	8.6	10.0
405T		—	125	—	—	10.5	11.5
444T		—	—	100	—	10.0	11.0
444T		—	—	—	75	9.5	10.5
444T	3.375	—	125	—	—	9.5	11.0
444T		—	150	—	—	10.5	—
445T		—	—	125	—	12.0	12.5
445T		—	—	—	100	12.0	12.5
445T	3.375	—	150	—	—	10.5	—
445T		—	200	—	—	13.2	—

*Approximate full load speeds.

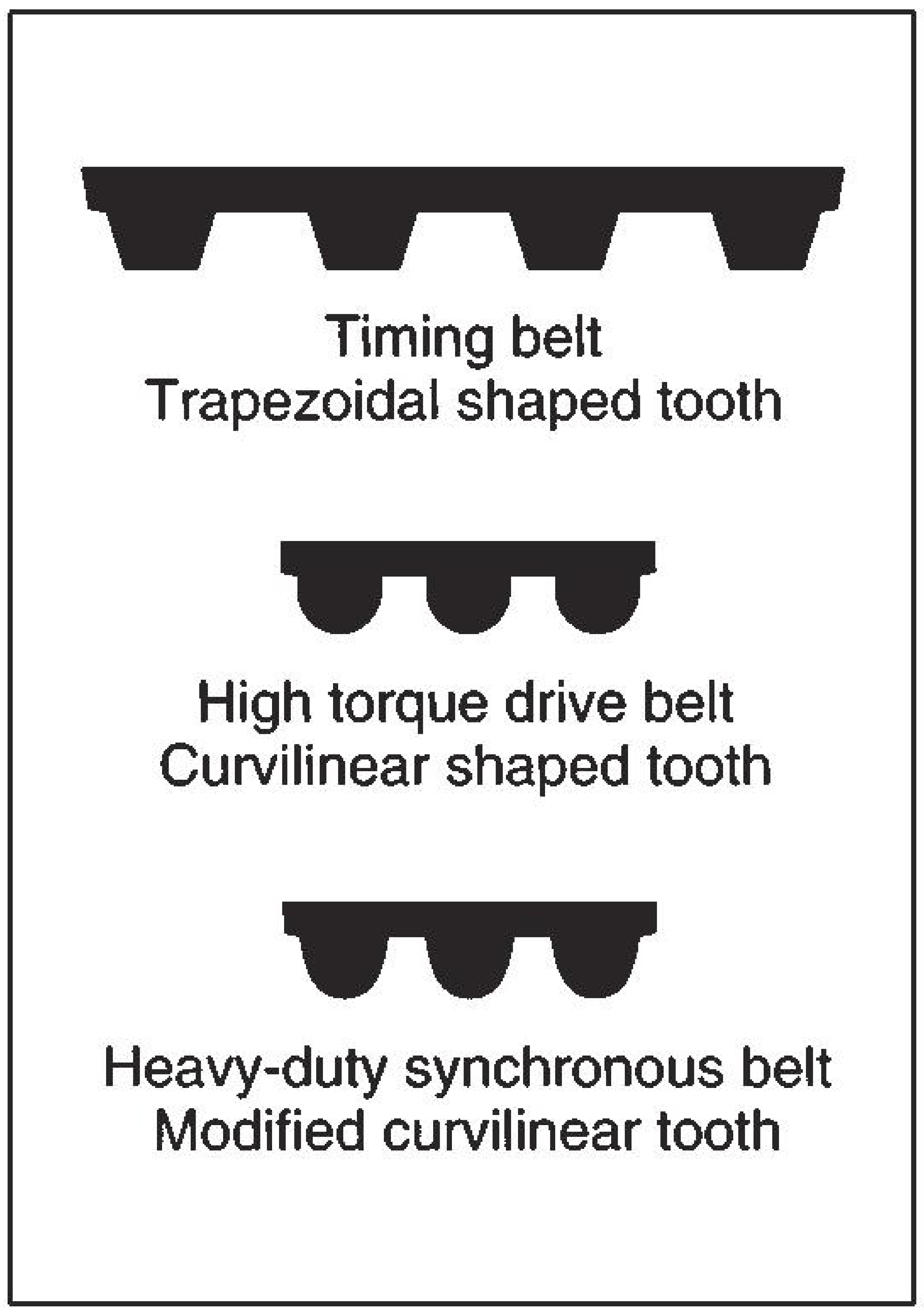

FIGURE 7.9 Synchronous belts.

7. Tensioning is critical on synchronous drives because of the phenomenon of self-generated tension. If a synchronous belt is undertensioned, it will tend to climb out of the sprocket grooves under load. This will accelerate belt tooth wear and in extreme cases may lead to belt ratcheting. Spring-loaded idlers are not recommended on synchronous drives due to this phenomenon.
8. Synchronous belts should not be used in multiple-belt configurations. They do not seat into grooves as V-belts do; therefore, load sharing between two or more synchronous belts is highly unlikely.

Awareness of these guidelines will optimize performance and minimize problems with synchronous drives.

MAINTENANCE

A properly maintained belt drive will run trouble-free for a long time. In order to maximize the performance and life of belt drives, it is important to maintain and develop a maintenance program. The frequency of maintenance is influenced by many factors, including drive speed, duty cycle, critical nature of the equipment, temperature extremes, environmental factors, and accessibility of the equipment. Drives operating under severe conditions of high speed, heavy loads, temperature extremes, and operation on critical applications require more frequent inspection. Following are some suggestions for frequency of maintenance. In this section, the term *pulley* will refer to both V-belt sheaves and synchronous pulleys and sprockets.

1. On critical drives, a quick visual and hearing inspection should be performed every 1 to 2 weeks.
2. On normal drives, a quick visual and hearing inspection should be performed once per month.
3. A drive shutdown to thoroughly inspect belts, pulleys, and other components should be performed every 3 to 6 months.

A visual and hearing inspection consists of the following three items:

1. Look and listen for unusual noise and vibration while observing the drive. A well-designed and well-maintained drive will operate quietly and smoothly.

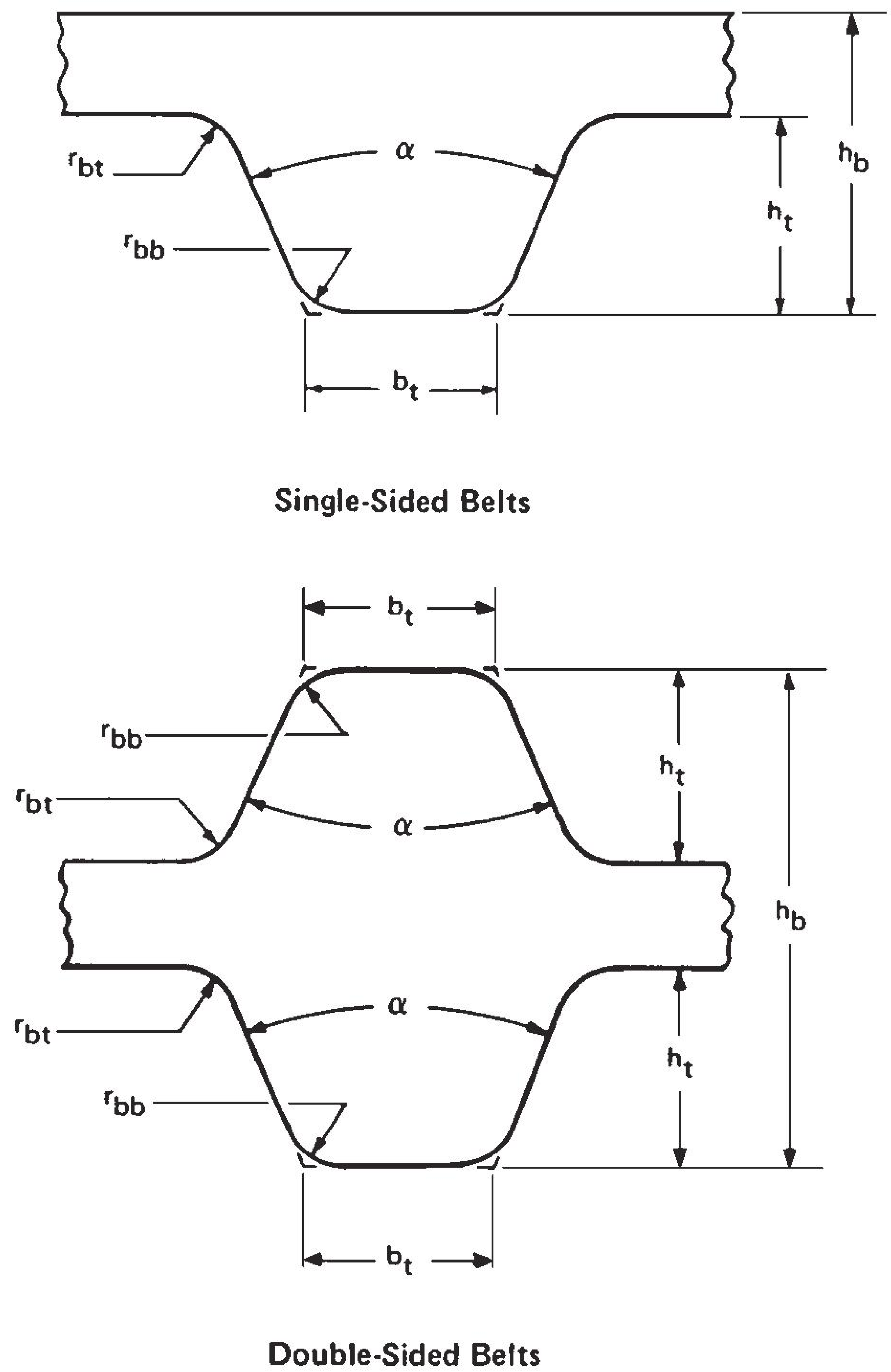

FIGURE 7.10 Trapezoidal-belt tooth dimensions.

TABLE 7.7 Standard Trapezoidal Belt Sections and Nominal Tooth Dimensions, in.

Tooth angle belt section	Pitch	Degrees	h_b	h_t	b_t	r_{bb}	r_{bt}
Single sided:							
MXL	0.080	40	0.045	0.020	0.030	0.005	0.005
XL	0.200	50	0.090	0.050	0.054	0.015	0.015
L	0.375	40	0.14	0.075	0.128	0.020	0.020
H	0.500	40	0.16	0.090	0.175	0.040	0.040
XH	0.875	40	0.44	0.250	0.313	0.047	0.062
XXH	1.250	40	0.62	0.375	0.477	0.060	0.090
Double sided:							
DXL	0.200	50	0.120	0.050	0.054	0.015	0.015
DL	0.375	40	0.180	0.075	0.128	0.020	0.020
DH	0.500	40	0.234	0.090	0.175	0.040	0.040

TABLE 7.8 Standard Trapezoidal Belt Widths and Tolerances, in.

	Standard belt widths		Tolerances on width for belt pitch lengths		
Belt section	Designation	Dimensions	Up to and including 33 in.	Over 33 in. up to and including 66 in.	Over 66 in.
	012	0.12			
MXL (0.080)	019	0.19	+0.02		
	025	0.25	−0.03		
	025	0.25			
XL (0.200)			+0.02		
			−0.03		
	037	0.38			
	050	0.50	+0.03	+0.03	
L (0.375)	075	0.75	−0.03	−0.05	
	100	1.00			
	075	0.75	+0.03	+0.03	+0.03
	100	1.00	−0.03	−0.05	−0.05
	150	1.50			
H (0.500)	200	2.00	+0.03	+0.05	+0.05
			−0.05	−0.05	−0.06
	300	3.00	+0.05	+0.06	+0.06
			−0.06	−0.06	−0.08
	200	2.00			
XH (0.875)	300	3.00	—	+0.19	+0.19
	400	4.00		−0.19	−0.19
	200	2.00			
XXH (1.250)	300	3.00			+0.19
	400	4.0			−0.19
	500	5.00			

2. Inspect the guard for looseness and damage. Make sure that it is clean. Any accumulation of foreign material on the guard acts as insulation and could cause excessive heat buildup in the drive.
3. Look for oil and grease dripping from the guard. These compounds will degrade the belt material if they contact the belt.

A thorough inspection requires that the drive be shut down. This procedure should include an inspection of the belt, the pulleys, the belt guard, belt tension, and associated drive components such as bearings, shafts, and takeup rails. Whenever a drive is inspected thoroughly, take the necessary safety precautions to avoid injury. Shut the power off, lock and tag the control box, and place all machine components in a neutral position.

The primary drive troubleshooting tool is belt inspection. Unusual belt wear is a symptom of possible drive problems. Make a mark on the belt, and inspect the entire circumference of the belt, checking for uneven belt wear, cracks, frayed covers, burn spots, and swelling. Also check for cracking of the undercord or notches or lost teeth on synchronous belts. Check the belt for excessive heat. A properly functioning belt drive operates at approximately 140°F. You should be able to comfortably hold the belt at this temperature. Belts that are hot to the touch indicate drive problems. Belts should be replaced if there is significant cracking, fraying, or loss of teeth in the case of synchronous belts.

The pulleys should be checked for wear and for proper mounting and alignment. Inspect pulleys for wear, nicks, and sharp edges. V-belt sheaves may be checked with plastic groove gauges, which

TABLE 7.9 Standard Trapezoidal Belt Pitch Lengths and Tolerances, in.

Belt length designation	Pitch length	Permissible deviation from standard length	Number of teeth for standard lengths					
			MXL (0.080)	XL (0.200)	L (0.375	H (0.500)	XH (0.875)	XXH (1.250)
36	3.600	±0.016	45					
40	4.000	±0.016	50					
44	4.400	±0.016	55					
48	4.800	±0.016	60					
56	5.600	±0.016	70					
60	6.000	±0.016	75	30				
64	6.400	±0.016	80					
70	7.000	±0.016		35				
72	7.200	±0.016	90					
80	8.000	±0.016	100	40				
88	8.800	±0.016	110					
90	9.000	±0.016		45				
100	10.000	±0.016	125	50				
110	11.000	±0.018		55				
112	11.200	±0.018	140					
120	12.000	±0.018		60				
124	12.375	±0.018			33			
124	12.400	±0.018	155					
130	13.000	±0.018		65				
140	14.000	±0.018	175	70				
150	15.000	±0.018		75	40			
160	16.000	±0.020	200	80				
170	17.000	±0.020		85				
180	18.000	±0.020	225	90				
187	18.750	±0.020			50			
190	19.000	±0.020		95				
200	20.000	±0.020	250	100				
210	21.000	±0.024		105	56			
220	22.000	±0.024		110				
225	22.500	±0.024			60			
230	23.000	±0.024		115				
240	24.000	±0.024		120	64	48		
250	25.000	±0.024		125				
255	25.500	±0.024			68			
260	26.000	±0.024		130				
270	27.000	±0.024			72	54		
285	28.500	±0.024			76			
300	30.000	±0.024			80	60		
322	32.250	±0.026			86			
330	33.000	±0.026				66		
345	34.500	±0.026			92			
360	36.000	±0.026				72		
367	36.750	±0.026			98			
390	39.000	±0.026			104	78		
420	42.000	±0.030			112	84		
450	45.000	±0.030			120	90		
480	48.000	±0.030			128	96		
507	50.750	±0.032					58	
510	51.000	±0.032			136	102		

TABLE 7.9 (*Continued*)

Belt length desig- nation	Pitch length	Permissible deviation from standard length	Number of teeth for standard lengths					
			MXL (0.080)	XL (0.200)	L (0.375	H (0.500)	XH (0.875)	XXH (1.250)
540	54.000	±0.032			144	108		
560	56.000	±0.032					64	
570	57.000	±0.032				114		
600	60.000	±0.032			160	120		
630	63.000	±0.034				126	72	
660	66.000	±0.034				132		
700	70.000	±0.034				140	80	56
750	75.000	±0.036				150		
770	77.000	±0.036					88	
800	80.000	±0.036				160		64
840	84.000	±0.038					96	
850	85.000	±0.038				170		
900	90.000	±0.038				180		72
890	98.000	±0.040					112	
1000	100.000	±0.040				200		80
1100	110.000	±0.042				220		
1120	112.000	±0.044					128	
1200	120.000	±0.044						96
1250	125.000	±0.046				250		
1260	126.000	±0.046					144	
1400	140.000	±0.048				280	160	112
1540	154.000	±0.052					176	
1600	160.000	±0.052						128
1700	170.000	±0.054				340		
1750	175.000	±0.056					200	
1800	180.000	±0.056						144

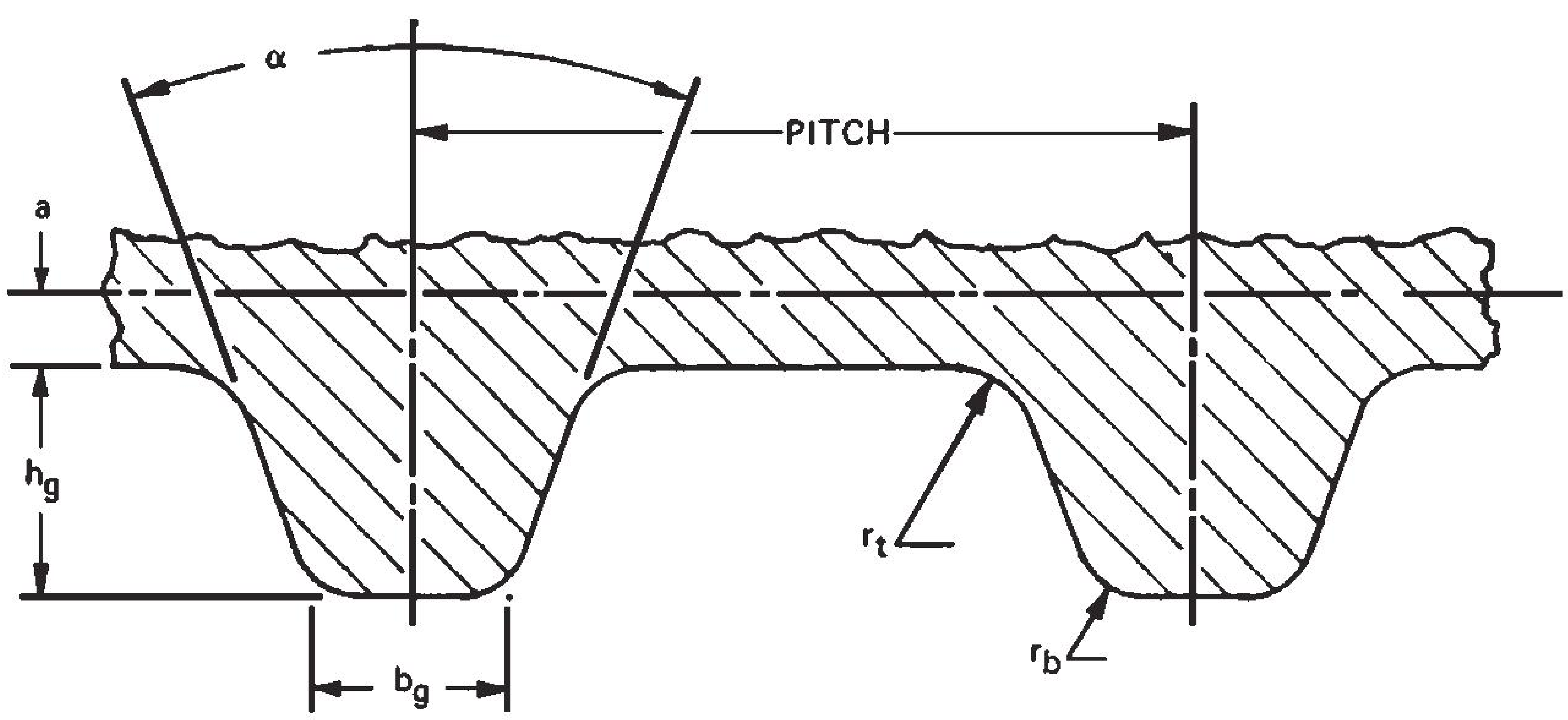

FIGURE 7.11 Trapezoidal pulley generating tool rack form.

are available from most manufacturers. If 1⁄32 in. or more of clearance is detected between the sheave and the groove gauge, then the sheave should be replaced. Any visible wear on synchronous sprockets is generally cause for replacement. Next check the alignment.

There are three possible causes of pulley misalignment:

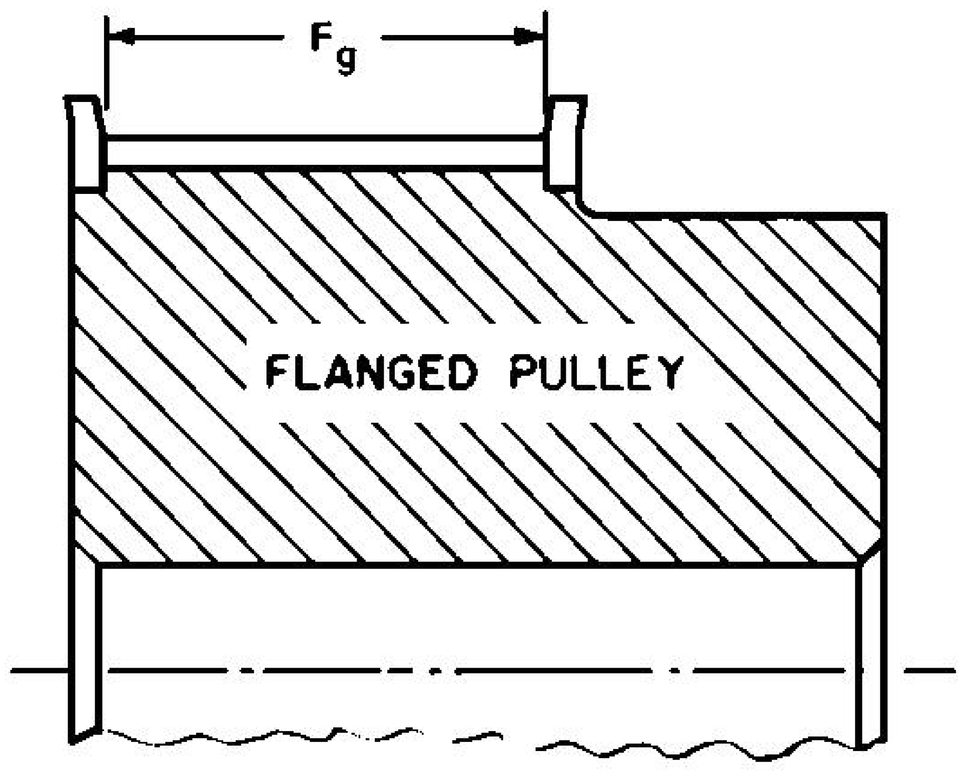

FIGURE 7.12 Trapezoidal flanged pulley.

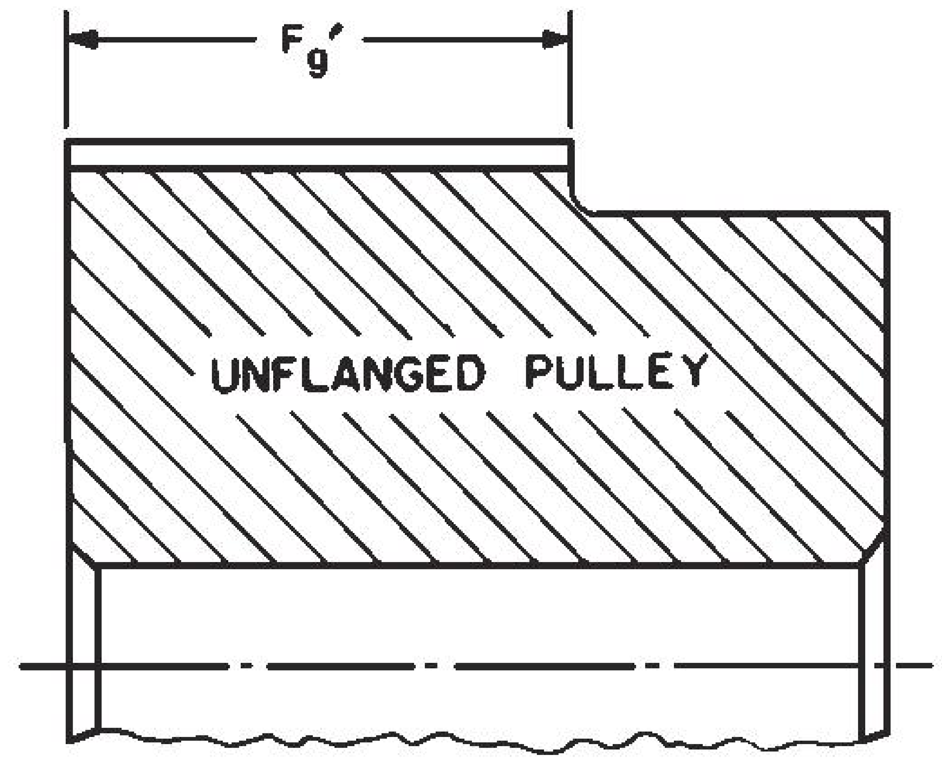

FIGURE 7.13 Trapezoidal unflanged pulley.

TABLE 7.10 Trapezoidal Pulley Generating Tool Rack Form, in.

Belt section	Number of grooves	Pitch ±0.0001	α ± 0.25 degrees	h_g +0.002 − 0.000	b_g +0.002 − 0.000	r_b ±0.001	r_t ±0.001	$2a$
MXL	10 & over	0.0800	40	0.026	0.033	0.010	0.005	0.020
XL	10 & over	0.2000	50	0.055	0.050	0.024	0.024	0.020
L	10 & over	0.3750	40	0.084	0.122	0.034	0.021	0.030
H	14 thru 19	0.5000	40	0.102	0.167	0.058	0.041	0.054
	Over 19	0.5000	40	0.102	0.167	0.058	0.056	0.054
XH	18 & over	0.8750	40	0.271	0.299	0.079	0.076	0.110
XXH	18 & over	1.2500	40	0.405	0.457	0.106	0.111	0.120

1. Driver and driven shafts are not parallel.
2. Pulleys are not properly located axially on the shaft.
3. Pulleys are tilted due to improper mounting.

These conditions will appear as both parallel and angular misalignment (see Fig. 7.14). Pulleys can be checked for tilting using a spirit level. Axial location of pulleys and shaft parallelism can be checked using a straightedge or a string. In general, sheave misalignment on V-belt drives, parallel plus angular, should not exceed ½° or ⅒ in. per foot of drive center distance. Sprocket misalignment on synchronous drives, parallel plus angular, should not exceed ¼° or 1⁄16 in. per foot of drive center distance. Check guards for wear or damage. A worn guard suggests interference with the belt drive. Clean the guard to prevent airflow restriction to the drive. Check bearings for correct alignment and lubrication, make sure that motor mounts are tightened, and ensure that guiderails are free of dirt, obstructions, and corrosion. If installing a new drive, inspect the pulleys and align the drive as previously discussed. Make certain that adequate shaft movement is provided for installation and takeup (see Table 7.13). Never pry or roll belts onto sheaves or sprockets. This will cause invisible damage to the tensile cord which will reduce belt life. For multiple V-belt drives, use matched sets of belts from one manufacturer. Do not mix old and new belts. The final step is to check the belt tension.

Belt Tension

Belt tension may be summarized with a few simple rules:

1. The best tension for a V-belt is the lowest tension at which the belt will not slip at the highest load. Since synchronous belts are more sensitive to tension, use the force-deflection numerical method described below and tension gauges to set tension for these belts.

TABLE 7.11 Standard Trapezoidal Pulley Diameters, in.

Number of grooves	MXL (0.080)		XL (0.200)		L (0.375)		H (0.500)		XH (0.875)		XXH (1.250)	
	Pitch	Outside	Pitch	Outside	Pitch	Outside	Pitch	Outside	Pitch	Outside	Pitch	Outside
10	0.255	0.235	0.637	0.617	1.194*	1.164						
12	0.306	0.286	0.764	0.744	1.432*	1.402						
14	0.357	0.337	0.891	0.871	1.671	1.641	2.228*	2.174				
16	0.407	0.387	1.019	0.999	1.910	1.880	2.546	2.492				
18	0.458	0.438	1.146	1.126	2.149	2.119	2.865	2.811	5.013	4.903	7.162	7.042
20	0.509	0.489	1.273	1.253	2.387	2.357	3.183	3.129	5.570	5.460	7.958	7.838
22	0.560	0.540	1.401	1.381	2.626	2.596	3.501	3.447	6.127	6.017	8.754	8.634
24	0.611	0.591	1.528	1.508	2.865	2.835	3.820	3.766	6.685	6.575	9.549	9.429
26	0.662	0.642			3.104	3.074	4.138	4.084	7.242	7.132	10.345	10.225
28	0.713	0.693	1.783	1.763	3.342	3.312	4.456	4.402	7.799	7.689		
30	0.764	0.744	1.910	1.890	3.581	3.551	4.775	4.721	8.356	8.246	11.937	11.817
32	0.815	0.795	2.037	2.017	3.820	3.790	5.093	5.039	8.913	8.803		
34	0.866	0.846									13.528	13.408
36	0.917	0.897	2.292	2.272	4.297	4.267	5.730	5.676				
40	1.019	0.999	2.546	2.526	4.775	4.745	6.366	6.312	11.141	11.031	15.915	15.795
42	1.070	1.050	2.674	2.654								
44	1.120	1.100	2.801	2.781	5.252	5.222	7.003	6.949				
48	1.222	1.202	3.056	3.036	5.730	5.700	7.639	7.585	13.369	13.259	19.099	18.979
60	1.528	1.508	3.820	3.800	7.162	7.132	9.549	9.495	16.711	16.601	23.873	23.753
72	1.833	1.813	4.584	4.564	8.594	8.564	11.459	11.405	20.054	19.944	28.648	28.528
84					10.027	9.997	13.369	13.315	23.396	23.286		
90											35.810	35.690
96							15.279	15.225	26.738	26.628		
120							19.099	19.045	33.423	33.313		

TABLE 7.12 Standard Trapezoidal Pulley Widths, in.

Belt section	Standard nominal pulley width	Standard pulley width designation	Minimum pulley width	
			Flanged F_g	Unflanged F'_s
MXL (9.080)	0.25	025	0.28	0.35
XL (0.200)	0.38	037	0.41	0.48
L (0.375)	0.50	050	0.55	0.67
	0.75	075	0.80	0.92
	1.00	100	1.05	1.17
H (0.500)	1.00	100	1.05	1.23
	1.50	150	1.55	1.73
	2.00	200	2.08	2.26
	3.00	300	3.11	3.29
XH (0.875)	2.00	200	2.23	2.46
	3.00	300	3.30	3.50
	4.00	400	4.36	4.59
XXH (1.250)	2.00	200	2.23	2.52
	3.00	300	3.30	3.59
	4.00	400	4.36	4.65
	5.00	500	5.42	5.72

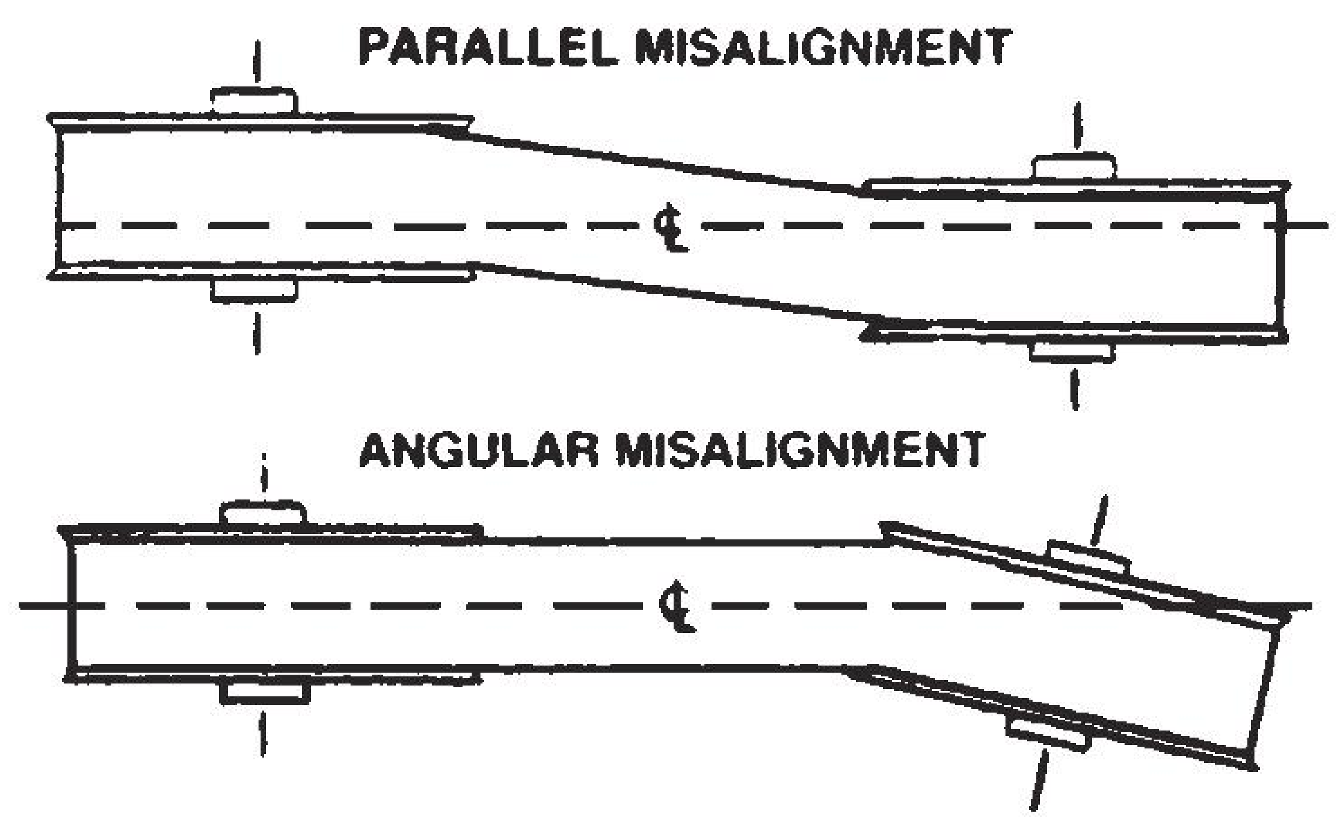

FIGURE 7.14 Types of belt misalignment.

2. When installing a new drive, set the tension, rotate the drive a few revolutions, and then recheck the tension. Check the tension once more after the first day of running.
3. For V-belt drives, check the belt tension periodically thereafter.

The preferred method of tensioning for obtaining the optimal tension described in rule 1 is the force-deflection method. This method translates the required static tension (belt tension when the drive is at rest) to a specific belt deflection upon application of the deflection force. The standard deflection for all drives is 1⁄64 in. per inch of span length. The force should be applied perpendicular to the belt at midspan (see Fig. 7.15). The force required should fall within a calculated range in order for the drive to be properly tensioned. To calculate this range, use the following procedure. This specific procedure applies only to V-belt drives. For synchronous drives, consult individual manufacturer's design manuals for specific tensioning procedures.

1. Determine the service factor using Table 7.14. (All drive design and tensioning theories assume ideal environmental and loading conditions. The service factor is used to account for nonideal

TABLE 7.13 V-Belt Installation and Takeup Allowance, in.

Standard length designation	Minimum allowance below standard center distance for installation of belts								Minimum allowance above standard center distance for maintaining tension all cross sections
	A AX	A, AX joined	B BX	BX B, joined	C CX	CX C, joined	D	D joined	
Up to and including 35	0.75	1.20	1.00	1.50					1.00
Over 35 to and including 55	0.75	1.20	1.00	1.50	1.50	2.00			1.50
Over 55 to and including 85	0.75	1.30	1.25	1.60	1.50	2.00			2.00
Over 85 to and including 112	1.00	1.30	1.25	1.60	1.50	2.00			2.50
Over 112 to and including 144	1.00	1.50	1.25	1.80	1.50	2.10	2.00	2.90	3.00
Over 144 to and including 180			1.25	1.80	2.00	2.20	2.00	3.00	3.50
Over 180 to and including 210			1.50	1.90	2.00	2.30	2.00	3.20	4.00
Over 210 to and including 240			1.50	2.00	2.00	2.50	2.50	3.20	4.50
Over 240 to and including 300			1.50	2.20	2.00	2.50	2.50	3.50	5.00
Over 300 to and including 390					2.00	2.70	2.60	3.60	6.00
Over 390					2.50	2.90	3.00	4.10	1.5% of belt length

Standard length designation	Minimum allowance below standard center distance for installation of belts						Minimum allowance above standard center distance for maintaining tension all cross sections
	3V 3VX	3V/3VX joined	5V 5VX	5V/5VX joined	8V	8V joined	
Up to and including 475	0.5	1.2					1.0
Over 475 to and including 710	0.8	1.4	1.0	2.1			1.2
Over 710 to and including 1060	0.8	1.4	1.0	2.1	1.5	3.4	1.5
Over 1060 to and including 1250	0.8	1.4	1.0	2.1	1.5	3.4	1.8
Over 1250 to and including 1700	0.8	1.4	1.0	2.1	1.5	3.4	2.2
Over 1700 to and including 2000			1.0	2.1	1.8	3.6	2.5
Over 2000 to and including 2360			1.2	2.4	1.8	3.6	3.0
Over 2360 to and including 2650			1.2	2.4	1.8	3.6	3.2
Over 2650 to and including 3000			1.2	2.4	1.8	3.6	3.5
Over 3000 to and including 3550			1.2	2.4	2.0	4.0	4.0
Over 3550 to and including 3750					2.0	4.0	4.5
Over 3750 to and including 5000					2.0	4.0	5.5

Standard effective outside length, in.*	Minimum allowance below center distance for installation of belt, in.				Minimum allowance above center distance for maintaining tension, in.	Total center distance adjustment for installation and take-up, in.			
	Belt section				All belt sections	Belt section			
	2L	3L	4L	5L		2L	3L	4L	5L
8 to 25	0.38	0.62	0.75		0.50	0.88	1.12	1.25	
25 to 36		0.62	0.75	1.00	0.50		1.12	1.25	1.50
36 to 61		0.75	0.88	1.00	0.75		1.50	1.62	1.75
61 to 80			1.00	1.12	1.12			2.12	2.24
80 through 100			1.12	1.25	1.50			2.62	2.75

*"To" means the range is to, but not including the second length.

TABLE 7.13 (*Continued*)

Center distance allowance for installation and tensioning, trapezoidal belts		
Belt length, in.	Standard installation allowance (flanged pulleys removed for installation)	Tensioning allowance (any drive)
3.6 to 5.0	0.02″	0.02″
Over 5.0 to 10.0	0.03″	0.03″
Over 10.0 to 20.0	0.04″	0.03″
Over 20.0 to 40.0	0.05″	0.04″
Over 40.0 to 60.0	0.07″	0.05″
Over 60.0 to 180.0	0.12″	0.08″
Additional center distance allowance for installation over flanged pulleys* (add to installation allowance)		
Pitch	Small pulley flanged	Both pulleys flanged
0.080″ (MXL)	0.33″	0.49″
0.200″ (XL)	0.46″	0.71″
0.375″ (L)	0.64″	0.85″
0.500″ (H)	0.64″	0.96″
0.875″ (XH)	1.14″	1.92″
1.250″ (XXH)	1.53″	2.65″

*For drives that require installation of the belt over one pulley at a time, use the value for both pulleys flanged—even if only one pulley is flanged.

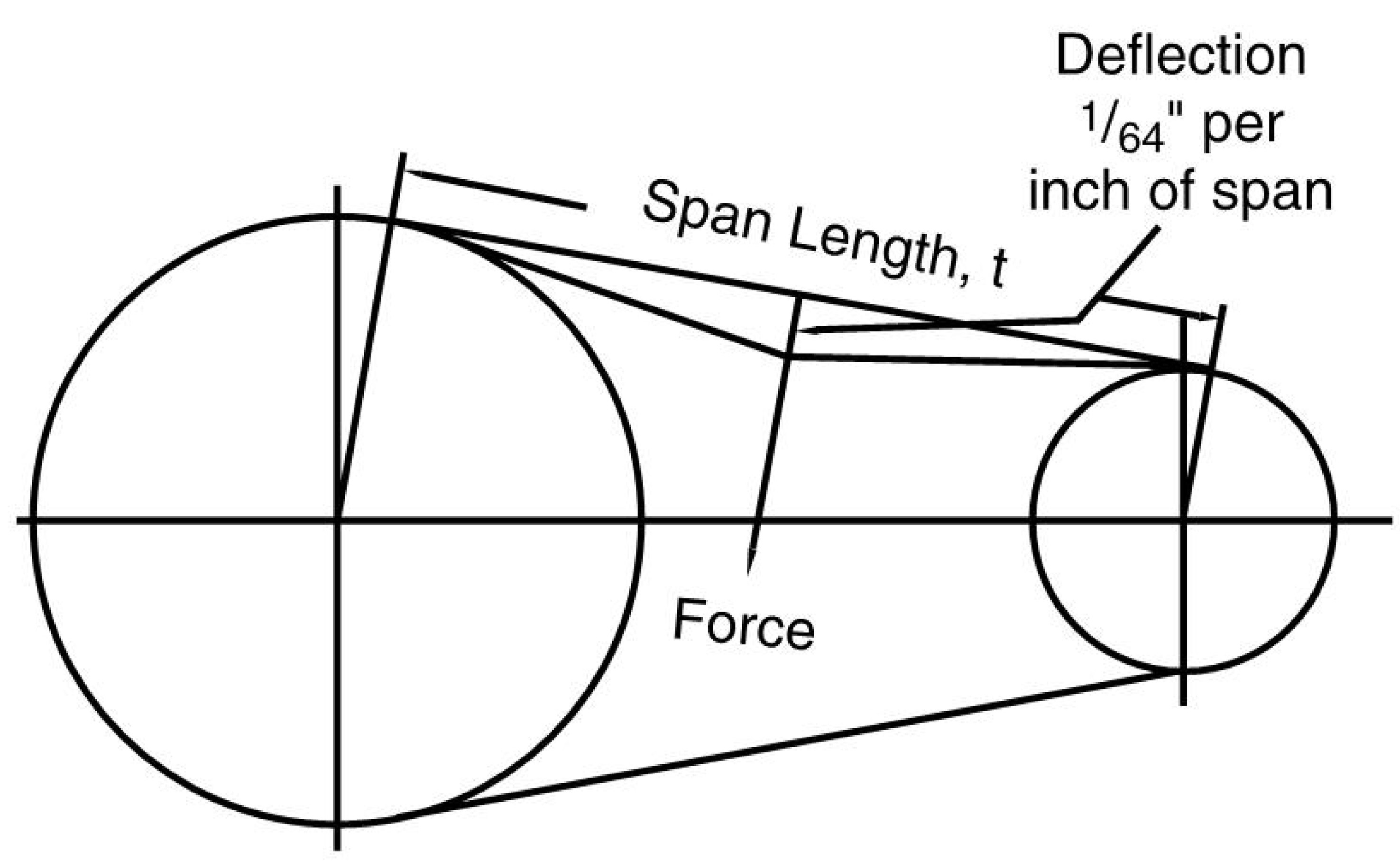

FIGURE 7.15 Measuring tension of V-belt drives.

conditions.) If either the driver or driven unit is not listed on the table, select a machine with similar operating characteristics.

2. Calculate the design horsepower using Formula 1.
3. Calculate the belt speed using Formula 2.
4. Determine factor *G* using Table 7.15. Drive theory is based on a 180° arc of contact on the small sheave. Factor G is used to correct the formula when the arc of contact is not 180°.
5. Calculate the static tension (T_{st}) using Formula 3 and Table 7.16.
6. Calculate the upper and lower recommended forces using Formulas 4 and 5 and Table 7.16.

TABLE 7.14 Recommended Service Factors

	DriveN			DriveR		
	AC motors: Normal torque, Squirrel cage, synchronous, split phase. DC motors: Shunt wound Engines: Multiple cylinder internal combustion.			AC motors: High torque, high slip, repulsion-induction, single phase, series wound, slip ring. DC motors: Series wound, compound wound. Engines: Single cylinder internal combustion. Line shafts Clutches		
The machines listed below are representative samples only. Select the group listed whose load characteristics most closely approximate those of the machine being considered.	Intermittent service 3–5 hours daily or seasonal	Normal service 8–10 hours daily	Continuous service 16–24 hours daily	Intermittent service 3–5 hours daily or seasonal	Normal service 8–10 hours daily	Continuous service 16–24 hours daily
Agitators for liquids Blowers and exhausters Centrifugal pumps and compressors Fans up to 10 horsepower Light-duty conveyors	1.0	1.1	1.2	1.1	1.2	1.3
Belt conveyors for sand, grain, etc. Dough mixers Fans-over 10 horsepower Generators Line shafts Laundry machinery Machine tools Punches-presses-shears Printing machinery Positive displacement rotary pumps Revolving and vibrating screens	1.1	1.2	1.3	1.2	1.3	1.4
Brick machinery Bucket elevators Exciters Piston compressors Conveyors (drag-pan-screw) Hammer mills Paper mill beaters Piston pumps Positive displacement blowers Pulverizers Saw mill and woodworking machinery Textile machinery	1.2	1.3	1.4	1.4	1.5	1.6
Crushers (gyratory-jaw-roll) Mills (ball-rod-tube) Hoists Rubber calenders-extruders-mills	1.3	1.4	1.5	1.5	1.6	1.8

TABLE 7.15 Arc of Contact Correction Factor G for V-V Drives

$\frac{D-d}{C}$	Arc of contact on small sheave, degrees	Factor G
0.00	180	1.00
0.10	174	0.99
0.20	169	0.97
0.30	163	0.96
0.40	157	0.94
0.50	151	0.93
0.60	145	0.91
0.70	139	0.89
0.80	133	0.87
0.90	127	0.85
1.00	120	0.82
1.10	113	0.80
1.20	106	0.77
1.30	99	0.73
1.40	91	0.70
1.50	83	0.65

*All standard V belts

TABLE 7.16 Factor M and Factor Y

Cross section	M	Y	Cross section	M	Y
			Hi-power II		
			A	0.51	7
Super HC molded			B	0.50	8
notch			C	1.5	18
3VX	0.29	4	D	3.0	27
5VX	0.78	13			
			Hi-power II power band		
			A	0.66	7
Super HC molded			B	1.0	9
notch power band			C	1.8	18
3VX	0.39	4	D	3.4	28
5VX	0.98	13			
Super HC					
5V	1.0	11			
8V	2.6	22			
			Tri-power		
Super HC power			AX	0.47	7
band			BX	0.76	8
5V	1.2	11	CX	1.31	15
8V	5.0	22			

Formula 1:

$$\text{Design hp} = (\text{service factor})\ (\text{hp})$$

Formula 2:

$$\text{Belt speed} = \frac{\text{(pitch diameter) (rpm of the same sheave)}}{3.82}$$

Formula 3:

$$T_{st} = 15 \frac{2.5 - G}{G} \frac{\text{(design hp) (1000)}}{NV} + \frac{MV^2}{10{,}000{,}000}$$

where G = arc correction factor from Table 7.15
N = number of belts
V = belt speed (ft/min) (Formula 2)
M = constant from Table 7.16

Formula 4:

$$\text{Lower recommended force} = \frac{T_{st} + Y}{16}$$

Formula 5:

$$\text{Upper recommended force} = \frac{(1.5)\, T_{st} + Y}{16}$$

where T_{st} = static tension
Y = constant from Table 7.16

Now measure the distance (in inches) between the points where the belts contact the sheave on the span which is to be deflected (see Fig. 7.13). Divide this value by 64 to obtain the required span deflection. If the actual force required to deflect one belt this amount is above the range calculated, the belts are too tight. Likewise, if the actual force is lower than the calculated range, the belts are too loose. Ideally, when tensioning the belts, the force should be at the high end of this range, and the drive should be retensioned when this force drops down to the lower end.

If joined belts are to be used, the force range needs to be increased proportionately. For example, if a three-strand joined belt is being deflected, both the upper and lower forces of this range should be multiplied by 3. (*Note:* This has no effect on total drive tension or shaft load.)

This force range can be approximated by using Tables 7.17, 7.18, and 7.19 if the drive meets all the conditions specified in the table. However, these tables assume that the drive is properly designed. If the drive was designed a number of years ago, it is possibly overdesigned because of the increased hp capacity of today's V-belts. If this is the case, using this table to obtain the deflection forces may overtension the drive.

TROUBLESHOOTING

Use the following guide as an aid in determining possible causes of drive problems.

Premature Belt Failure

Symptoms	Probable cause	Corrective action
• Broken belt(s)	1. Underdesigned drive 2. Belt rolled or pried onto sheave 3. Object falling into drive 4. Severe shock load	1. Redesign, using design manual. 2. Use drive takeup when installing. 3. Provide adequate guard or drive protection. 4. Redesign to accommodate shock load.

TABLE 7.17 Recommended Deflection Force per Belt for Narrow Cross-Sectioned V-Belts

V-belt cross section	Small sheave diam. range, in.	Small sheave rpm range	Speed ratio range	Recommended deflection force, lb	
				Min	Max
	2.50–3.50	1200–3600		3.3	4.8
3V	3.51–6.00	900–1800	2.00–4.00	5.0	7.5
	7.0–12	600–1200		13	20
5V	12.1–16	400–900	2.00–4.00	15	23
	12.5–17	400–900		35	51
8V	17.1–24	200–700	2.00–4.00	40	60

TABLE 7.18 Recommended Deflection Force per Belt for Classical V-Belts

V-belt cross section	Small sheave diam. range, in.	Speed ratio range	Recommended deflection force, lb	
			Min	Max
	3.0–3.2		2.4	3.4
	3.4–3.6		2.5	3.7
A	3.8–4.2	2.0–4.0	2.9	4.2
	4.6–7.0		3.5	5.0
	4.6		4.4	6.3
	5.0–5.4		4.9	7.1
B	5.6–6.4	2.0–4.0	5.4	7.8
	6.8–9.4		6.2	9.0
	7.0		7.5	11
	7.5–8.0		8.4	12
C	8.5–10.0	2.0–4.0	9.5	14
	10.5–16.0		11	17
	12.0–13.0		17	24
D	13.5–15.5	2.0–4.0	19	27
	16.0–22.0		22	31
E	21.6–24.0	2.0–4.0	32	47

Severe or Abnormal V-Belt Wear

Symptoms	Probable cause	Corrective action
• Wear on top surface of belt	1. Rubbing against guard	1. Replace or repair guard.
	2. Idler malfunction	2. Replace idler.
• Wear on top corner of belt	1. Belt-to-sheave fit incorrect (belt too small for groove)	1. Use correct belt-to-sheave combination.
• Wear on belt sidewalls	1. Belt slip	1. Retension until slipping stops.
	2. Misalignment	2. Realign sheaves.
	3. Worn sheaves	3. Replace sheaves.
	4. Incorrect belt	4. Replace with correct belt size.
• Wear on bottom corner of belt	1. Belt-to-sheave fit incorrect	1. Use correct belt-to-sheave combination.
	2. Worn sheaves	2. Replace sheaves.

TABLE 7.19 Static Tensions for Light-Duty V-Belts on Normal Two-Sheave Drives

	Motor hp		
Motor rpm	1160	1750	3450
Static tension for no. 1 (3L) V-belt cross section, lb			
1⁄12	8–10	6–8	4–5
1⁄10	9–11	6–8	4–5
1⁄8	10–13	8–10	5–6
1⁄6	11–14	9–11	6–8
1⁄4	13–16	10–13	8–10
1⁄3		12–15	9–11
1⁄2			10–13
Static tension for no. 2 (4L) V-belt cross section, lb			
1⁄10	8–10	6–8	4–5
1⁄8	10–13	7–9	5–6
1⁄6	12–15	9–11	6–8
1⁄4	15–19	11–14	7–9
1⁄3	19–24	13–16	9–11
1⁄2	20–25	16–20	12–15
3⁄4	24–30	20–25	15–19
1		22–28	17–21
Static tension for no. 3 (5L) V-belt cross section, lb			
1⁄8	7–9	5–6	8–10
1⁄6	9–11	7–9	8–10
1⁄4	12–15	12–15	10–13
1⁄3	14–18	11–14	11–14
1⁄2	19–24	15–19	14–19
3⁄4	24–30	19–24	17–21
1	28–35	23–29	20–25
1 1⁄2	34–43	29–36	26–33

Symptoms	Probable cause	Corrective action (*Continued*)
• Wear on bottom surface of belt	1. Belt bottoming on sheave groove 2. Worn sheaves 3. Debris in sheaves	1. Use correct belt/sheave match. 2. Replace sheaves. 3. Clean sheaves.
• Undercord cracking	1. Sheave diameter too small 2. Belt slip 3. Backside idler too small 4. Improper storage	1. Use larger diameter sheaves. 2. Retension. 3. Use larger diameter backside idler. 4. Don't coil belt too tightly, kink or bend. Avoid heat and direct sunlight.
• Undercord or sidewall burn or hardening	1. Belt slipping 2. Worn sheaves 3. Underdesigned drive 4. Shaft movement	1. Retension until slipping stops. 2. Replace sheaves. 3. Refer to design manual. 4. Check for center distance changes.
• Belt surface hard or stiff	1. Hot drive environment	1. Improve ventilation to drive.
• Belt surface flaking, sticky or swollen	1. Oil or chemical contamination	1. Do not use belt dressing. Eliminate sources of oil, grease or chemical contamination.

V-Belts Turn Over or Come Off Drive

Symptoms	Probable cause	Corrective action
• Involves single or multiple belts	1. Shock loading or vibration	1. Check drive design. Use joined belts.
	2. Foreign material in grooves	2. Shield grooves and drive.
	3. Misaligned sheaves	3. Realign the sheaves.
	4. Worn sheave groves	4. Replace sheaves.
	5. Damaged tensile member	5. Use correct installation and belt storage procedure.
	6. Incorrectly placed flat idler pulley	6. Carefully align flat idler on slack side of drive as close as possible to driver sheaves.
	7. Mismatched belt set	7. Replace with new set of matched belts. Do not mix old and new belts.
	8. Poor drive design	8. Check for center distance stability and vibration dampening.

Belt Noise

Symptoms	Probable cause	Corrective action
• Belt squeals or chirps	1. Belt slip	1. Retension.
	2. Contamination	2. Clean belts and sheaves.
• Slapping sound	1. Loose belts	1. Retension.
	2. Mismatched set	2. Install matched belt set.
	3. Misalignment	3. Realign pulleys so all belts share load equally.
• Rubbing sound	1. Guard interference	1. Repair, replace, or redesign guard.
• Grinding sound	1. Damaged bearings	1. Replace, align, and lubricate.
• Unusually loud drive	1. Incorrect belt	1. Use correct belt size. Use correct belt tooth profile for sprockets on synchronous drive.
	2. Worn sheaves	2. Replace.
	3. Debris in sheaves	3. Clean sheaves, improve shielding, remove rust, paint, or remove dirt from grooves.

Problems with Synchronous Belts

Symptoms	Probable cause	Corrective action
• Unusual noise	1. Misaligned drive	1. Correct alignment.
	2. Too low or high tension	2. Adjust to recommended value.
	3. Backside idler	3. Use inside idler.
	4. Worn sprocket	4. Replace.
	5. Bent guide flange	5. Replace.
	6. Belt speed too high	6. Redesign drive.
	7. Incorrect belt profile for sprocket	7. Use proper belt/sprocket combination.
	8. Subminimal diameter	8. Redesign drive using larger diameters.
	9. Excess load	9. Redesign drive for increased capacity.
• Tensile break	1. Excessive shock load	1. Redesign drive for increased capacity.
	2. Subminimal diameter	2. Redesign drive using larger diameters.
	3. Improper belt handling and storage prior to installation	3. Follow proper storage and handling procedures
	4. Debris or foreign object in drive	4. Remove objects and check guard.
	5. Extreme sprocket run-out	5. Replace sprocket.

(*Continued*)

Symptoms	Probable cause	Corrective action
• Premature tooth wear	1. Too low or high belt tension	1. Adjust to recommended value.
	2. Belt running partly off unflanged sprocket	2. Correct alignment.
	3. Misaligned drive	3. Correct alignment.
	4. Incorrect belt profile for sprocket	4. Use proper belt/sprocket combination.
	5. Worn sprocket	5. Replace.
	6. Rough sprocket teeth	6. Replace sprocket.
	7. Damaged sprocket	7. Replace
	8. Sprocket not to dimensional spec--ification	8. Replace.
	9. Belt hitting drive bracketry or other structure	9. Remove obstruction or use idler.
	10. Excessive load	10. Redesign drive for increased capacity.
	11. Insufficient hardness of sprocket material	11. Use a more wear-resistant sprocket.
	12. Excessive debris	12. Remove debris, check guard.
	13. Cocked bushing/sprocket	13. Install bushing per instructions.
• Tooth shear	1. Excessive shock loads	1. Redesign drive for increased capacity.
	2. Less than 6 teeth-in-mesh	2. Redesign drive.
	3. Extreme sprocket run-out	3. Replace sprocket.
	4. Worn sprocket	4. Replace.
	5. Backside idler	5. Use inside idler.
	6. Incorrect belt profile for the sprocket	6. Use proper belt/sprocket combination.
	7. Misaligned drive	7. Realign.
	8. Belt undertensioned	8. Adjust tension to recommended value.
• Vibration	1. Incorrect belt profile for the sprocket	1. Use proper belt/sprocket combination.
	2. Too low or high belt tension	2. Adjust tension to recommended value.
	3. Bushing or key loose	3. Check and reinstall per instructions.

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CHAPTER 8
MECHANICAL VARIABLE-SPEED DRIVES

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Because of their versatility and simplicity, mechanical variable-speed drives have enjoyed wide popularity for almost 100 years. Even today, with the advent of low-cost electronic variable-speed methods, mechanical drives remain in wide industrial usage. Major segments of the automotive industry are investigating mechanical adjustable-speed (MAS) as the wave of the future for an efficient, widely variable method of power transmission for both accessory drives and main engine-to-wheel drives. Because of the similar nature of most MAS drives, an understanding of the principles of operation will greatly aid maintenance and troubleshooting. In this chapter we will (1) define similar types and principles of operation and (2) offer generalized maintenance comments for similar types. For specific or unusual problems, the best source of information will be obtained from the manufacturers' literature and representatives.

The most popular general type of MAS drive involves the use of a sliding cone-face pulley with a wide V-belt. The basic mechanism involved entails some method of varying the running pitch diameter of a belt. Two fixed-pitch pulleys, driven at a constant speed, will provide a constant output (driven) speed. This driven speed will produce constant input speed, as determined by the ratio of the pulleys' pitch diameters. If we reduce the pitch diameter of the constant-speed or driving pulley (or increase the diameter of the driven pulley), the output speed (or driven pulley speed) will be reduced, and vice versa. This is the basic principle; the methods of accomplishing it vary widely. Functional problems can frequently be pinpointed by analysis of the problem and principal; i.e., if the output speed doesn't change, then for some reason the pulley pitch diameters are not changing.

GENERAL FRICTION-TYPE BELT

Most MAS drives rely on the axial movement of cone-faced disks to vary running pitch diameters. These disks normally have equal and opposite inclined faces with a shaft through their centers. The face angles are designed to mate with the edge angles or tapered sides on the belt and thereby transmit torque through the frictional forces developed between the belt edge and disk face. Using belts with configurations or face angles significantly different from those for which the disks were designed will reduce this frictional force and thus the power-transmitting ability and life of the drive. Since disks and belts are designed as a geometrically integrated package, the use of a belt whose

width or thickness is not equal to the recommended belt cross section will modify the position or pitch diameter that the belt attains for a given disk position. The result, if the unit functions at all, will be a reduction in the speed range capability and component life.

With one exception, most MAS belt-type drives rely on dry friction between the belt and disks. Grease, oil, water, or other contaminants on the disk belt interface will reduce the coefficient of friction and hence the torque transmitting capacity of the drive. Belt life also will be affected. Disk faces and belt edges should always be inspected for, and kept as free as possible from, contaminants.

Packaged drives are usually available with sealed enclosures to prevent contamination. Open drives must be protected from these impurities. Wash-down applications are common in the food-processing industry and should be well protected to extend component life. The exception to this occurs with traction and metal-belt drives that run in an oil bath. The lubricant used should be checked periodically for level and contamination and replaced according to manufacturers' recommendations.

One problem in mechanical adjustable-speed drives occurs when the axially movable sliding disks will not slide on their shafts. If movable disks cannot move on command, no speed variation can be obtained. This condition is frequently caused by the products of fretting corrosion lodging between the shaft and the bore of the rotating disk hub, effectively binding the pieces together. Some drives that have run at one setting for a very long time may even be locked by actual wear.

Until a few years ago, most MAS drives used grease lubrication between mating sliding parts. This system, because of its simplicity, proven performance, and long life when properly maintained, is still often used. The present field population of both old and new units demanding periodic relubrication is extensive, exceeding those which do not require such regular service. Therefore, all MAS drives should be inspected to determine whether periodic relubrication is required, and the manufacturers' recommendations for frequency and lube type should be strictly adhered to.

The interface between the axially movable (sliding) disk and its shaft must perform several functions: The fit must be loose enough to allow axial movement yet tight enough to prevent oscillating and impost loading, both of which contribute to the previously mentioned fretting corrosion. See Fig. 8.1. Most designs incorporate single- or multiple-key keyway arrangements in this area to transmit torque from the sliding disk to the shaft. Note that some designs use an external method of transmitting sliding-disk torque, such as a keyed external collar. There are also units employing polygon-shaped or splined shafts for this purpose. Regardless of the specific design, those which have direct metal-to-metal contact must be lubricated to minimize fretting corrosion and to flush out its products.

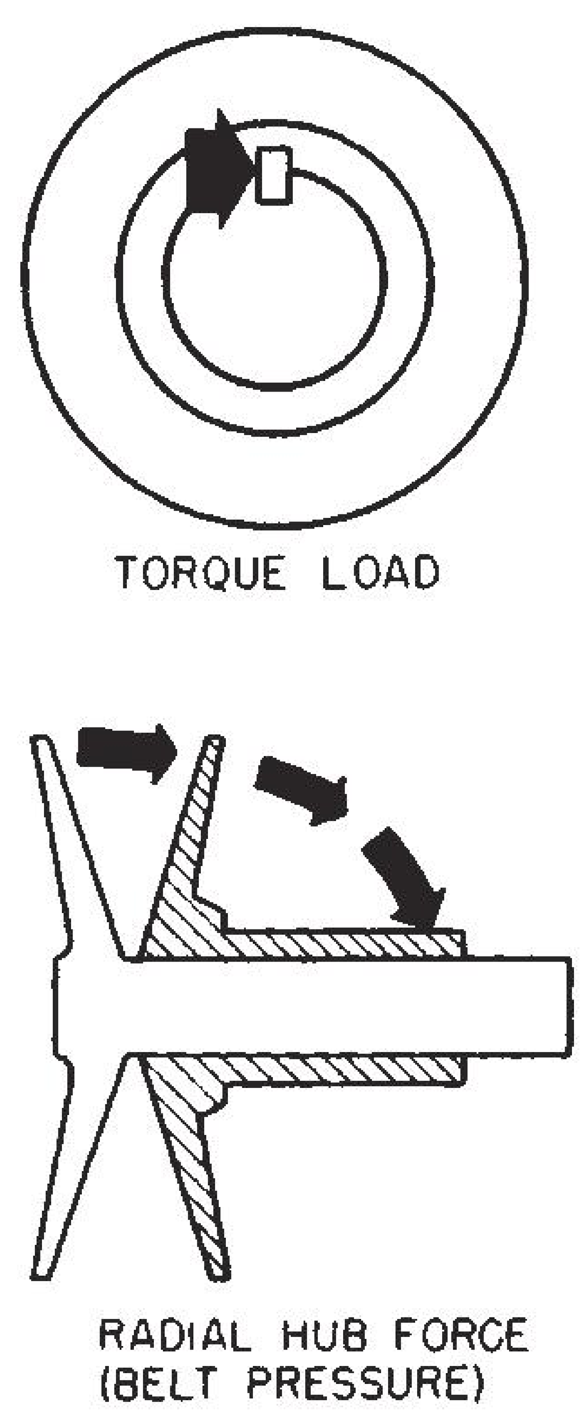

FIGURE 8.1 Radial hub force tends to squeeze lubricant from mating surface.

Depending on the load, duty cycle, and environmental considerations, a common recommendation is to use a high-quality grade of NCGI no. 1 grease, pumped into the appropriate fittings every 1 to 4 weeks. Enough should be used to ensure flushing of the old grease but not so much as to force large quantities out the ends of the sliding disks. Too much lubricant may lead to disk-face contamination and a general mess. If possible, the drive should be cycled through its speed range both during lubrication and at least once daily (to prevent the start of fretting). This may seem excessive, but experience shows that the results produce many years of reliable service.

A typical example of a grease lubricant design is shown in Fig. 8.2. This is a Reeves unit and incorporates a close grooving hub bore design that effectively distributes lubricant between hub and shaft. This design also prevents the buildup of large fretted areas. If fretting does start, the grooves cause grease to flow into the affected areas, minimizing the corrosive action.

Many new mechanical adjustable-speed devices are designed to eliminate periodic relubrication. These usually incorporate some form of nonmetallic bearing material inside the sliding-disk hubs. In many designs, this bearing material is replaceable. In others, it is not. The replaceable-insert designs have a definite advantage, since periodic inspection and replacement of the insert will return the drive to almost new condition. Periodic inspection is important here. If the insert material is allowed to completely wear through, the hubs and shafts themselves will become worn and not function properly when the inserts are replaced. This will lead to very rapid wear of the new inserts and unacceptable drive performance. The only recourse is to replace the entire shaft and disk-hub assembly.

Most manufacturers of no-lube pulleys provide instructions for easily gauging wear on the hub inserts. This may be by physical inspection or by rocking the sliding-disk heel and toe or rotating it to detect an increase in internal clearance. If looseness is detected, a more thorough inspection by disassembly must be performed. Any suspect part should be checked immediately; insert kits are much less expensive than disk assemblies. Running a drive to the failure point is false economy.

When replacing no-lube bushings, the manufacturer's recommendations should be followed. These are usually included with the bushing kits. Some designs merely snap into place; others require glues or physical retention methods. For either type, thoroughly clean and inspect all components and remove all traces of old bushings, grease, debris, etc. Also clean and inspect disk faces, belts, control mechanisms, springs, and the like. A thorough checking and cleaning will help prevent unexpected downtime from undetected sources.

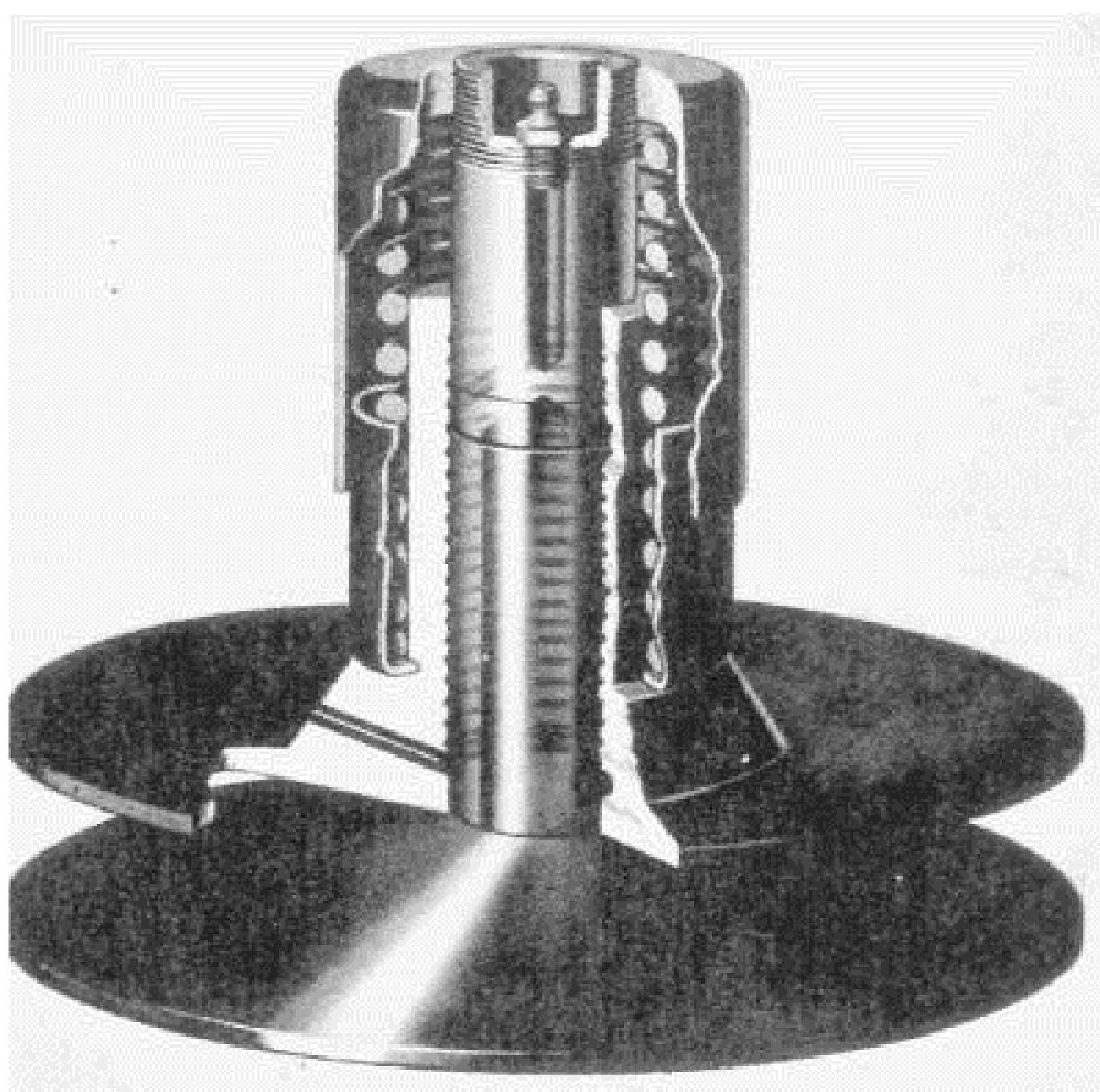

FIGURE 8.2 "Close grooving" of motor pulley.

TYPES OF FRICTION DRIVES (BELT AND SHEAVE)

Adjustable While in Motion (Nonenclosed)

These are usually sold as separate, complete pulleys, designed for mounting directly on a motor shaft (see Fig. 8.3). These pulleys are normally spring-loaded and may have either one or both of the disk halves capable of axial movement (see Figs. 8.4 and 8.5). They are designed to drive a fixed-diameter sheave. Output speed variation at the fixed sheave is controlled by varying the center distance between the fixed and spring-loaded sheave. This usually involves a sliding-motor mounting base.

Pulleys with one fixed and one movable disk may be used to drive either a flat-face or a V sheave. Note that as the pitch diameter traversed by the belt on the spring-loaded sheave changes, the center line of the belt moves axially. If a flat-faced pulley is the driven element, both motor shaft and driven shaft must be parallel, and the motor should move perpendicularly to the driven shaft.

The side movement of the belt results in its tracking across the flat face of the driven units while remaining perpendicular to both shafts. The flat face must be positioned so that the belt does not run off the sides of the sheave at any time. Centering at midrange is best.

If the driven pulley is a V-type, the sliding motor base must be angled so that the motor also moves sideways during forward and backward speed adjustment in such a way as to maintain the position of the belt center line. Note that the motor base must be angled but that the motor must be mounted on the base so that its shaft is parallel to the driven shaft. Manufacturers' instructions as to angle and direction should be followed.

Shortened belt life on sheave arrangements is usually due to misalignment, caused by inadequate sheave alignment or incorrect motor base angle. Extreme wear on the sliding rods or rails of the motor base will allow this angle to change, increasing wear.

The problem of belt-tracing alignment with a V sheave is solved by using a pulley where both disks are movable axially. In some of these designs, each disk half has its own spring. But all share a common shaft and are sized to accept the motor shaft. With this arrangement, care must be taken to ensure that both disks are easily and equally movable on the common shaft. If one disk begins to bind, the other will take up the slack, resulting in belt misalignment. If this condition occurs, it may be corrected by disassembly, cleaning, and relubrication. Both springs should be inspected carefully for binding or breakage. Required compression force on each spring must be equal by 65 percent.

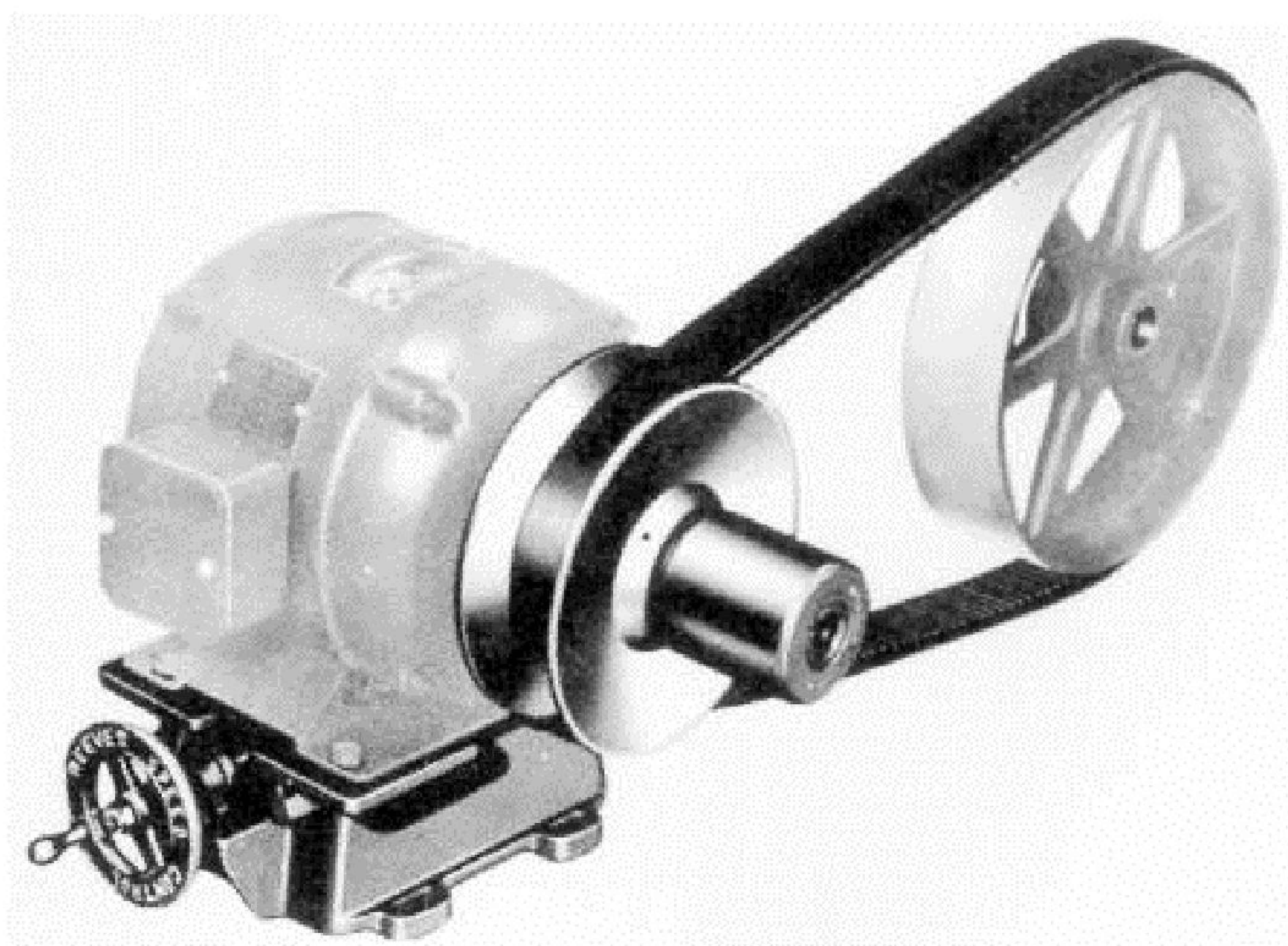

FIGURE 8.3 (*a*) Spring-loaded split pulley, adjustable with drive-in operation.

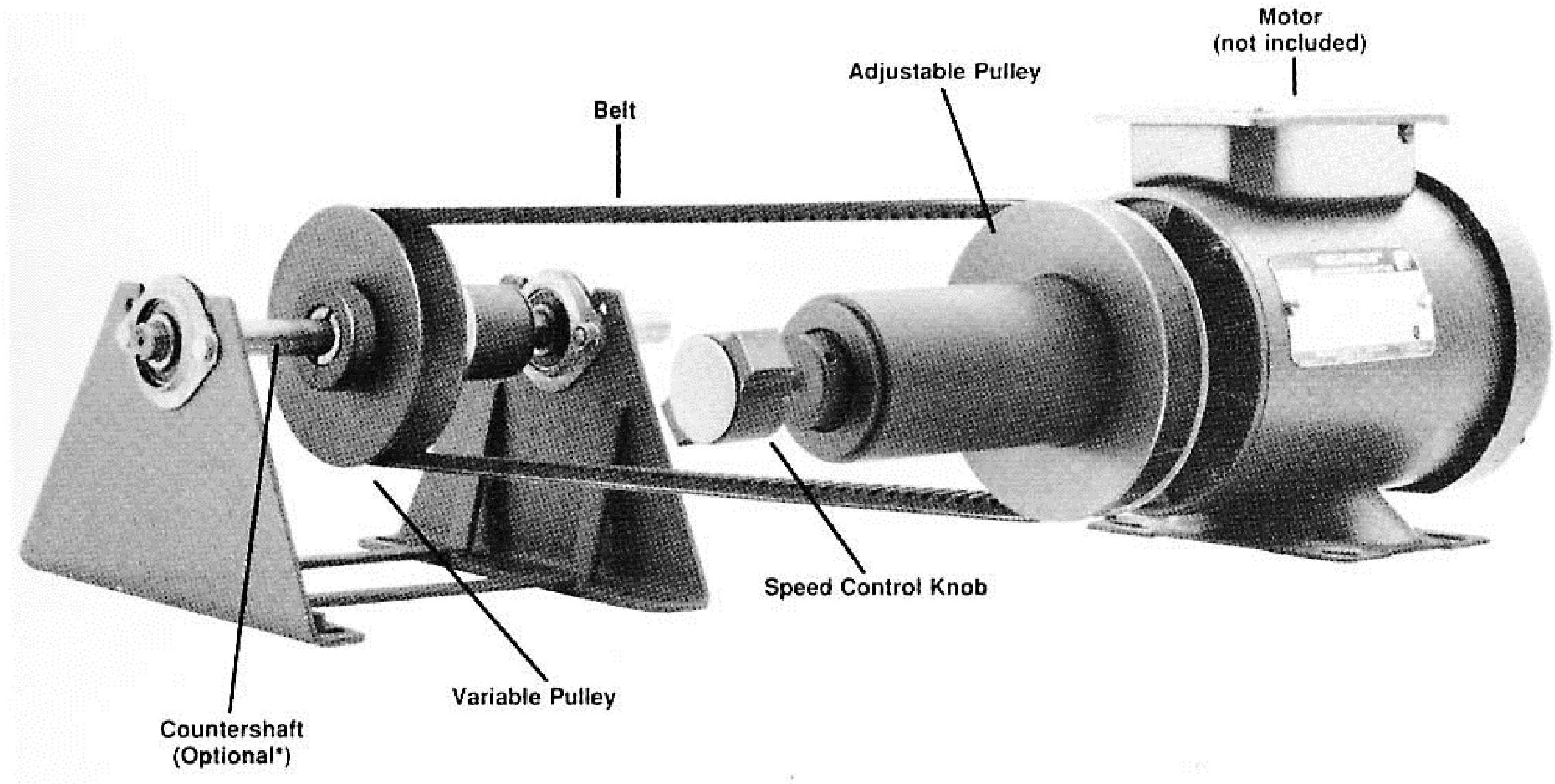

FIGURE 8.3 (*Continued*) (*b*) Fixed-center-distance compound-pulley arrangement. (*Reeves Div.*)

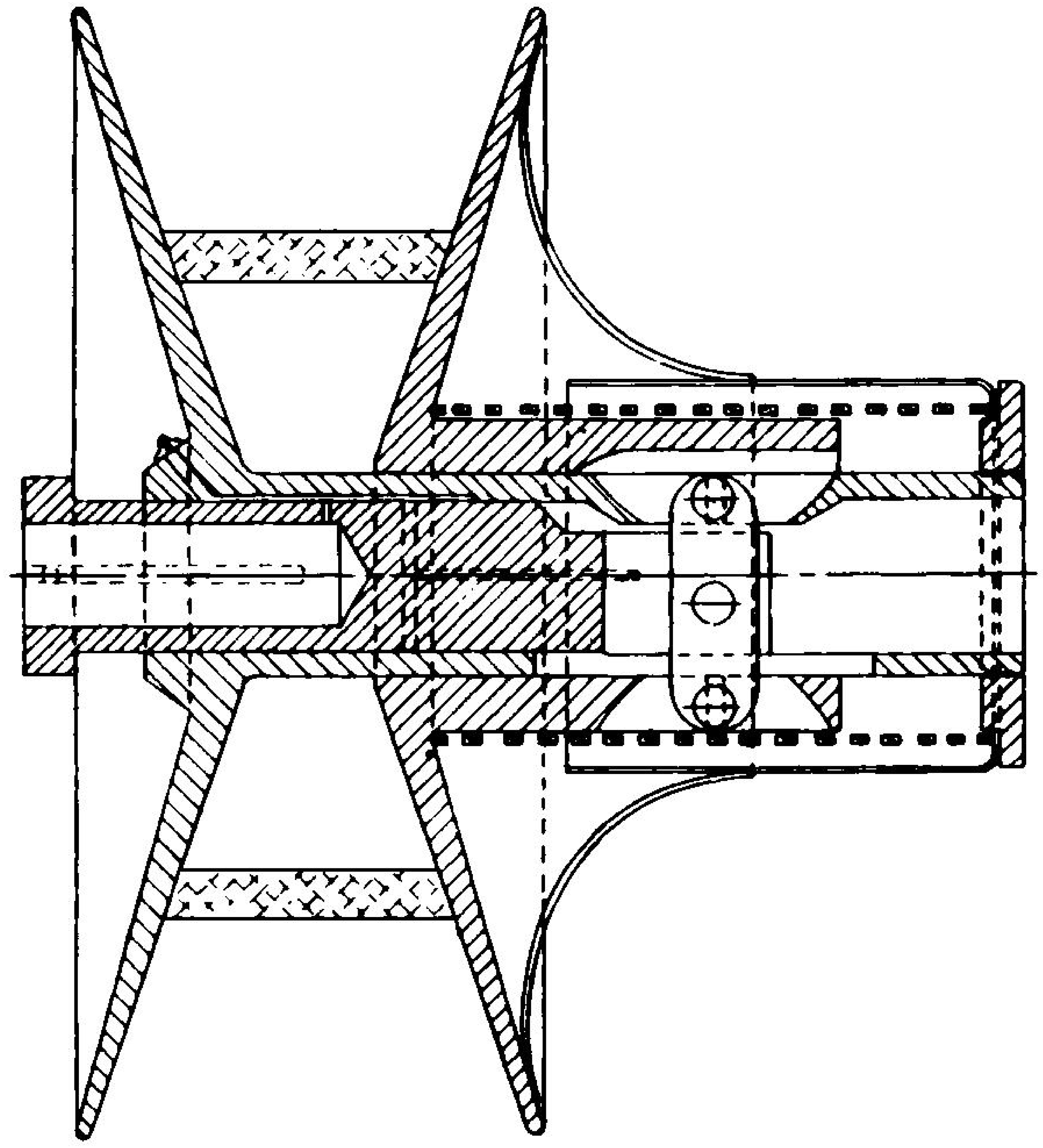

FIGURE 8.4 Pulley halves are given equal lateral movement as speed is adjusted. (*Lewellen Mfg. Co.*)

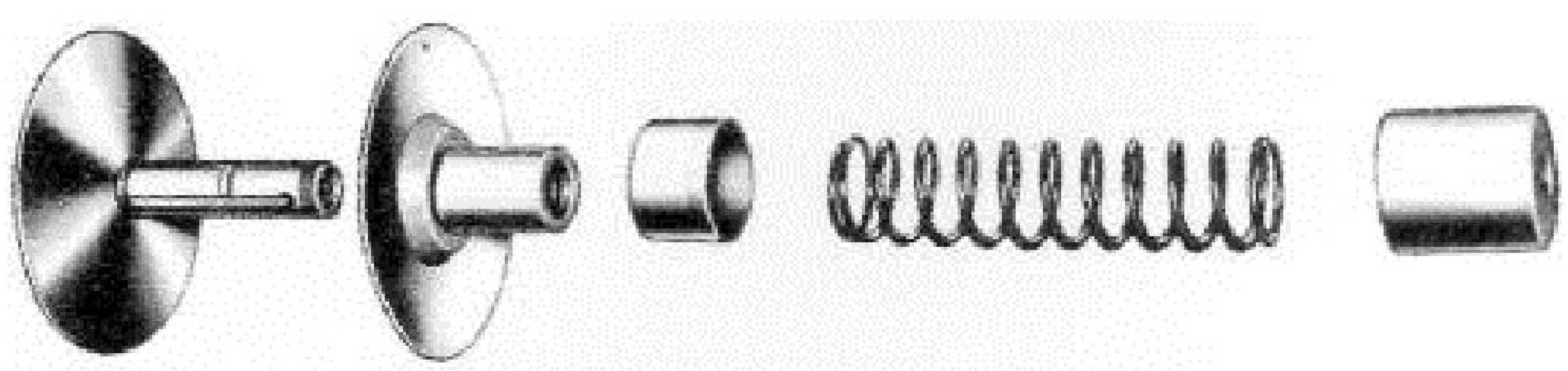

FIGURE 8.5 Exploded view of motor pulley disks.

In order to overcome the problem of unequal spring forces and binding, some double-acting designs use a single spring and a cam or linkage arrangement that mechanically restricts each disk to equal and opposite movement. These linkages must be periodically inspected and lubricated, since most are prone to binding and seizing. The vibrating and oscillating motion that wears disks and hubs can cause rapid linkage wear.

Up to this point, we have considered single open pulleys with movable flanges driving to a fixed-pitch sheave. These devices seldom exceed a 3 to 1 ratio range in speed variation. In order to provide greater speed range at a low cost, the compound pulley arrangement was developed. This system incorporates variable-pitch sheaves on both the driving and driven ends. Such combinations operate on fixed center distances. Speed variation is obtained by adjusting the pitch of the motor pulley mechanically. The driven pulley is spring-loaded. As the pitch diameter of the motor pulley is adjusted, belt slack is taken up by the spring-loaded pulley. Since the pitch diameters of both pulleys vary, speed ranges of a 9 to 1 ratio are common. Maintenance of the hub and shaft portion of the motor pulley and the entire driven pulley is similar to that for single pulleys. However, additional attention must be paid to the speed-changing mechanism. This mechanism normally uses either a handwheel and screw with bearings arrangement or a remotely pivoted lever system. The screw and bearing, set

within the constant-speed pulley hub, is the most common. This arrangement should be inspected for loose or worn parts or bearing failures. The system incorporates a reaction arm that must be fixed against a stop to prevent rotation while remaining free to move axially with pulley adjustment. Note that since both pulleys are single-acting, with movable flanges on opposite sides, they must be installed with both motor shaft and output shaft parallel. Belt alignment is critical.

One additional type of wide-range assembly uses a compounded variable-pitch sheave assembly mounted on an intermediate countershaft between fixed driving and driven pulleys (see Figs. 8.6 and 8.7). Changing the position of the countershaft, usually on a pivoted arm, causes the compound floating flange to shift in position under belt pressure and thereby vary the operating diameters to achieve the desired speed. The necessarily short hub on the common flange frequently experiences high wear rates.

Static-Adjustment Types

Adjustable-while-stopped pulleys are designed for the machine which at the outset may require a slight speed adjustment to operate at optimal performance but once set will rarely need adjustment.

The significant makes of stationary-control adjustable pulleys use wide-section V-belts to obtain speed ratios up to 3 to 1 (see Figs. 8.8 to 8.10). They are available in single- and multiple-groove designs from fractional to 100-hp rating.

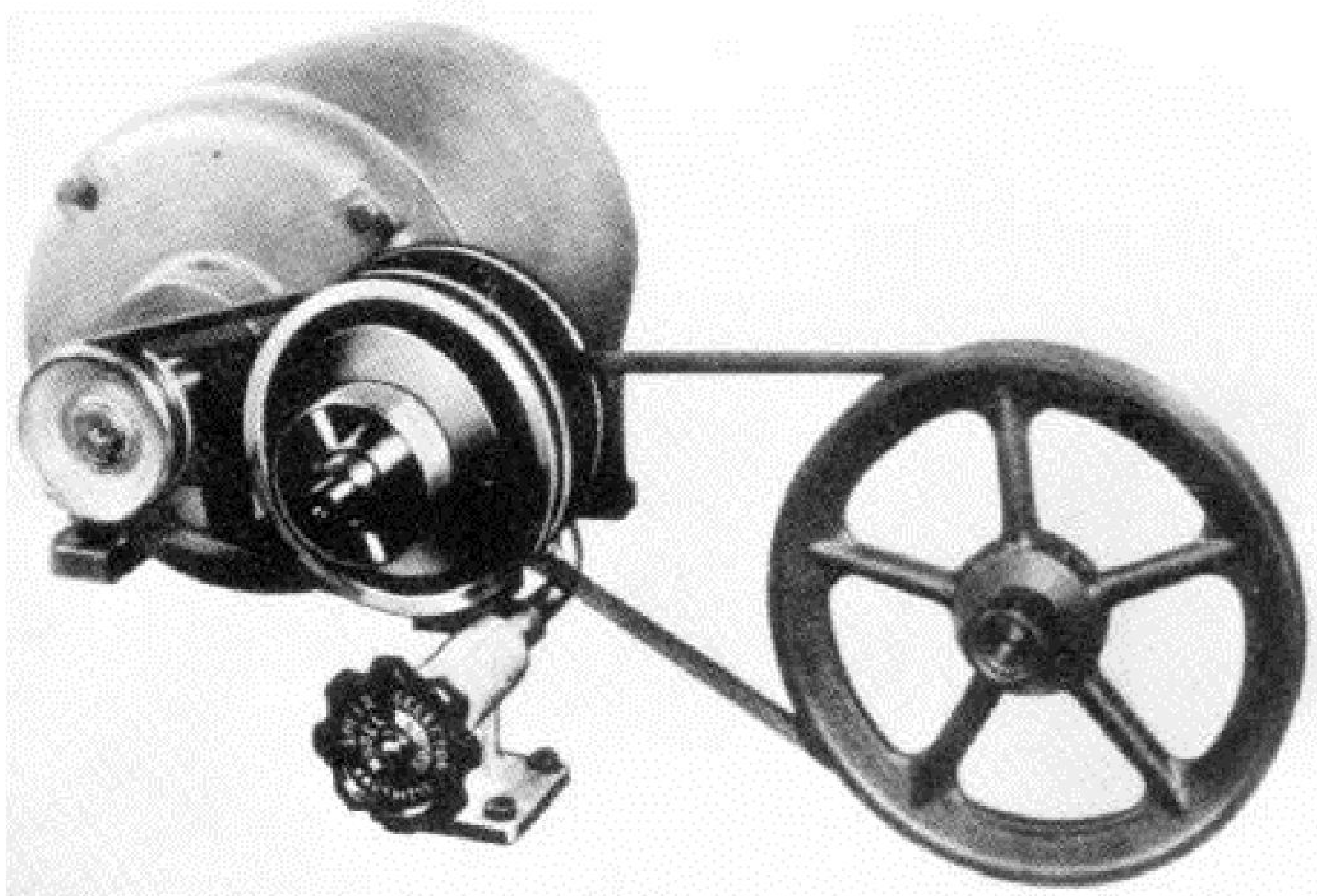

FIGURE 8.6 Use of compounded variable-pitch sheave to obtain wide speed range. (*Speed Selector, Inc.*)

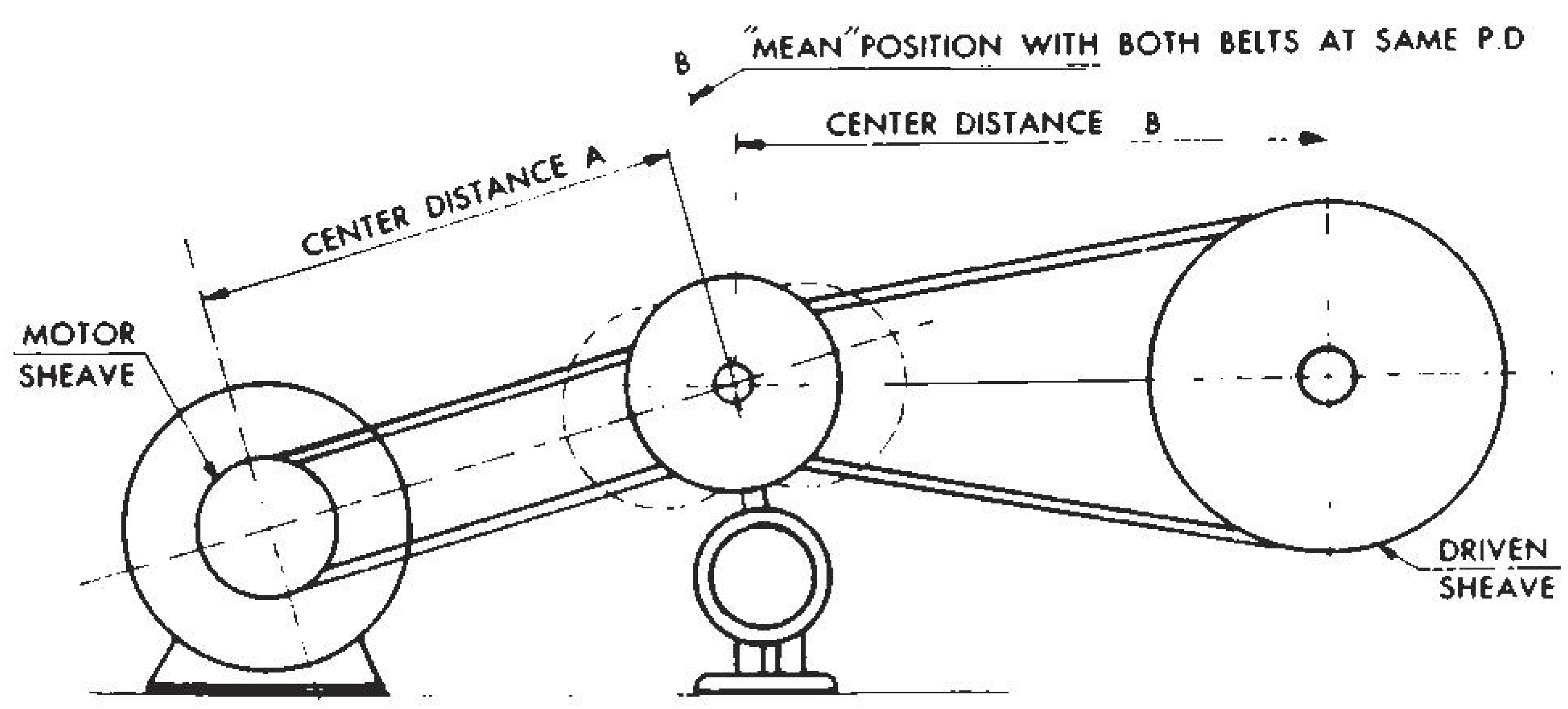

FIGURE 8.7 Diagrammatic scheme of drive shown in Fig. 8.9.

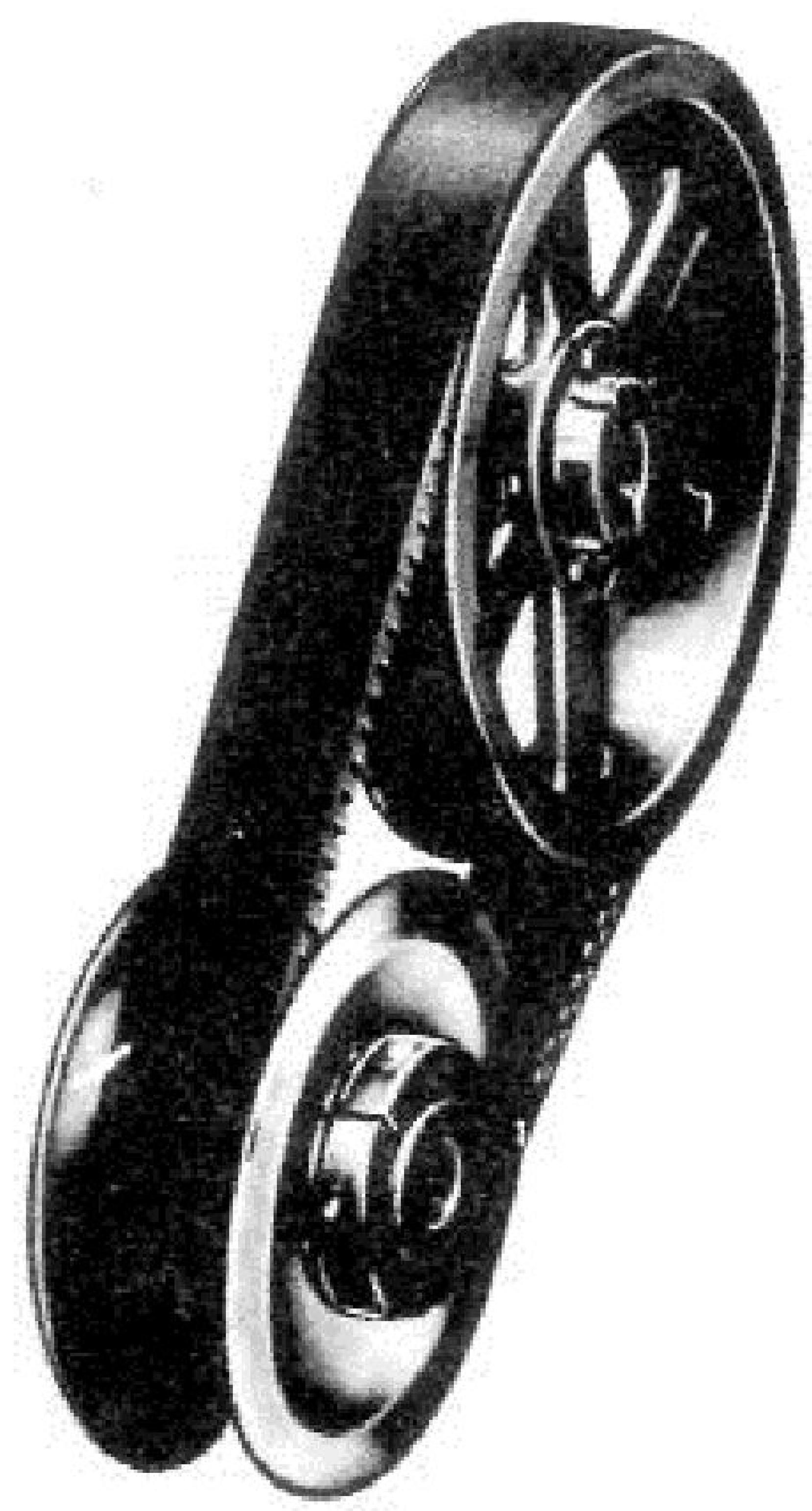

FIGURE 8.8 Adjustable-while-stopped control, speed ratios up to 3 to 1. (*T. B. Wood's Sons Co.*)

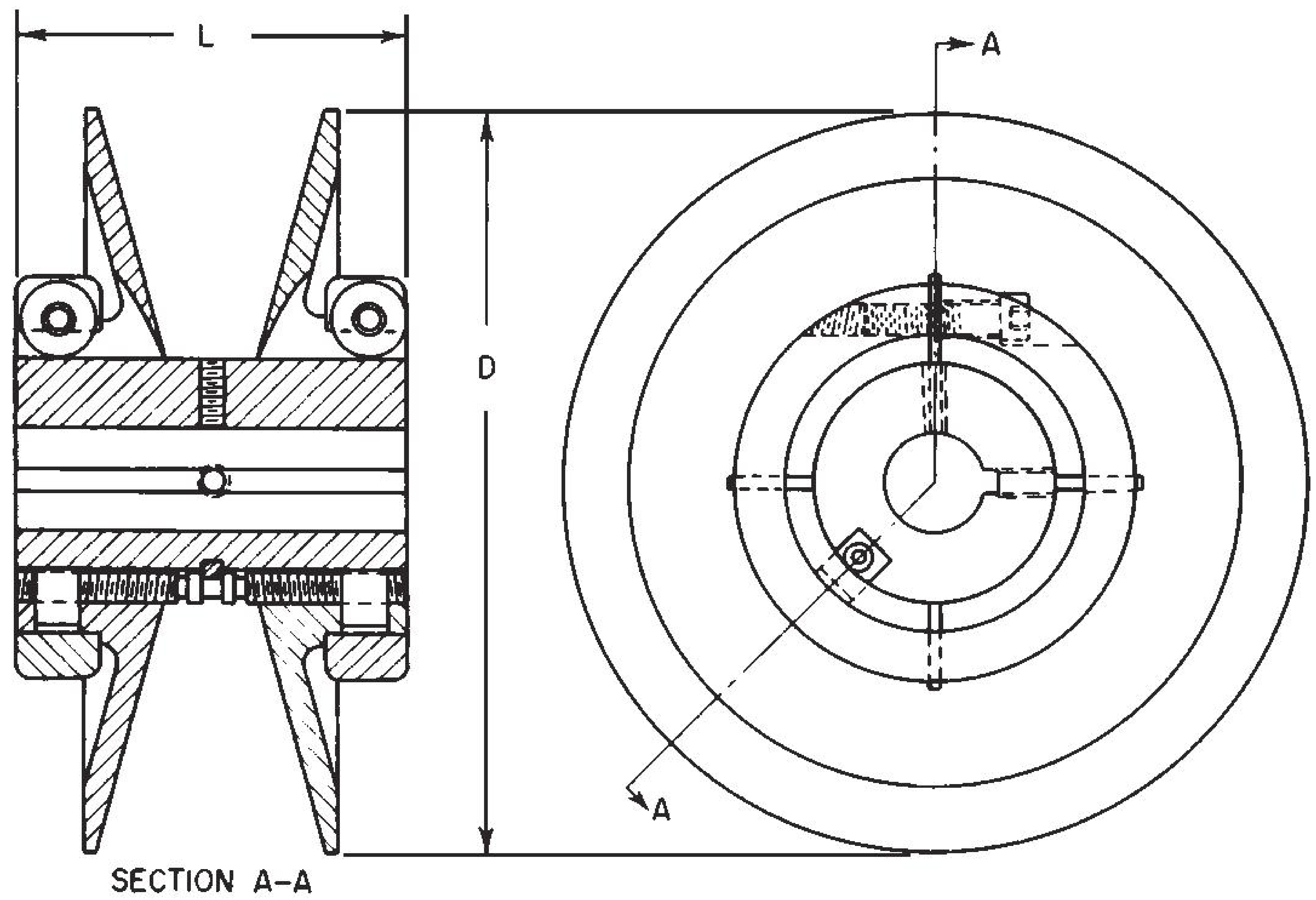

FIGURE 8.9 Section of control shown in Fig. 8.3.

Of significance to the maintenance man is the fact that once adjusted, the various parts are designed to be held together in a fixed relation one to another by set screws or holding nuts, thus minimizing the wear and lubrication problems mentioned above. Care must be exercised to ensure that the holding devices are quite secure, for even the slightest oscillatory motion in the absence of lubrication will quickly result in a seizing of the mating surfaces. (See Table 8.1.)

Speed adjustment can be done only while the machine is stopped. The most critical problem with wide-section multiple-belt pulleys is the matching of belt lengths to ensure a sharing of the load.

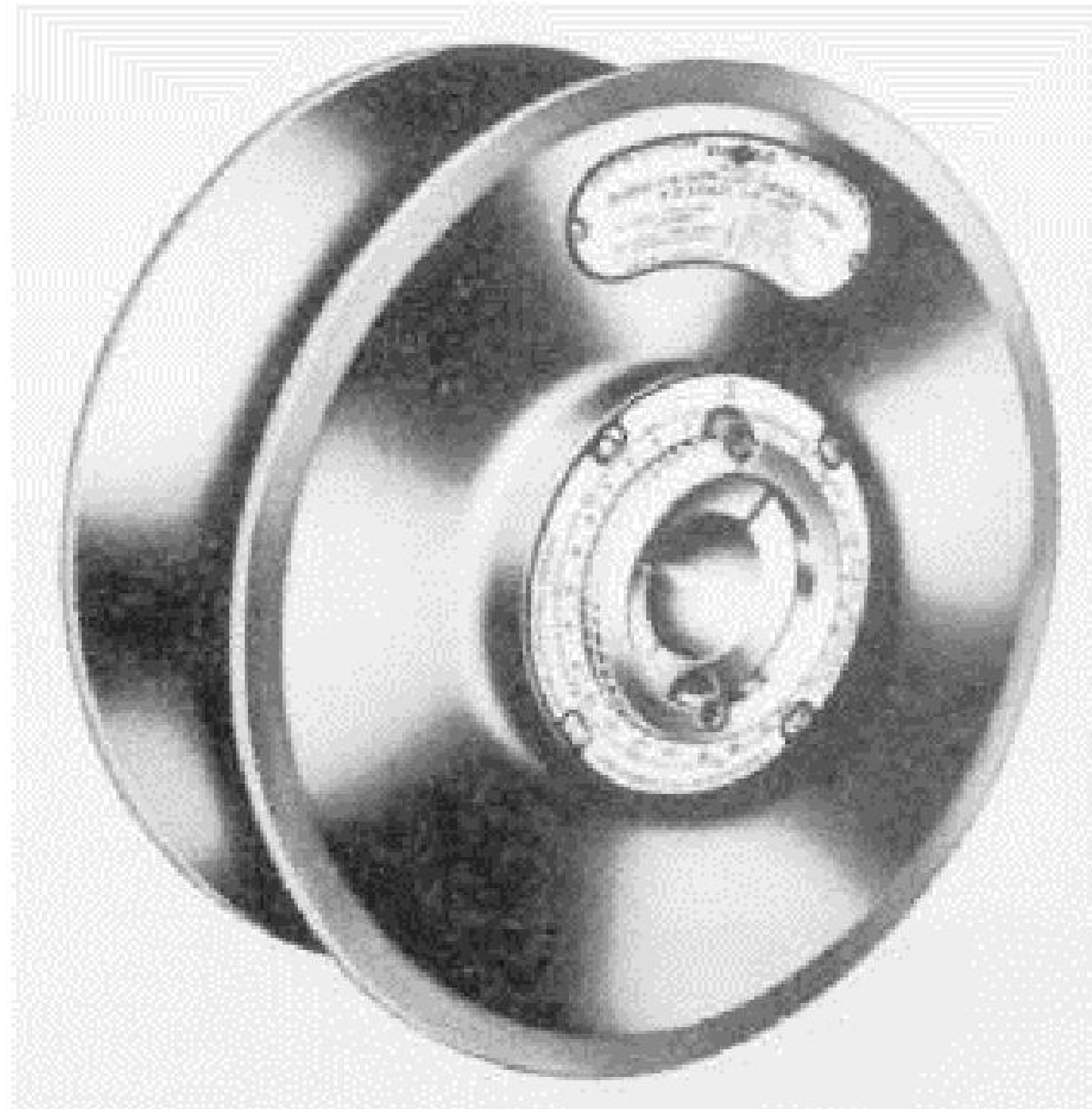

FIGURE 8.10 Adjustable-while-stopped control, speed ratios up to 3 to 1. (*Dodge Manufacturing Corp.*)

TABLE 8.1 Adjustable Pulleys—Stationary Control

Trouble	Cause	Correction	Prevention
I*a*. Accelerated belt wear	Excessive misalignment of driver and driven pulleys	Realign pulleys	
I*b*. Accelerated belt wear	Continuous flexing over small diameters	Select driven pulley of proper diameter to avoid this condition.	
I*c*. Accelerated belt wear	Excessive heat, cold, moisture, acid fumes, abrasives, etc.		
I*d*. Accelerated belt wear	Overloaded, excessive shock loads, excessive belt speeds		Do not exceed ratings set by manufacturers
I*e*. Accelerated belt wear	In multiple-belt pulleys one or a few belts taking entire load	Belt lengths must be matched more closely	
I*f*. Accelerated belt wear	Excessive belt tension	Adjust center distance	
I*g*. Slipping belt	Insufficient tension	Adjust center distance, or use idler roll	
I*h*. Slipping belt	Pulley faces greasy	Clean	
I*i*.	Slipping belt	Overloaded	

Unfortunately, these must be matched more closely than standard tolerances set up by the belt manufacturers, which call for a trial-and-error matching of belts.

This, together with the definite stretch a belt will take its first several hundred hours of operation, calls for replacement of the complete set of belts when any one particular belt goes bad. Belts can be purchased as matched sets.

This tendency for the belts to stretch during their initial break-in, along with the changing of operating diameters, requires some method of take-up to maintain proper belt tension, which is usually accomplished by adjusting the motor position on slide rails or by some type of idler roll.

Table 8.2 lists troubles, their causes, and cures experienced with the adjustable pulleys, stationary control.

TABLE 8.2 Adjustable Pulley—Controlled in Motion

Trouble	Cause	Correction	Prevention
II*a*. Accelerated belt wear	See items I*a,* I*b,* I*c,* I*d**		
II*b*. Slipping belt	See items I*h,* I*i**		
II*c*. Slipping belt or belt not running level	Spring-loaded disks do not compensate owing to sticking of disks from improper or insufficient lubrication	Stop at once; disassemble; clean until parts slide freely	Lubricate every 1–4 weeks; shift speed range each day if possible

Belt Transmissions

Belt transmissions of the type shown in Fig. 8.11 have, through the years, earned a most enviable reputation among plant maintenance men for ruggedness and reliability with a minimum of attention. They are available in capacities from fractional to 75 hp and up to 16 to 1 speed range. Standard variations include vertical or horizontal mountings, open or enclosed, and with a variety of controls.

The heart of these transmissions is the time-honored block-belt design. Wedge-shaped wooden blocks tipped with leather are bolted to and carried on a wide strip of belting. This design has the advantage of separating the handling of the torque load (belt pull) and the radial wedging forces. Little attention is required to the belt other than an occasional check to ensure that the disk faces are clean and free from grease, acid, or water. If adverse conditions of dust, water, chemical fumes, or live steam are present, an enclosed type of transmission should be used. Table 8.3 lists troubles, causes, and cures experienced with belt transmissions.

Usually, one shaft is driven at constant speed, speed adjustment being accomplished by a lever arrangement which positively synchronizes the position of all four flanges. Two screw arrangements are provided: one by adjusting speed by controlling the position of these levers, which, in turn, control the pulley operating diameters, the other, for controlling belt tension and horsepower capacity, by adjusting the center distance of the pivotal points of these synchronizing levers.

Belt tensioning is accomplished in at least two different ways. Older designs rely completely on the natural wedging action of the belt between the disks. The belt-tensioning screw (located between the disk sets and acting on the pivot points of the shifting levers) must be adjusted while the

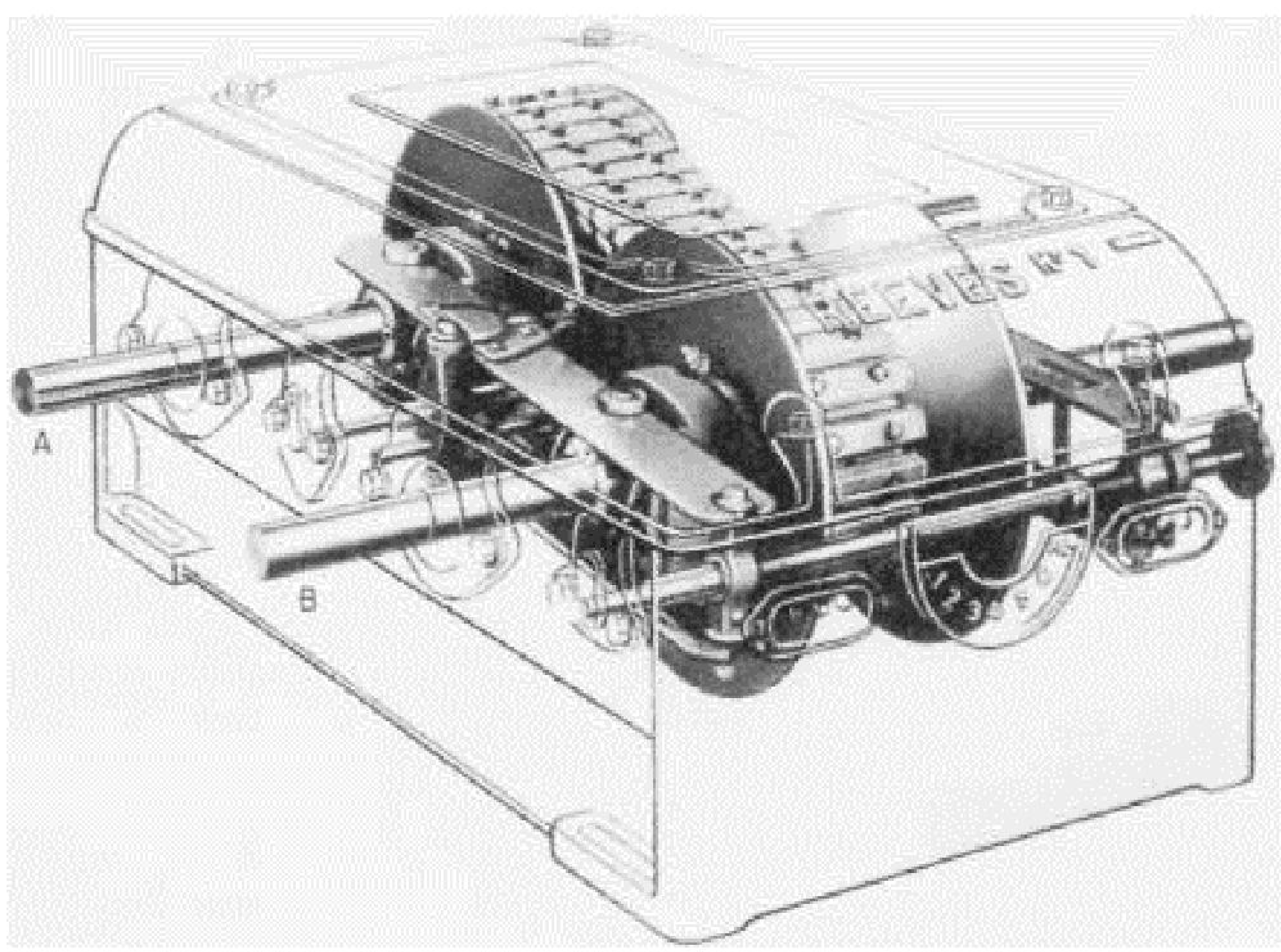

FIGURE 8.11 Belt transmission.

TABLE 8.3 Belt Transmission

Trouble	Cause	Correction	Prevention
III*a*. Accelerated belt wear	Misalignment of disk assembly	Constant-speed and variable-disk assemblies should be parallel at mean speed	
III*b*. Accelerated belt wear	See also items I*b*, I*c*, and I*d**		
III*c*. Slipping belt	Pulley faces greasy, usually from overlubrication of thrust bearings	Clean	Avoid overgreasing thrust bearings
III*d*. Slipping belt	Constant-speed shaft too slow	Increase input speed by changing sheaves	
III*e*. Slipping belt	Insufficient belt tension	Adjust tension screw, but only while drive is running	Belt should have a slight sag on the loose side
III*f*. Creaking belt	Excessive belt tension	Adjust tension screw, but only while drive is running	
III*g*. Bearing failures	Belt too tight	Adjust tension	
III*h*. Bearing failures	Excessive overhung load		Do not exceed loads specified by manufacturer
III*j*. Bearing failures	Insufficient or excessive lubrication		See manufacturer's instructions
III*k*. Bearing failure	Atmosphere: abrasive particles, moisture, corrosion	Use enclosures where necessary	
III*l*. Bearing failures	Bent shaft or improperly assembled		
III*m*. Cannot adjust speed	"Sticking disks" due to improper or insufficient lubrication	Stop at once; disassemble; clean disk hub and shaft with solvent	Lubricate every 2–5 weeks; shift through entire speed range each day, if possible

*Reference is to items in Table 8.1.

drive is running. The proper adjustment is only to where the belt has some slack or droop on the loose side. Do not pull the belt up tight; to do so may destroy the belt.

Newer designs tension the belt with short, strong springs acting on the pivot points of the shifting levers. These springs are adjusted with the tensioning screw. These are normally adjusted by tightening the tensioning screw until it stops (tight) while rolling the drive over by hand and then backing the screw off a turn or two.

Note that transmissions all require periodic disk and bearing lubrication. Drives should be inspected carefully for the presence of grease fittings, since some have internal bearing lube points which also require service. As with pulleys, transmissions should be lubricated while running and shifted through the speed range to distribute the lubricant.

All-metal versions of the belt transmissions are also available. These are totally enclosed, oil-filled units that use a belt made up of linked transverse laminations (Fig. 8.12). These laminations mate with radial grooves on the conical pulley inner flanges. These units offer higher power and positive speed at the expense of greater mechanical complexity and more sensitivity to shock and overload than block-belt types. Adequate lubrication and internal alignment are extremely critical to drive life.

Packaged Belt Drives

These drives are very common and find wide use in many industries because of their all-in-one compactness and versatility. Similar in concept and function to the previously described compound-pulley arrangement, they go further by incorporating motor and variable-speed pulleys with control and

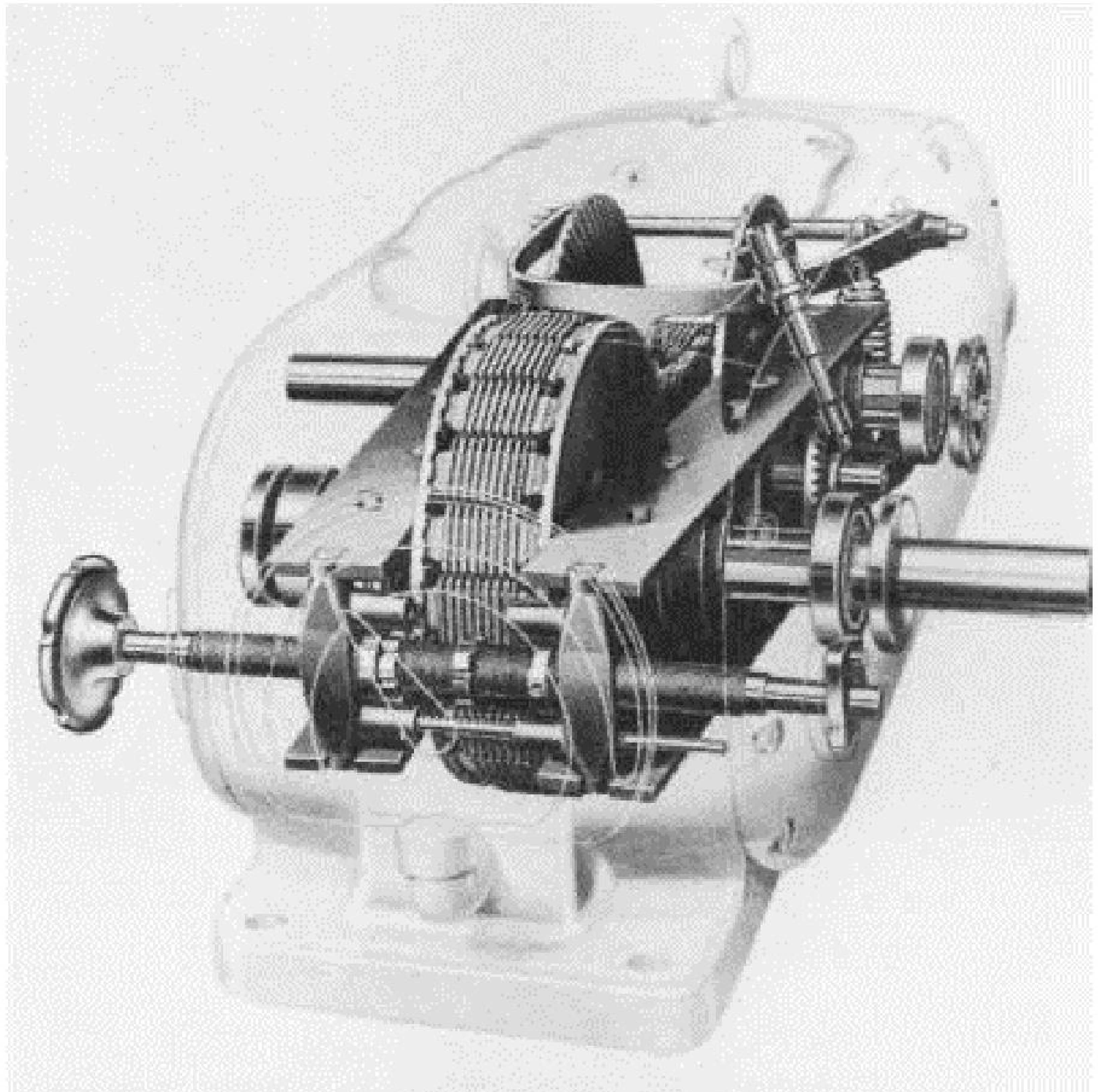

FIGURE 8.12 All-metal belt transmission—P.I.V. (*Link-Belt Div., FMC Corp.*)

gear reducer in one package. The available combinations of horsepower, case size, output speeds, controls, and physical arrangements seem limitless—a manufacturer such as Reeves can provide over 10,000 combinations, from ¼ to 50 hp at speeds from less than 1 to over 13,000 rpm. Both parallel and right-angle reducers are available. Control arrangements range from manual handwheels through electrical remote control and pneumatic arrangements. These may be tailored to provide output speeds that follow an input signal in any relationship—linear, logarithmic, or exponential.

Maintenance of packaged belt drives is similar to that for open pulleys. Both relubricatable and permanent-lube designs exist, with perm-lube predominating newer drives. However, the population of older drives is very large; therefore, all drives should be inspected for periodic maintenance requirements. Also, some motor bearings have lube fittings. Those units with gear reducers must have oil levels checked and oil replaced periodically. Grade and type of oil should follow manufacturer's specifications. Older units commonly use SAE gear oil or nondetergent crankcase oil. Nondetergent oil is preferred because it allows contaminants to settle out into the bottom of the gearcase rather than being held in suspension in the oil. Oil grade is usually a function on temperature. Newer parallel reducers (and most right-angle worm and worm-helical combinations) may require synthetic or specific lubricants. Table 8.4 lists troubles, causes, and cures for packaged belt drives.

As packaged units, these drives normally require little attention. If disassembly is necessary (internal parts can be easily replaced), all parts should be inspected before reassembly. Most drives require little realignment, the exception being correctly locating the constant-speed disk assembly on the motor shaft before tightening it down if the motor or constant-speed disk assembly must be replaced. Some types locate the constant-speed disk assembly relative to the motor shaft with spacers or adjusting screws in the bottom of the fixed constant-speed disk motor bore.

Common practice has been to mount the fixed constant-speed disk directly on the motor shaft (the belt pull becomes an overhung load on the motor bearings) and securing it with either a key and set screws or with a clamp collar or collet arrangement. The latter types may not have a keyway. These types must never be used on a damaged or undersized motor shaft because they may not hold adequately.

Some designs support the constant-speed disk assembly between bearings and connect to the motor with a flexible coupling.

TABLE 8.4 Packaged Belt Drives

Trouble	Cause	Correction	Prevention
IV*a.* Accelerated belt wear	See items I*b,* I*c,* I*d,* III*a**		
IV*b.* Slipping belt	Broken spring in variable-speed disk assembly	Replace spring	
IV*c.* Slipping belt	Pulley faces greasy owing to excessive lubrication of thrust bearings	Clean with solvent	
IV*d.* Bearing failures	See Items III*h,* III*j,* III*k,* III*l**		
IV*e.* Cannot adjust speed; belt not running level	See item III*m*		
IV*f.* Chatter in gearing	Insufficient oil in gear case		Check oil level every 30 days

*Reference is to items in Tables 8.1 and 8.3

European designs take advantage of the tapped-hole standard available in the end of IEC-type motors. Disk assemblies are secured to the motor shaft with a draw bolt extending completely through the length of the hub of the fixed constant-speed disk and into the tapped motor shaft. Instruction manuals should be consulted for critical measurements. Failure to set these properly will result in premature belt and disk wear.

As with other mechanical devices, a little preventive maintenance and inspection will yield a good return in life, performance, and freedom from downtime.

Friction-Disk-Type Drives

These devices employ an old principle updated. Two disks are used. One is a flat or slightly angled disk coupled directly to and driven at constant speed by the motor. The second disk is usually an annular ring of replaceable friction material attached to a carrier shaft that is supported in bearings at a slight angle from parallel to the first disk. Only a “patch” (theoretically a radial line segment) or friction material actually contacts the driver disk face.

Speed change is effected by altering the contact area radius of the annular ring from the center line of the driver (constant-speed) disk. Single-disk-pair (narrow speed range) drives usually solidly mount the bearings for the output annular friction ring. The motor is usually slide mounted so that it can be moved radially. Speed range is set by stops that limit the radial travel of the motor slide.

Wide-speed-range friction drives may have two disk and ring pairs. See Fig. 8.13 for principles of operation. Drives are usually packaged with the motor and can be obtained with gear reducers.

Maintenance consists of regular bearing lubrication and periodic inspection and cleaning of disk assemblies. The most important maintenance consideration is to ensure that all friction surfaces are absolutely clean, dry, and free from any contaminants. Even fingerprints can cause performance degradation. Some incorporate a torque-sensing (disk-loading) cam arrangement that is subject to wear and also should be checked regularly. Friction disks should be replaced when they become worn, following the manufacturer’s recommended minimum thickness.

Premature disk or friction material wear may be due to shock loading that causes skidding, grooving, or flat spots.

Traction-Type Drives

These differ from previously discussed friction types by relying on power transmission through an oil film which microscopically separates extremely hard metal elements rather than through direct contact between parts.

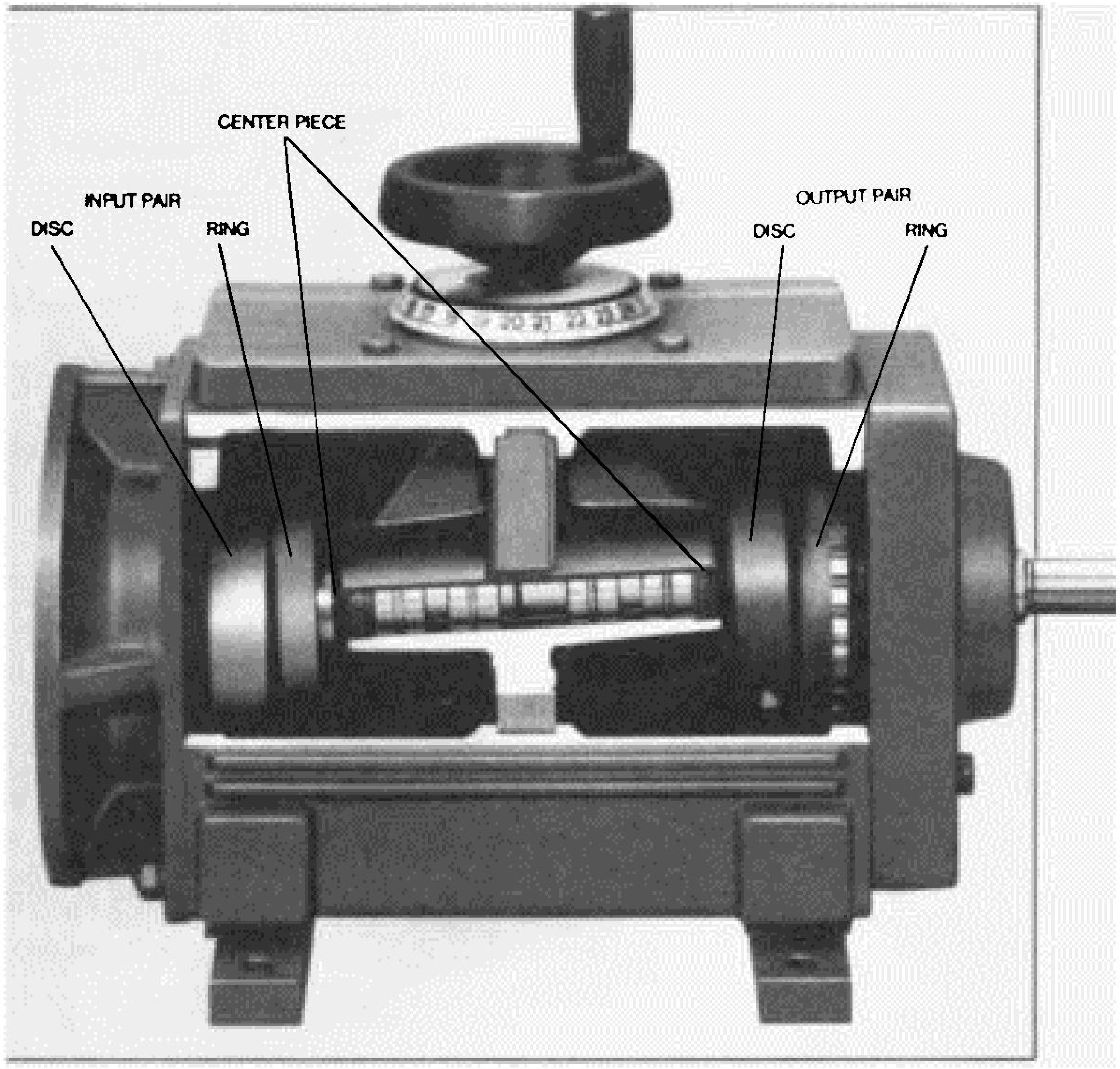

FIGURE 8.13 Wide-speed range friction—disk-type drive. (*Reeves Div.*)

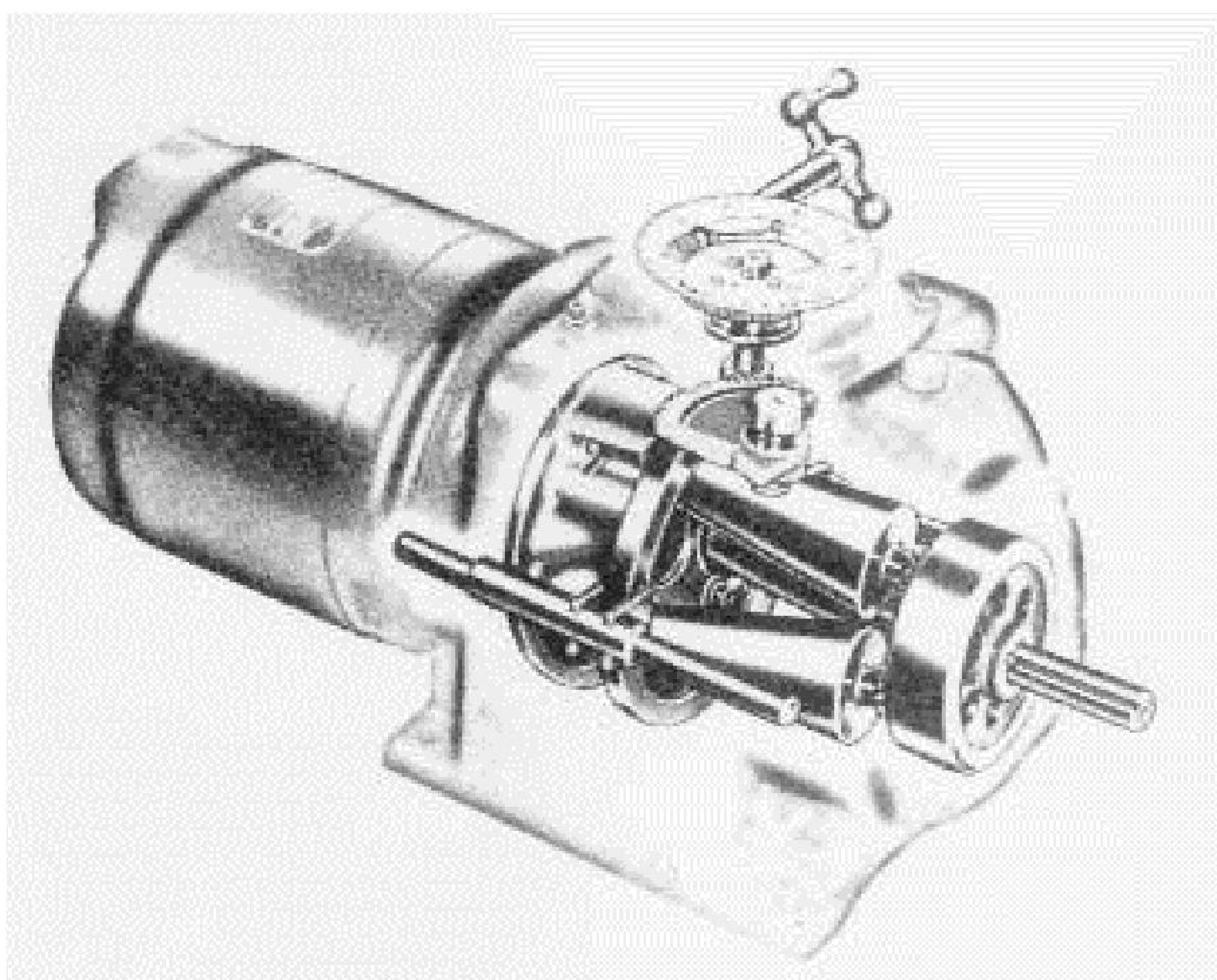

FIGURE 8.14 Drive with metal surfaces in frictional contact. (*Graham Transmission, Inc.*)

These metal elements may take the form of balls, cones, disks, or rings. Two such designs are shown in Figs. 8.14 and 8.15. Traction drives are built to extreme precision and, as such, are very sensitive to rigid maintenance schedules and correct application. Lubricant levels, operating conditions, and temperature are critical. All use specially developed traction fluids which can serve as lubricants but are not normally interchangeable. Internal repair should only be done by specialists.

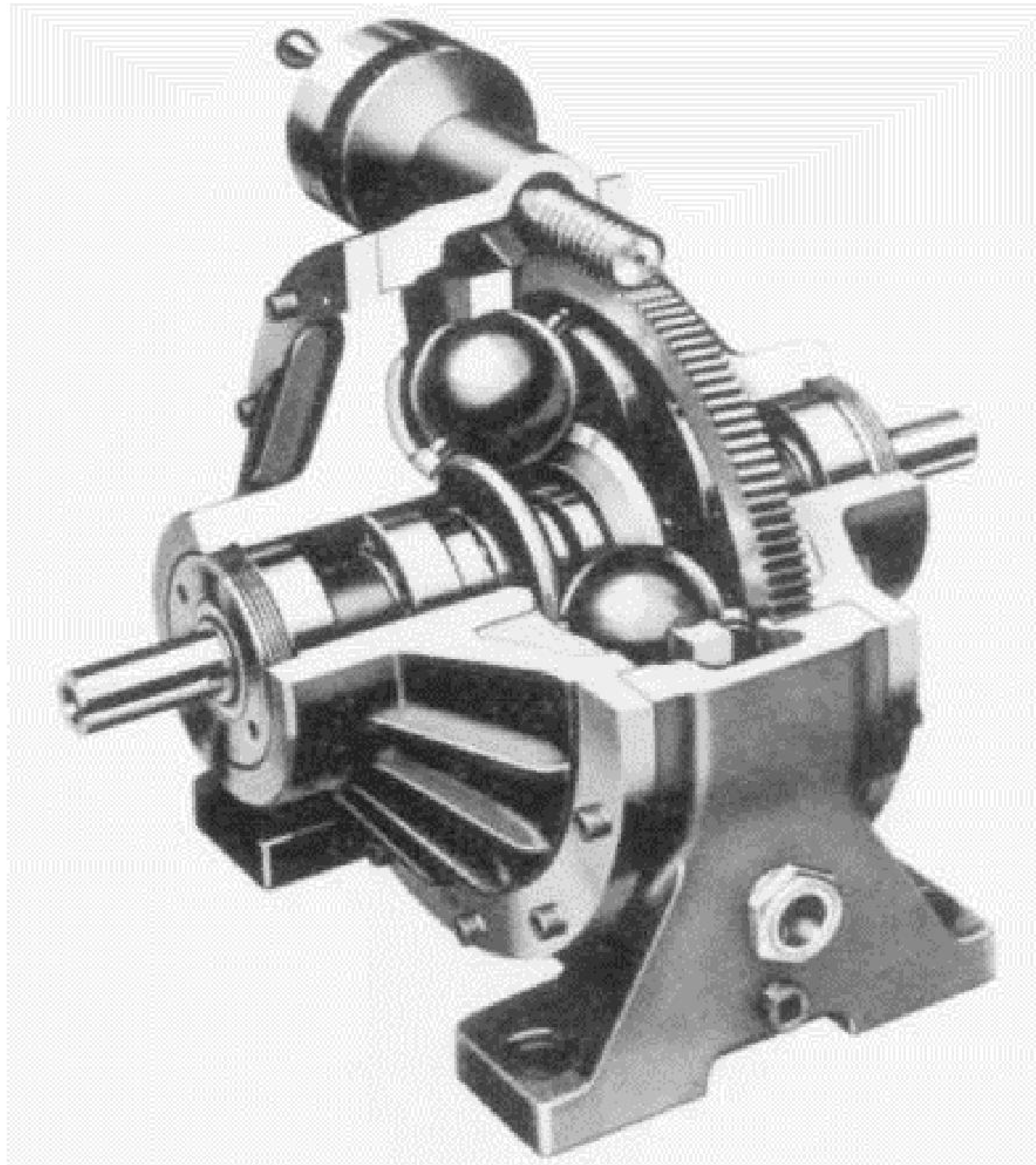

FIGURE 8.15 Drive with metal surfaces in frictional contact. (*The Cleveland Worm & Gear Co.*)

TABLE 8.5 All-Metal Traction Systems

Trouble	Cause	Correction	Prevention
V*a.* Cannot shift speed; OK if shifted only slightly; will tend to slide back to original setting	Extended operation at one speed has caused concentrated wear tending to lock the sliding surfaces in a fixed position	Return to factory for replacement of damaged parts	Avoid running at one speed for extended periods
V*b.* Pronounced thumping	Sudden load change reversal or overload has caused scoring of contact surfaces	Return to factory for replacement of damaged parts Replenish to proper level	Avoid use of traction-type drives on this type of application
V*c.* Severe overheating	Insufficient oil	Flush. Replenish with oil of proper viscosity	
V*d.* Excessive slipping	Use of too heavy an oil		Use appropriately thin oil as recommended by manufacturer. Change every 1000 hours of operation

These drives must operate free of shocks, overloads, reversals, or excessive temperatures. If the fluid film between mating parts is even momentarily interrupted, scoring of surface may result. Metal particles especially will destroy the drive. Table 8.5 lists troubles, causes, and cures for metal traction systems.

Geared Differential Drives

Differential gearing can be attached to any parallel-shaft cone-pulley transmission to permit infinite speed variation down to zero speed and, with proper selection of components, on into the reverse direction. Figure 8.16 shows such a differential integrally mounted with an all-metal belt transmission.

The most important consideration with these geared-differential drives is the proper recognition of the internal circulating power, which may reach as high as six times the input power, and if the motor transmission and gear components are not properly matched, disastrous internal overloading can result. Since each of the components must be of sufficient size and capacity to handle these maximum conditions, the cost of the total unit approaches that of the more familiar electrical variable-speed drives.

For troubles, their causes, and remedies, refer to the tables which give that information for belt transmissions and gear units.

Flat-Belt Drives

Occasionally, a flat belt driving between two cone-shaped pulleys of the type shown in Fig. 8.17 will be found as a method of obtaining variable speed. There is no particular manufacturer who merchandises a complete line of these pulleys; rather, they are holdovers from the very early days or are specially designed to meet particular requirements.

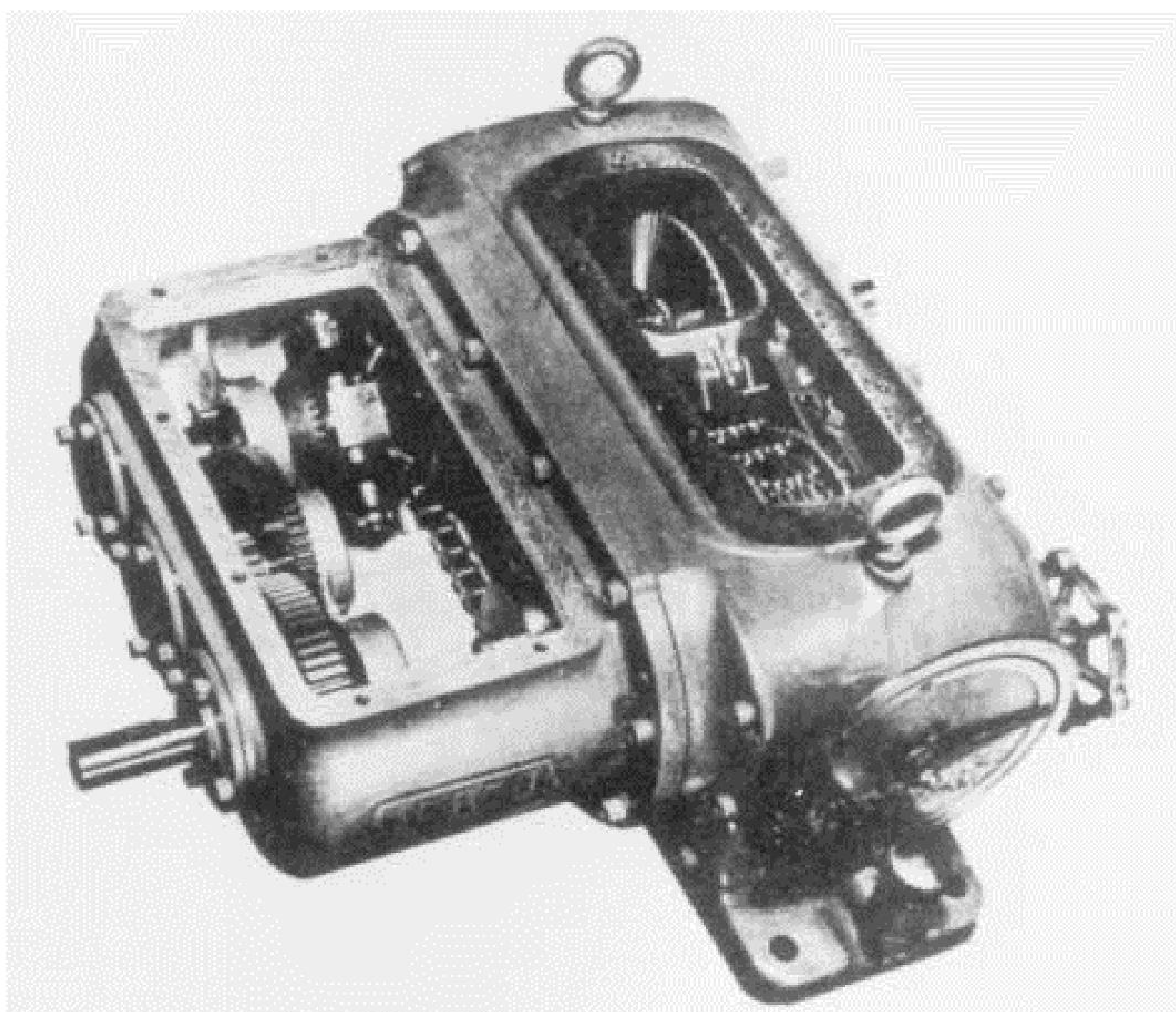

FIGURE 8.16 Differential gearing integrally mounted with all-metal belt transmission. (*Fairchild Engine and Airplane Corp.*)

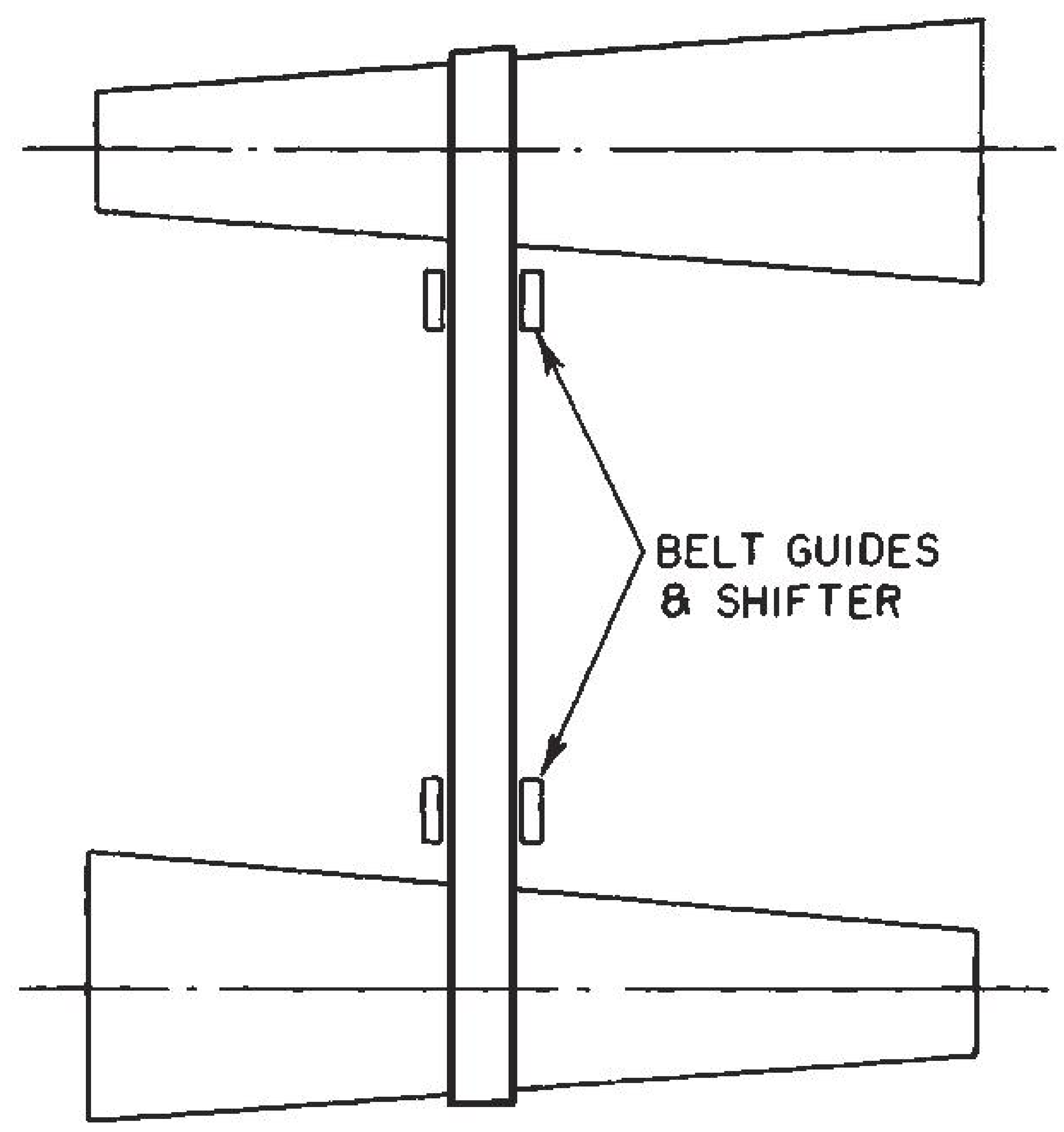

FIGURE 8.17 Flat belt and cone drive.

Line contact is the theoretical ideal and, as such, requires that the belt width be kept quite narrow, which, of course, limits the power-transmitting capacity of the system. This same consideration means that contact surface speed varies across the face of the belt, producing an inherent slip and its consequent effect on belt life. See Table 8.6.

TABLE 8.6 Flat-Belt Drives

Trouble	Cause	Correction	Prevention
Short belt life	Inherent slippage; contact surface speed varies across face of belt	None	
Slipping belt	Overloaded; remember belts are necessarily narrow to approach theoretically ideal line contact	Reduce loading	Do not overload
Slipping belt	Belt has stretched	Extend center distance between pulleys or use idler roll	
Slipping belt	Pulley faces greasy	Clean with solvent	Keep clean

CHAPTER 9
GEAR DRIVES AND SPEED REDUCERS

Robert G. Smith
Director of Engineering, Philadelphia Gear Corporation, King of Prussia, Pa.

Gear drives and speed reducers are widely used where changes of speed, torque, shaft direction, or direction of rotation are required between a prime mover and the driven machinery.

The design approach taken by gear manufacturers today is to consider the gear drive as a component of a mechanical system. Its functional characteristics are engineered to be fully compatible with those of the prime mover and the driven equipment and to take into account such factors as static and dynamic loading, range of torque and operating speed, expected service life, duty cycle, ambient temperature, size and weight restrictions, and total system efficiency.

Gear drives consist of one or more sets of gears mounted on shafts and bearings, a positive method of lubrication, and an enclosed casing with appropriate gaskets, oil seals, and air breathers. They also must be equipped with an integral electric motor, baseplates or other mounting structure, outboard bearings, a device that provides overload protection, a means of preventing reverse rotation, and a variety of other accessory devices.

In many power-transmission applications, the preferred prime mover operates at a relatively high speed because of the superior economy and efficiency of high-speed motors, gas and steam turbines, etc. The driven equipment, however, often requires a much lower shaft speed and high torque. The gear drive not only reduces shaft speed to the value needed to operate the driven machine but also converts the relatively low torque output of the high-speed prime mover to the high torque needed to drive the low-speed driven device.

Rotary compressors operate more efficiently at high shaft speed and often require a speed-increasing gear drive. In most cases, increasers are not simply speed reducers driven backward but involve design considerations different from those encountered in reducers.

Common Gear Types

Common types of gears used in industrial gear drives include spur, helical, or double-helical, bevel, spiral bevel, hypoid, zerol, worm, and internal gears (see Fig. 9.1).

Spur gears transmit power between parallel shafts without end thrust or axial displacement. They are commonly used on drives of moderate speeds such as marine auxiliary equipment, hoisting equipment, mill drives, and kiln drives. Simplicity of manufacture, absence of end thrust, and general economy of maintenance recommend the use of spur gearing wherever practicable.

Helical gear teeth are cut on a helix (oblique) angle across the gear-wheel face. Mating helical gears permit several teeth to be in mesh at the same time. This increases load-carrying capacity,

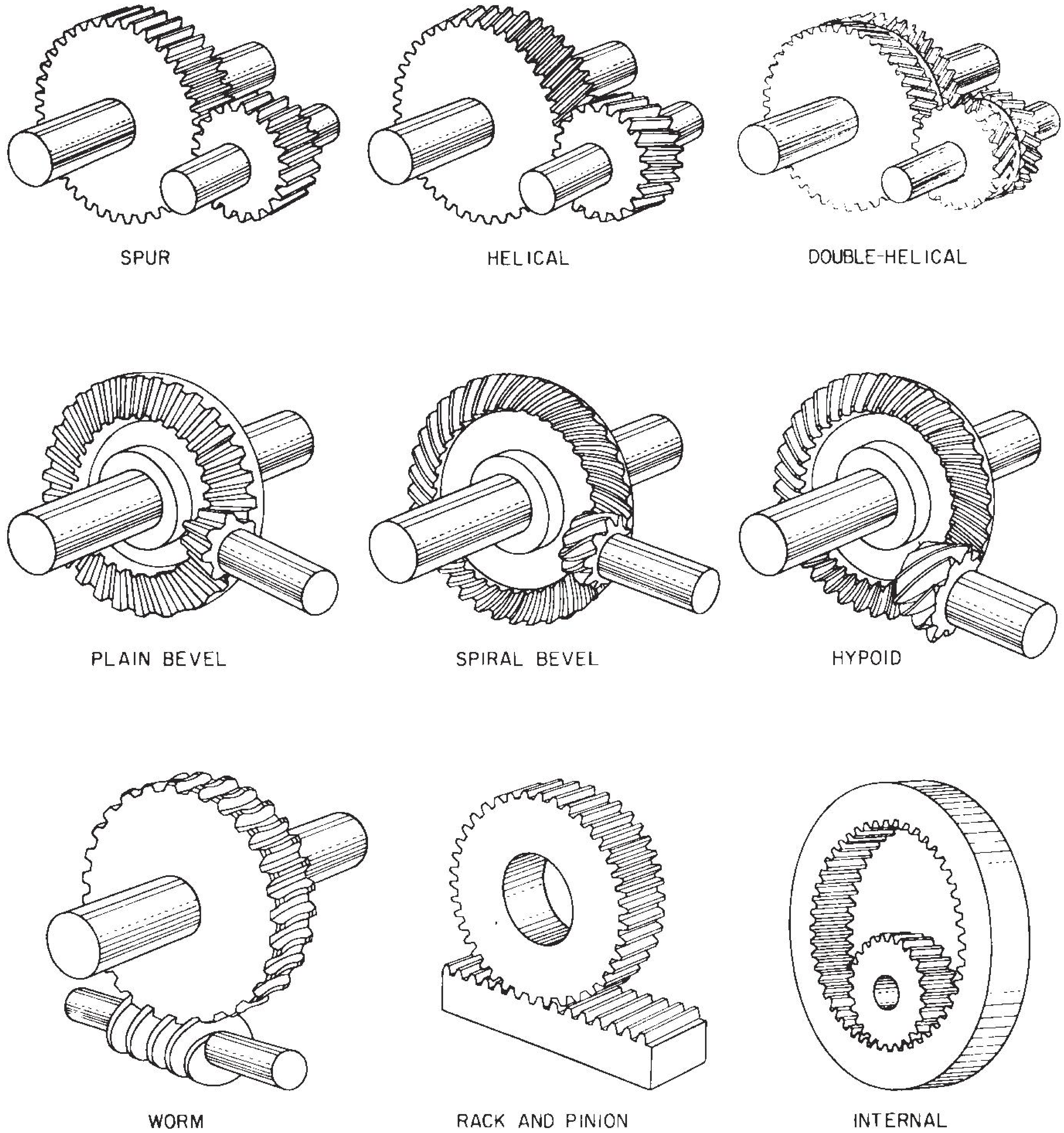

FIGURE 9.1 Basic types of gears.

ensures transmission of constant velocity, and reduces noise and vibration. Helical gears produce end thrust along the axis of rotation, which must be accommodated by thrust bearings.

Helical and double-helical gears are used where loads and speeds may be higher than can be conveniently met by spur gearing. They are also used where shock and vibration are present, or where a high reduction ratio is necessary in a single gear train. Because double-helical gears are actually opposed helical gears, end thrust is practically eliminated. See Fig. 9.2 for examples of double-helical drives.

However, where external thrust loads or thermal expansion of elements in a system are involved, single-helical gears are preferred. External thrust loads may tend to unload one helix of double-helical gears and thus overload the opposite helix.

Where very high speeds (over 20,000 fpm) are involved, critical tolerances of relative tooth positions in each helix may cause the apex of the double-helical tooth arrangement to "run out," which would tend to cyclically unload one helix and set up axial vibrations of the pinion.

Bevel gears transmit power between two shafts, usually at right angles with each other. However, shafts positioned at other than 90° can be used. Straight bevel gears may be used for right-angle

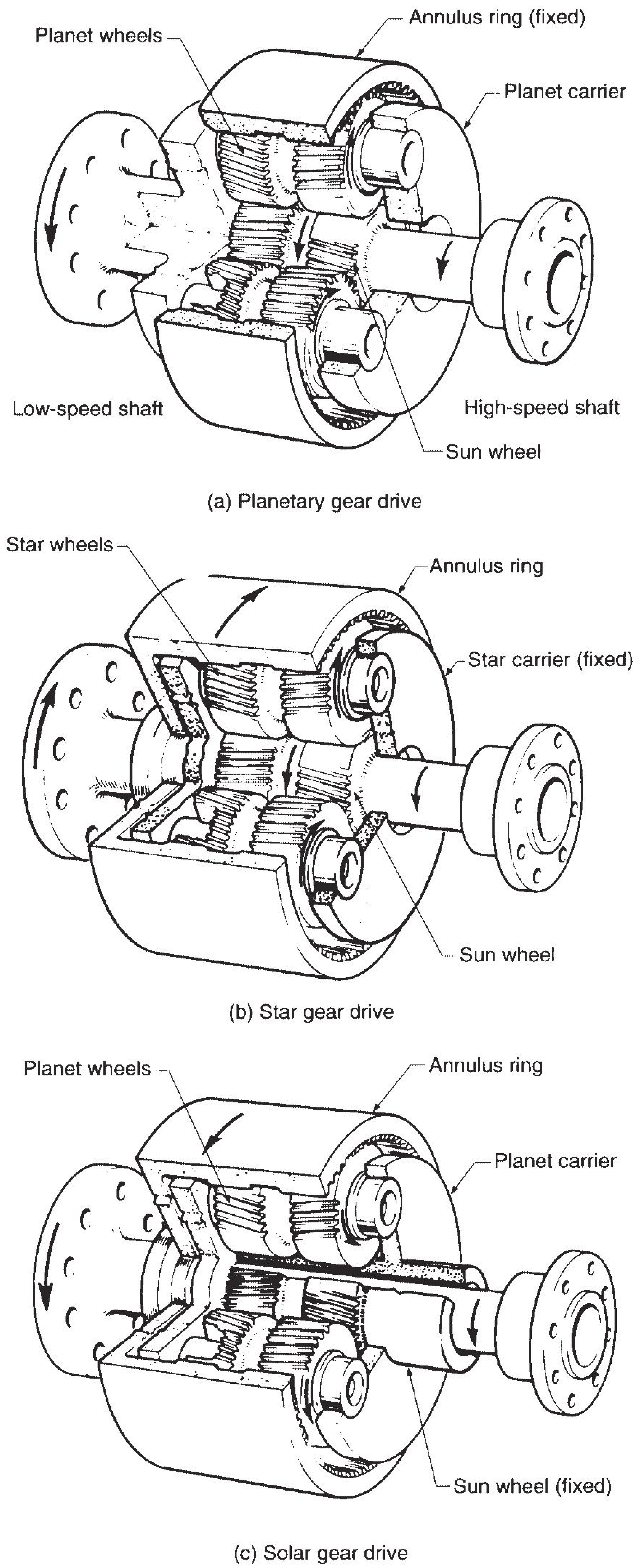

FIGURE 9.2 Double helical drives.

power transmission where operating conditions do not warrant the superior characteristics of spiral bevel gearing. Since bevel gearing creates thrust loads along the supporting shafts, adequate bearings must be provided.

Spiral bevel, zerol, and hypoid types of gears are generally considered under the heading of spiral-bevel-gear units. Loading of spiral bevel gears is always distributed over two or more teeth. Tooth action is smooth and quiet. Accuracy of tooth contact can be closely controlled through hard cutting, precision grinding, and/or lapping. Axial thrust of spiral bevel gears is slightly higher than for straight bevel gearing and varies with direction of rotation and hand of cut of the gear and pinion. Where possible, gears should be designed so that axial thrust tends to move the pinion out of mesh.

Worm gearing has won wide acceptance for industrial drives because of its many advantages of conjugate tooth action, arrangement, compactness, and load-carrying capacity. Worm-gear drives are quiet and vibration-free and produce a constant output speed. They are well suited to service where heavy shock loading is encountered. The many variable mounting arrangements possible with worm gears allow for compactness of design not otherwise obtainable. Since action between worm thread and the teeth of the driven worm-gear wheel is predominantly sliding rather than rolling, greater heat generation and reduced mechanical efficiencies result at higher speed-reduction ratios.

Internal gears are more compact than external gears of the same ratio. In general, they have greater load-carrying capacity and run more smoothly. Internal gearing usually employs spur, helical, or double-helical teeth. Owing to the nature of their construction, internal gears are limited in speed-reduction ratios obtainable on a given center distance.

Basic Gear Drives

Gear drives are used to transmit power between a prime mover and driven machinery. In addition to the simple transmission of power, gear drives usually change or modify the power being transmitted by (1) reducing speed and increasing output torque, (2) increasing speed, (3) changing the direction of shaft rotation, or (4) changing the angle of shaft operation.

Gear drives are generally considered packaged units, manufactured in accordance with accepted and advertised specifications, to be used for a wide range of power-transmission applications (see Figs. 9.3 through 9.11). Published standards of the American Gear Manufacturers Association (AGMA) are accepted as the basis for design, manufacture, and application of modern gear drives.

In addition to general gear drives and speed reducers, AGMA has established specific standards for certain special types of gear drives which are used explicitly for driving particular types of

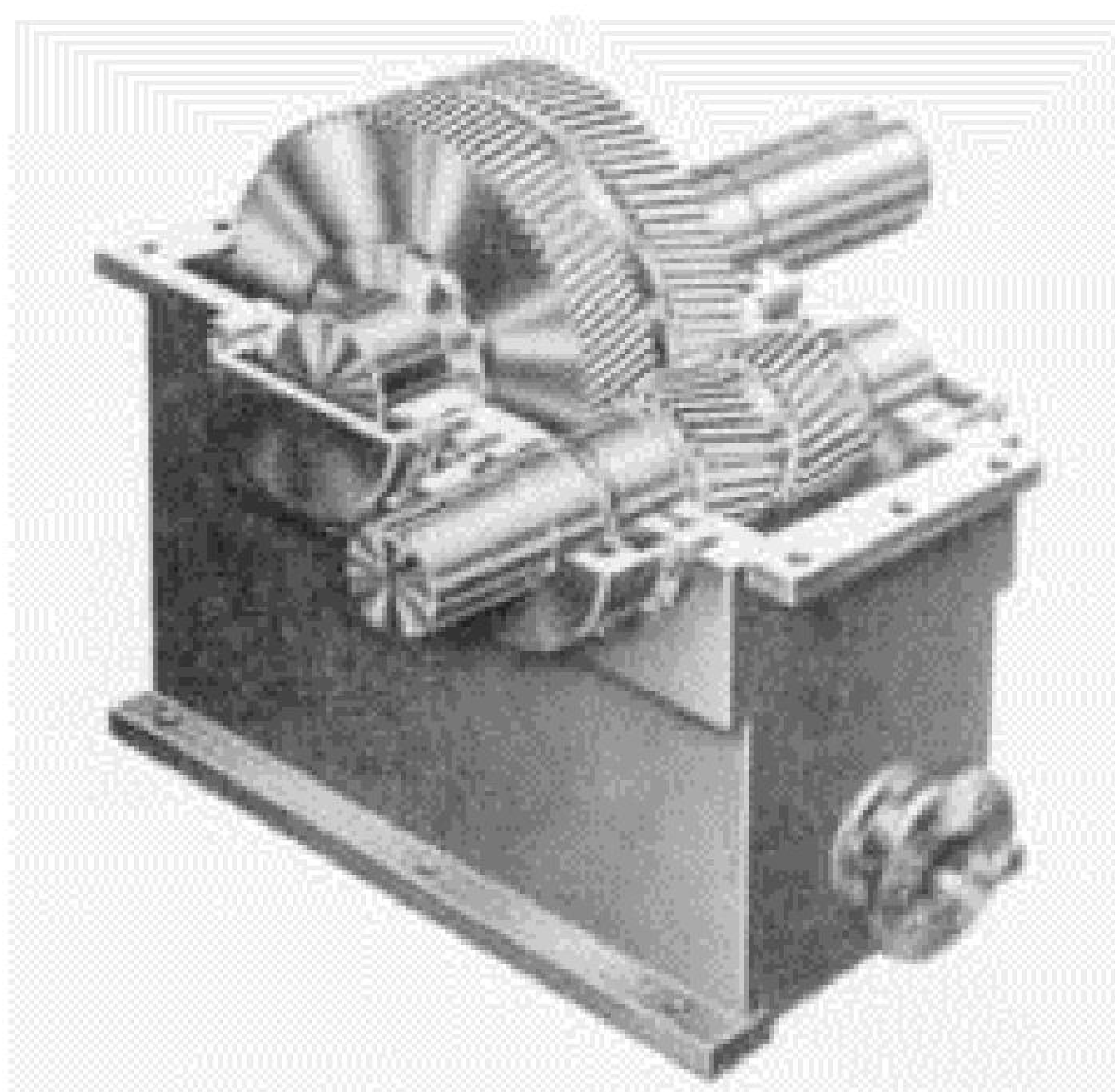

FIGURE 9.3 Single-reduction double-helical drive.

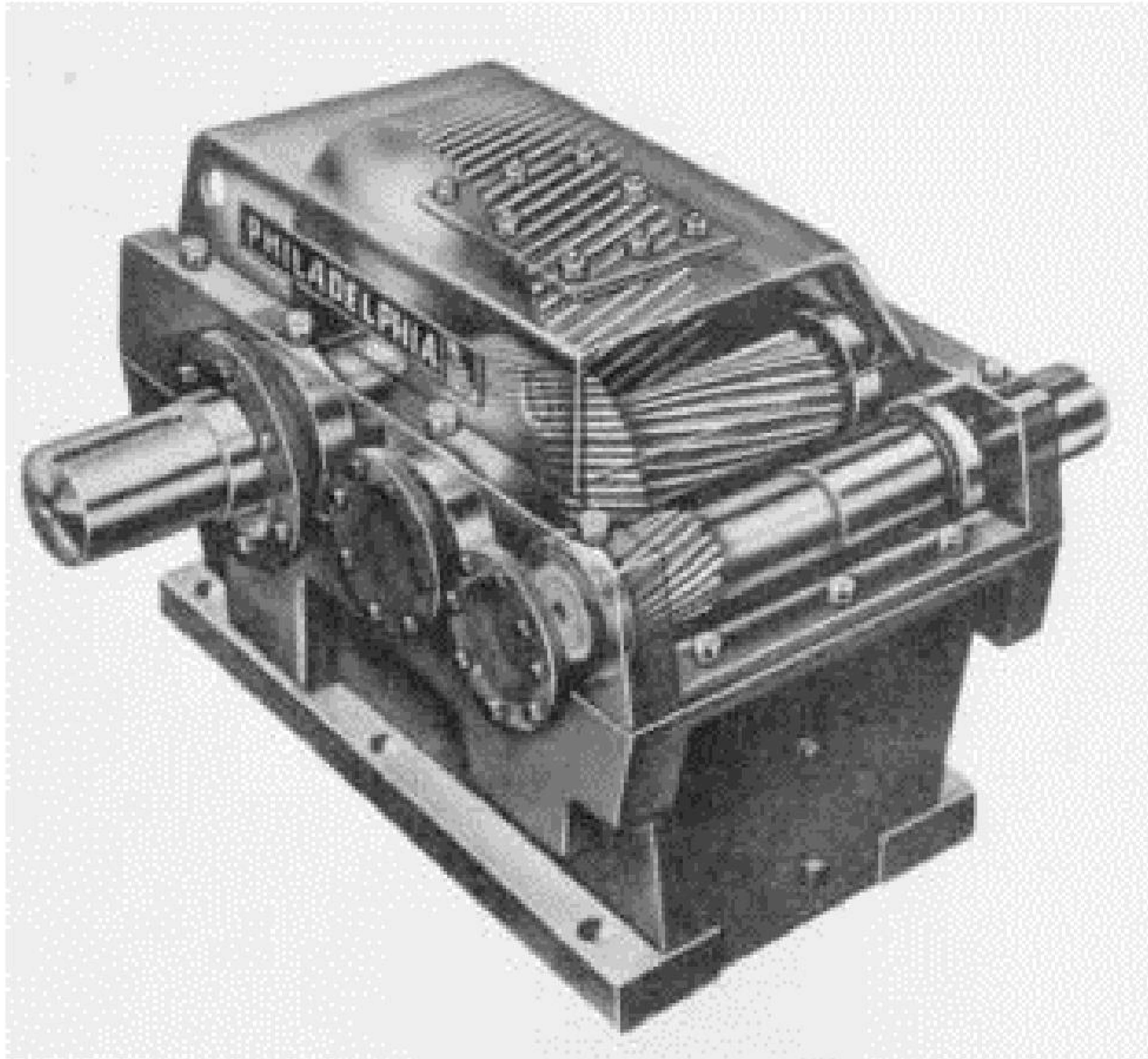

FIGURE 9.4 Double-reduction single-helical drive.

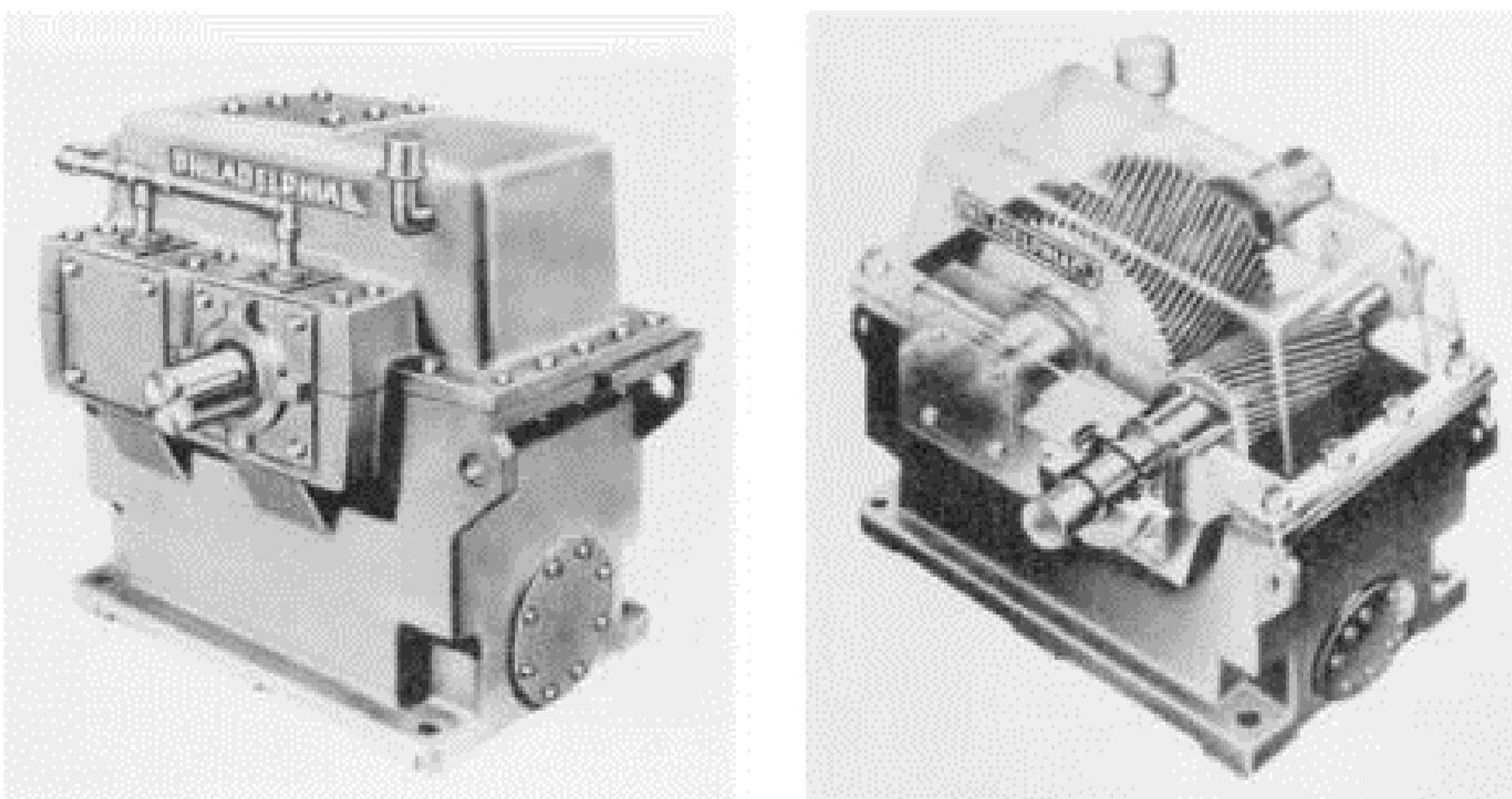

FIGURE 9.5 (*a*) High-speed gearbox, enclosed. (*b*) High-speed gearbox, showing internal parts.

machinery, such as deep-well pumps, cooling-tower fans, steel-mill pinion stand drives, paper-machine sectional drives, cement mills, and compressors.

Motorized gear drives (commonly called *gearmotors* or *motor reducers*) are used extensively throughout industry. These units differ from conventional gear drives in that the prime mover (usually an electric motor) is designed as an integral component of the assembly. Any of the basic gear drives (see Figs. 9.3 through 9.11) can be manufactured as motorized units.

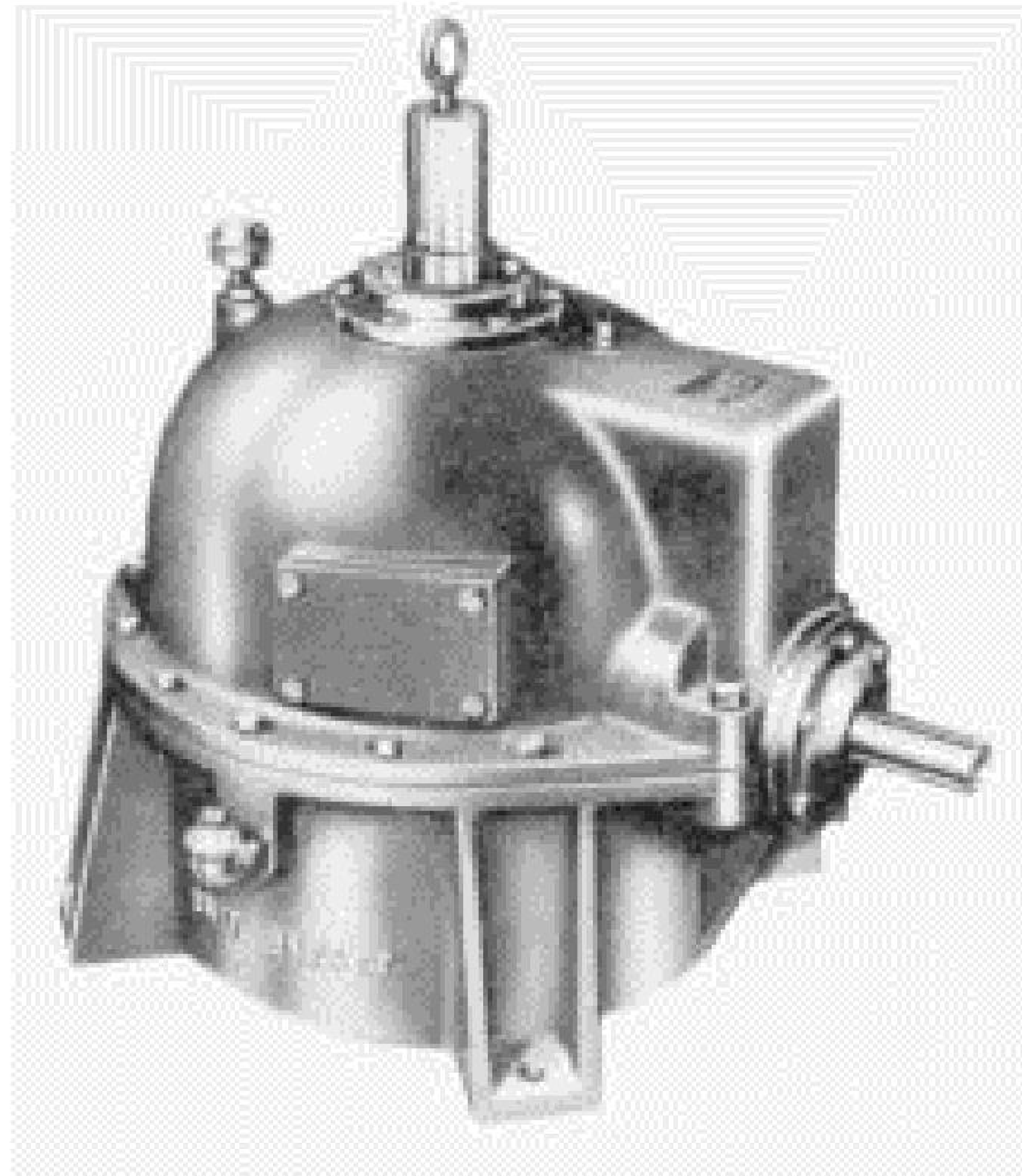

FIGURE 9.6 Cooling-tower drive.

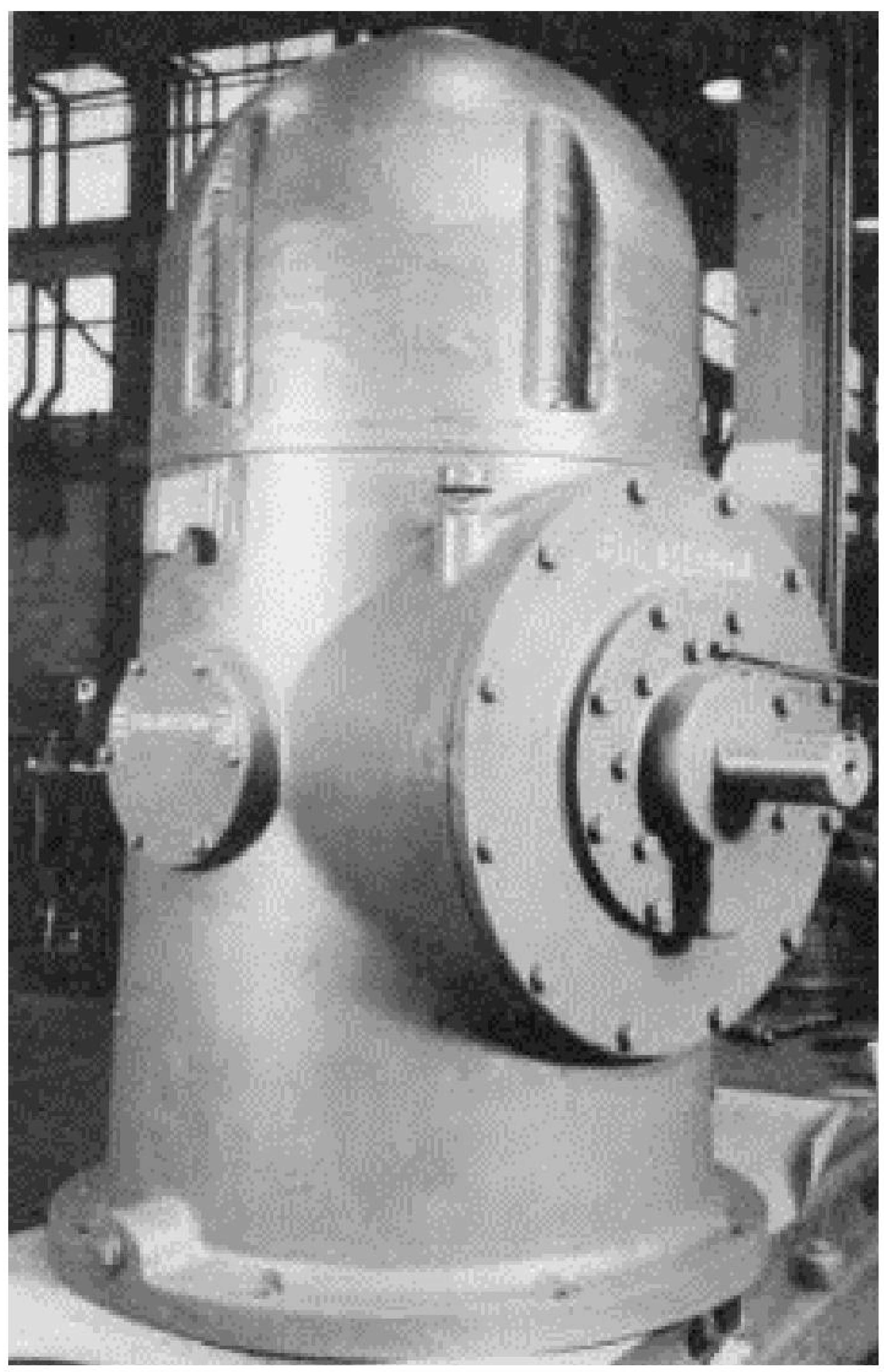

FIGURE 9.7 Vertical pump drive.

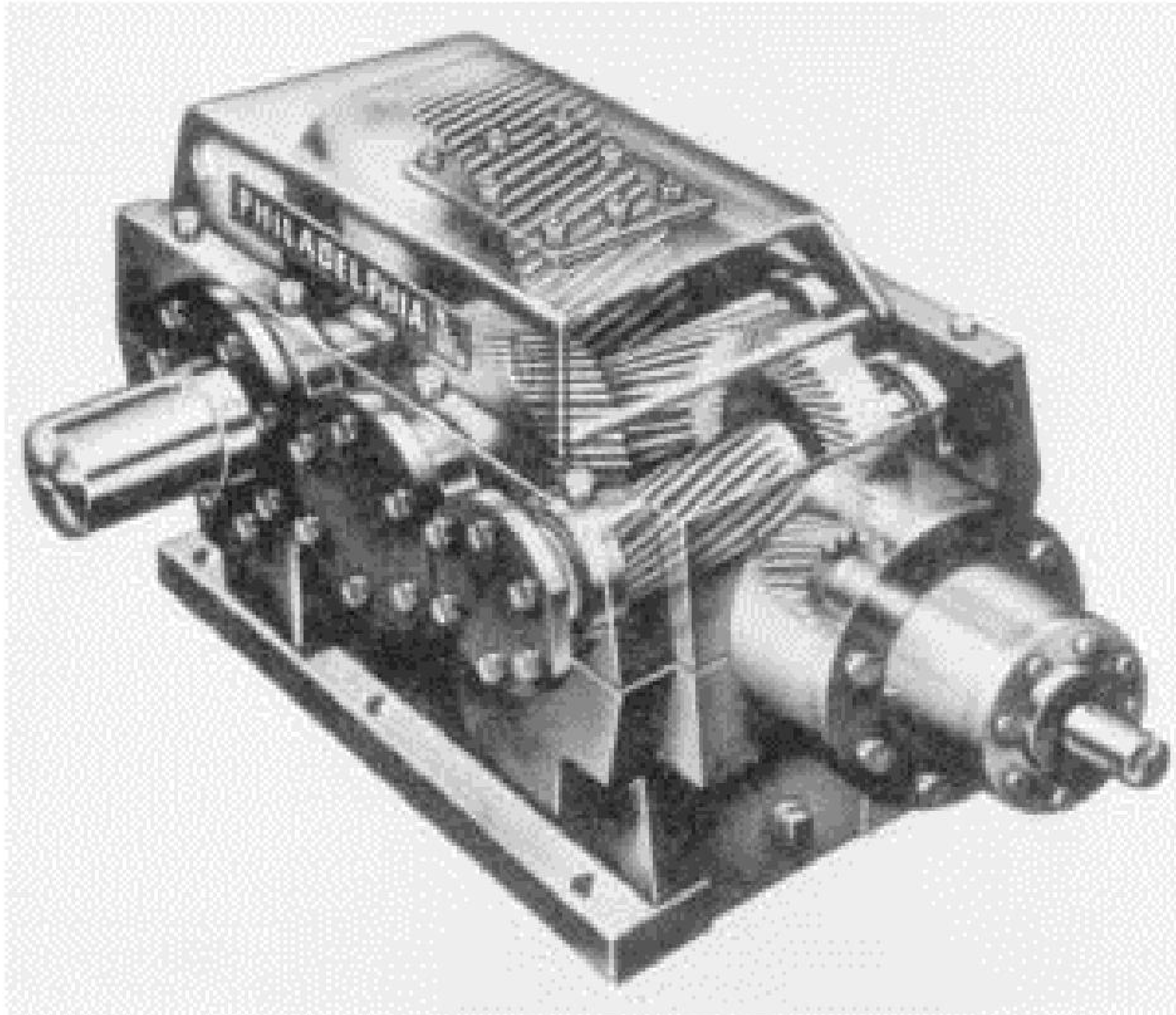

FIGURE 9.8 Triple-reduction spiral-bevel helical drive.

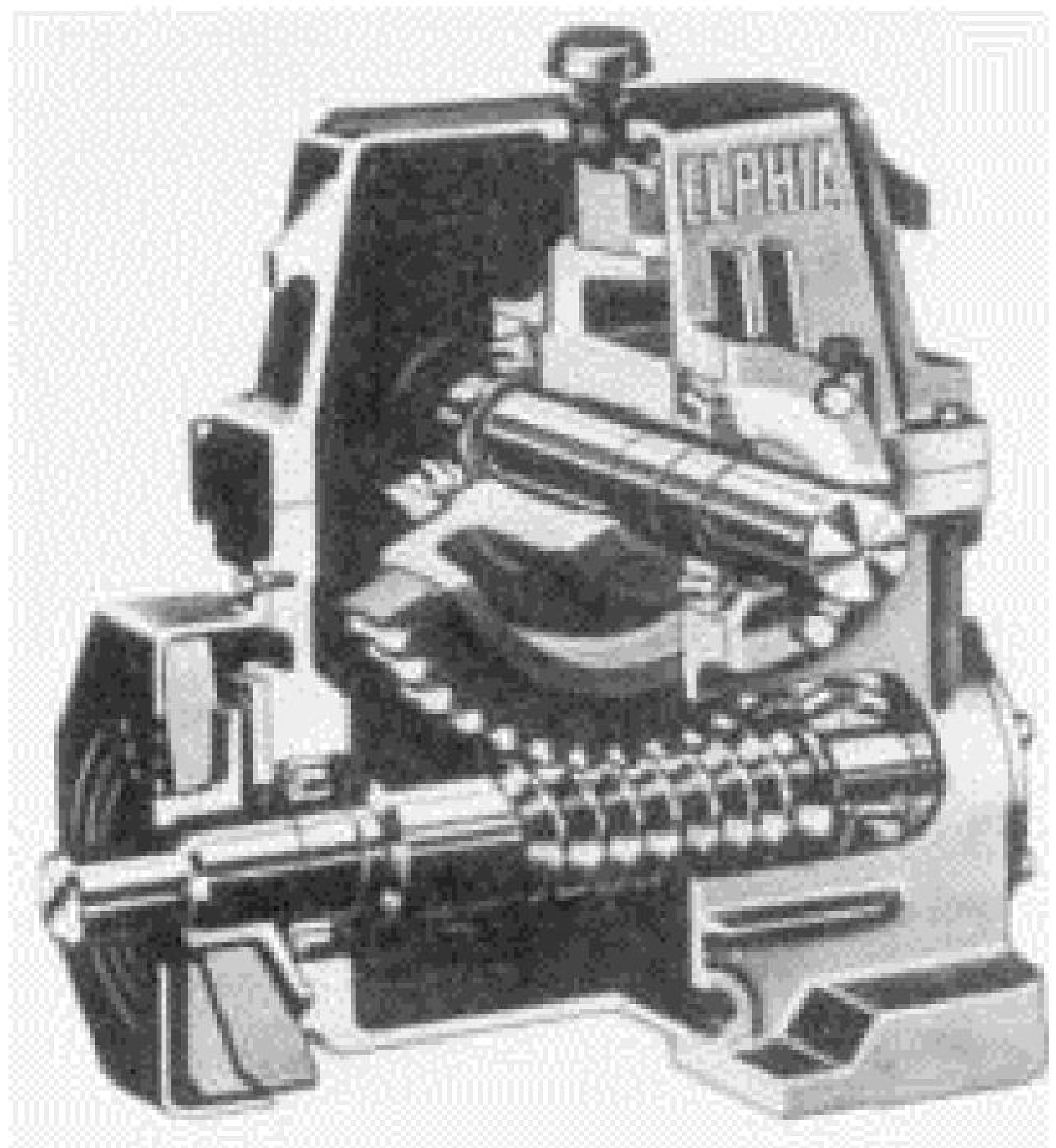

FIGURE 9.9 Single-reduction worm-gear reducer.

Epicyclic Gear Drives

In an epicyclic gear drive, power is transmitted between prime mover and driven machinery through multiple paths. The term *epicyclic* designates a family of designs in which one or more gears move around the circumference of meshing, coaxial gears, which may be fixed or rotating about their own axis. Individual gears within an epicyclic drive may be spur, helical, or double-helical.

Because of the multiple power paths, an epicyclic gear drive will normally provide the smallest drive for a given load-carrying capacity. Other advantages include high efficiency, low inertia for a given duty, high stiffness, and a high torque/power capability.

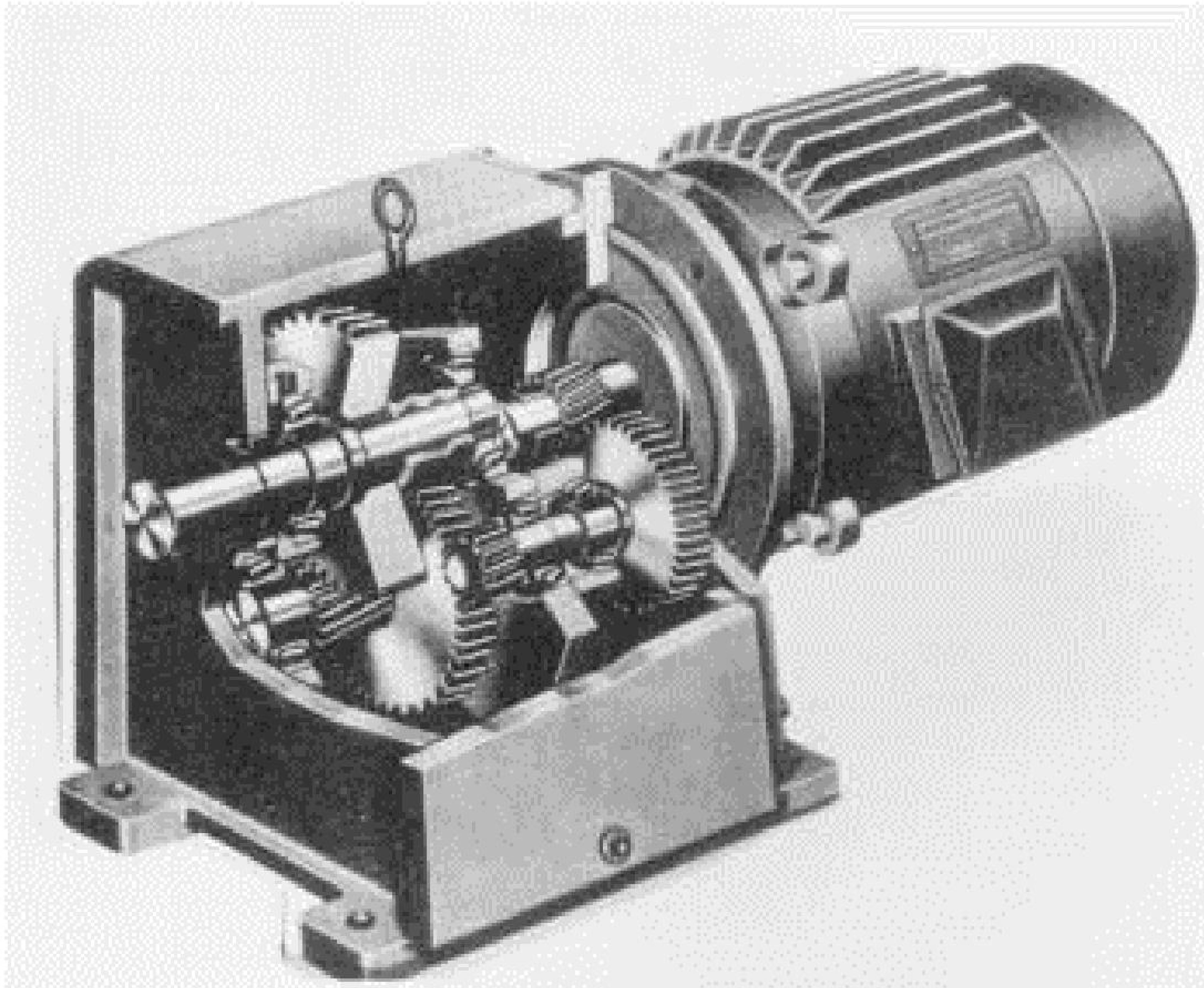

FIGURE 9.10 Double-reduction gearmotor.

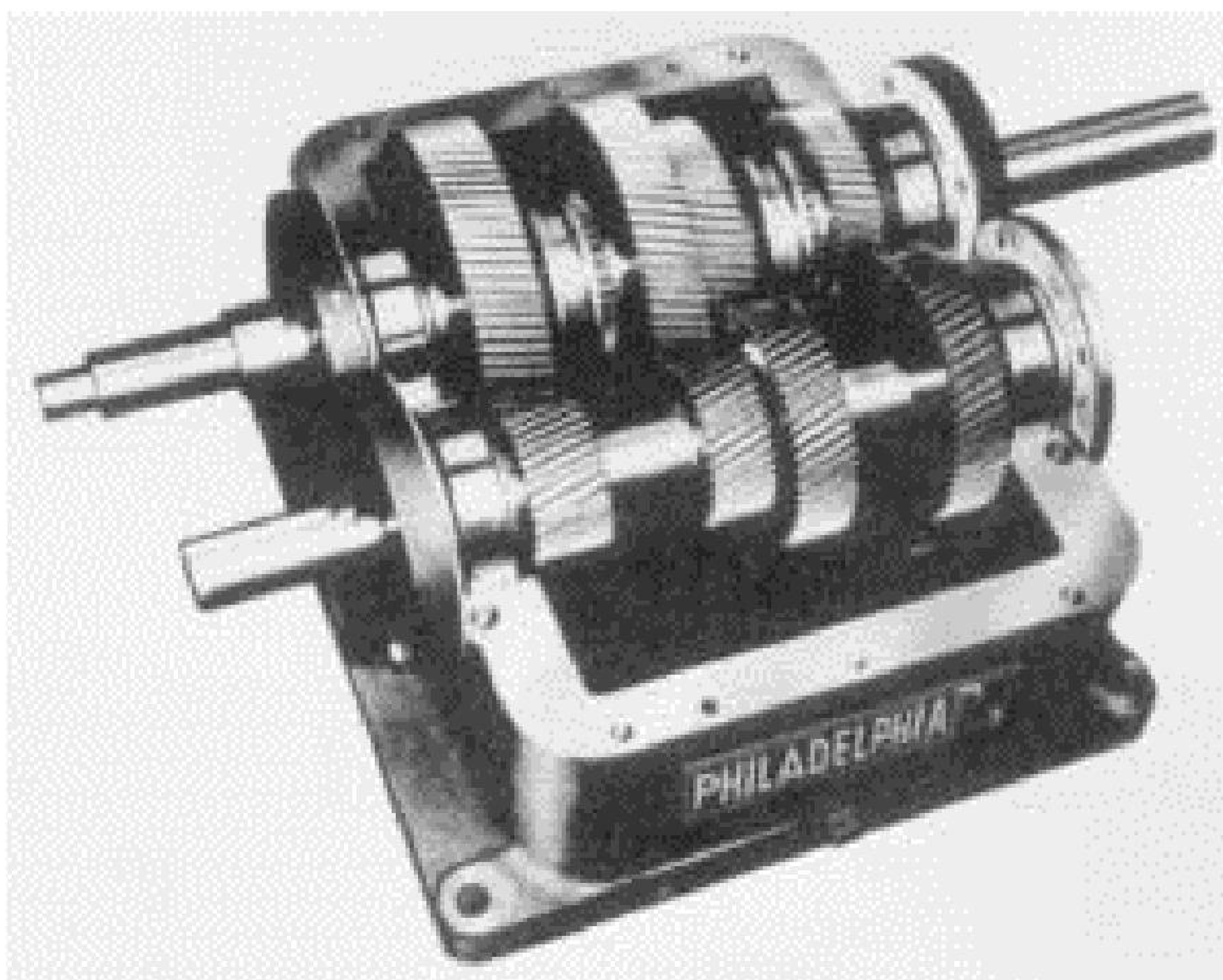

FIGURE 9.11 Four-speed gearbox.

The basic elements of an epicyclic drive are a central sunwheel, an internally toothed annulus ring, a planet or star carrier, and planet or star wheels. Depending on which of the first three elements is fixed, three types of epicyclic drives are possible: a planetary gear drive, a star gear drive, or a solar gear drive.

In a planetary gear drive (Fig. 9.2*a*), the annulus ring is fixed. In this design, the input and output shafts rotate in the same direction, and ratios are normally in the 3 to 1 to 12 to 1 range. The planetary gear drive is usually the most compact and cost-efficient for a given torque capacity and is therefore the most common. One limitation is that the speed range of the carrier is limited by the centrifugal loading of the planet bearings.

To overcome this problem of centrifugal loading in high-speed applications, a star gear drive (Fig. 9.2*b*) may be use din which the carrier assembly is fixed. In this design, input and output shafts rotate in opposite directions, and ratios in the range of 2 to 1 to 11 to 1 are possible.

The third epicyclic configuration is the solar gear drive (Fig. 9.2*c*), in which the sunwheel is fixed. With this design, input and output shafts rotate in the same direction, and ratios are on the order of 1.1 to 1 to 1.7 to 1. Because of the limited ratio range, solar gear drives are found only in very special applications.

Select the Proper Drive

Satisfactory performance of a gear drive depends on proper design and manufacture of the drive itself, selection of the proper type and size of unit for a given application, proper installation, proper use of the unit in service, and proper maintenance of the unit throughout its entire service life.

The official AGMA identification plate (Fig. 9.12) signifies that the manufacturer is a member of AGMA. Do not hesitate to request the assistance of trained engineers from a reputable manufacturer in selecting and properly installing gear-drive units.

In selecting the proper gear drive for any application, you will need to determine the horsepower requirement of the unit and the speed ratio between input and output shafts. You will already know (1) the horsepower and speed required to drive your machinery and (2) the output speed and horsepower of the prime mover.

To calculate horsepower required in any gear drive, first adjust for efficiency losses in the unit. Generally speaking, the efficiency of spur, helical, double-helical, straight bevel, spiral bevel, zerol, and hypoid gears is taken as 98 percent per gearset. The efficiency of worm gears in reducers can vary widely (10 to 95 percent) depending on speed, ratio, materials, etc. Consult the gear manufacturer for efficiency in specific applications. Where multiple gearsets are used, the overall efficiency is the product of the efficiencies of the gearsets. Most gear-drive manufacturers publish efficiency ratings on their individual units.

Having adjusted required horsepower upward to compensate for efficiency losses, now adjust for the type of service the gear drive must handle. For standardization and convenience, certain commonly used driven machines have been classified according to the types of service they normally require. Table 9.1 shows this classification, where U = uniform shock load, M = moderate shock load, and H = heavy shock load.

Determine the load characteristics of your application from this table. Then multiply the horsepower (adjusted for efficiency losses) by the appropriate service factor (Tables 9.2 to 9.4 for that type of unit, prime mover, load, and usage.

Next, to calculate the speed ratio of any gear drive, simply divide the rpm of the input shaft by the rpm required on the output shaft. Then, select a gear drive that will meet the speed and increased horsepower requirements.

After a gear-drive unit has been selected for mechanical rating, check the actual horsepower to be transmitted against the manufacturer's thermal rating for that unit. Thermal rating is the maximum

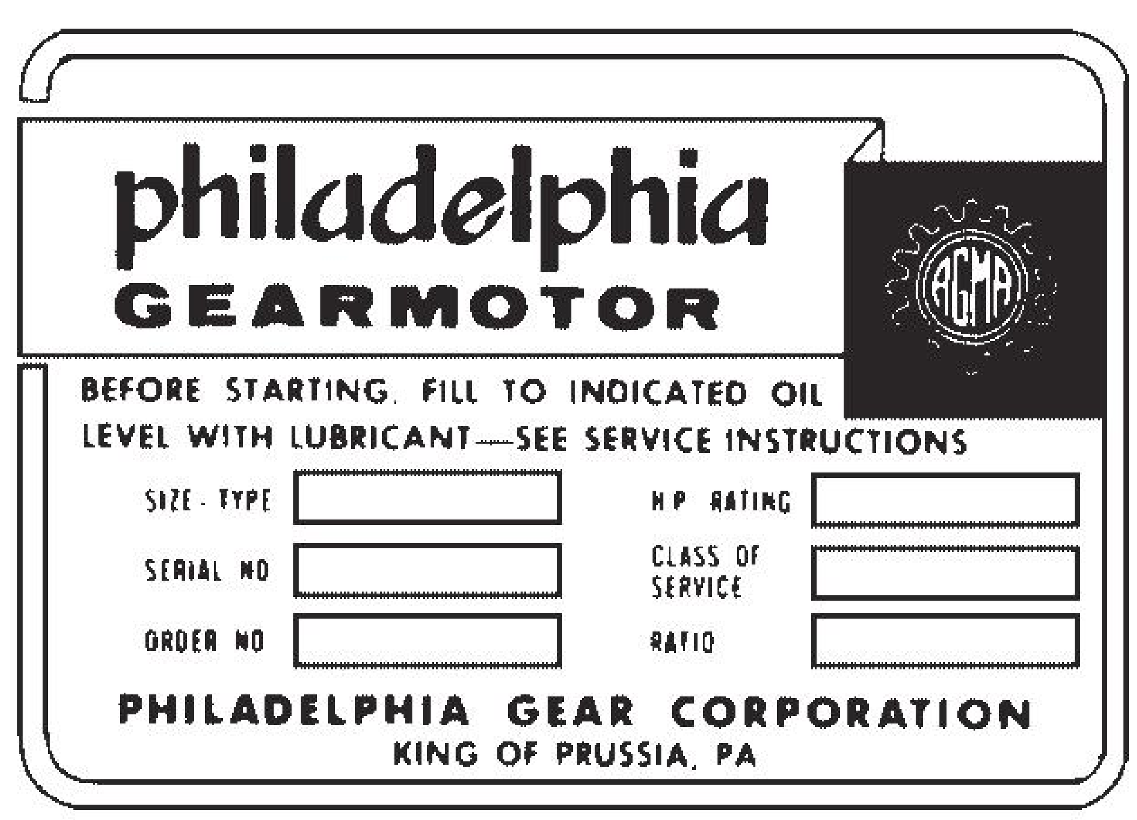

FIGURE 9.12 AGMA identification plate.

TABLE 9.1 Load Characteristics

Application	Uniform load	Moderate shock	Heavy shock
Agitators:			
Pure liquids	U		
Liquids and solids		M	
Liquids—variable density		M	
Blowers:			
Centrifugal	U		
Lobe		M	
Vane	U		
Brewing and distilling:			
Bottling machinery	U		
Brew kettles, cont. duty	U		
Cookers—cont. duty	U		
Mash tubs—cont. duty	U		
Scale hopper, frequent starts		M	
Can filling machines	U		
Cane knives		M	
Car dumpers			H
Car pullers		M	
Clarifiers	U		
Classifiers		M	
Clay-working machinery:			
Brick press			H
Briquette machine			H
Clay-working machinery		M	
Pug mill		M	
Compressors:			
Centrifugal	U		
Lobe		M	
Reciprocating, multicylinder		M	
Reciprocating, single-cylinder			H
Conveyors—uniformly loaded or fed:			
Apron	U		
Assembly	U		
Belt	U		
Bucket	U		
Chain	U		
Flight	U		
Oven	U		
Screw	U		
Conveyors—heavy duty, not uniformly fed:			
Apron		M	
Assembly		M	
Belt		M	
Bucket		M	
Chain		M	
Flight		M	
Live roll*			
Oven		M	
Reciprocating			H
Screw		M	
Shaker			H
Cranes:			
Main hoists	U		
Bridge travel*			
Trolley travel*			
Crusher:			
Ore			H
Stone			H
Sugar†		M	
Dredges:			
Cable reels		M	
Conveyors		M	
Cutter head drives			H
Jig drives			H
Maneuvering winches		M	
Pumps		M	
Screen drive			H
Stackers		M	
Utility winches		M	
Dry-dock cranes:			
Elevators:			
Bucket—uniform load	U		
Bucket—heavy load		M	
Bucket—cont.	U		
Centrifugal discharge	U		
Escalators	U		
Freight		M	
Gravity discharge	U		
Man lifts*			
Passenger*			
Fans:			
Centrifugal	U		
Cooling towers			
Induced draft*			
Forced draft	U		
Induced draft		M	
Large (mine, etc.)		M	
Large (industrial)		M	
Light(small diameter)	U		
Feeders:			
Apron		M	
Belt		M	
Disk	U		
Reciprocating			H
Screw		M	
Food industry:			
Beet slicer		M	
Cereal cooker	U		
Dough mixer		M	
Meat grinders		M	
Generators (not welding)	U		
Hammer mills			H

TABLE 9.1 Load Characteristics (*Continued*)

Application	Uniform load	Moderate shock	Heavy shock
Hoists:			
Heavy duty			H
Medium duty		M	
Skip hoist		M	
Laundry washers:			
Reversing		M	
Laundry tumblers		M	
Line shafts:			
Driving processing equipment		M	
Light	U		
Other line shafts	U		
Lumber industry:			
Barkers		M	
Spindle feed		M	
Main drive			H
Carriage drive*			
Conveyors:			
Burner		M	
Main or heavy duty			
Main log			H
Resaw			
Merry-go-round		M	
Slab			H
Transfer		M	
Chains:			
Floor			
Green		M	
Cut-off saws:			
Chain		M	
Drag		M	
Debarking drums			H
Feeds:			
Edger			
Gang			H
Trimmer	U	M	
Log deck			H
Log hauls—incline—well type			H
Log turning devices			H
Planer feed		M	
Planer tilting hoists			
Rolls—live-off brg.—roll cases			H
Sorting table		M	
Tipple hoist		M	
Transfers:			
Chain		M	
Craineway		M	
Tray drives		M	
Veneer lathe drives*			
Machine tools:			
Bending roll		M	
Punch press—gear driven			H
Notching press—belt driven*			
Plate planers			H
Tapping machine			H
Other machine tools:			
Main drives		M	
Auxiliary drives	U		
Metal mills:			
Draw bench carriage and main drive		M	
Pinch, drier and scrubber rolls, reversing*			
Slitters		M	
Table conveyors			
Nonreversing			
Group drives		M	
Individual drives			H
Reversing*			
Wire drawing and flattening machine		M	
Wire winding machine		M	
Mills, rotary type:			
Ball†			H
Cement kilns†		M	
Dryers and coolers†		M	
Kilns		M	
Pebble†		M	
Rod, plain and wedge bar†		M	
Tumbling barrels			H
Mixers:			
Concrete mixers, continuous		M	
Concrete mixers, intermittent		M	
Constant density	U		
Variable density		M	
Oil industry:			
Chillers		M	
Oil-well pumping*			
Paraffin filter press		M	
Rotary kilns		M	
Paper mills:			
Agitators (mixers)		M	
Barker—mechanical		M	
Barking drum			H
Beater and pulper		M	
Calendars		M	
Calendars—super			H
Converting machine, except cutters, platers		M	
Conveyors	U		
Couch	U		
Cutters—platers	U		H
Cylinders	U		
Dryers			
Jordans		M	
Presses	U		

TABLE 9.1 Load Characteristics (*Continued*)

Application	Uniform load	Moderate shock	Heavy shock
Paper mills (*Cont.*):			
Pulp machine reel	U		
Washers and thickeners		M	
Winders	U		
Printing presses:*			
Pullers:			
Barge haul			H
Pumps:			
Centrifugal	U		
Proportioning		M	
Reciprocating			
Single acting, 3 or more cylinders		M	
Double acting, 2 or more cylinders		M	
Single acting, 1 or 2 cylinders*			
Double acting, single cylinder*			
Rotary—gear type	U		
Lobe vane	U		
Rubber industry:			
Intensive internal mixers			
Batch mixers			
Continuous mixers		M	
Mixing mill—two smooth rolls		M	
Batch drop mill—two smooth rolls		M	
Cracker warmer—two roll		M	
One corrugated roll			H
Two corrugated rolls		M	
Holding feed and bland mill—two roll		M	
Refiner—two roll		M	
Calendars		M	
Extruders			
Continuous screw operation		M	
Intermittent screw operation		M	
Sand muller		M	
Sewage-disposal equipment:			
Bar screens	U		
Chemical feeders	U		
Collectors, circuline or straight-line	U		
Dewatering screws		M	
Scum breakers		M	
Slow or rapid mixers		M	
Thickeners		M	
Vacuum filters		M	
Screens:			
Air washing	U		
Rotary-stone or gravel		M	
Traveling water intake	U		
Slab pushers		M	
Steering gear*			
Stokers	U		
Sugar industry:			
Cane knives†		M	
Crushers†		M	
Mills†			H
Textile industry:			
Batchers		M	
Calendars		M	
Cards		M	
Dry cans		M	
Driers		M	
Dyeing machinery		M	
Knitting machines*			
Looms		M	
Mangles		M	
Nappers		M	
Pads		M	
Range drives*			
Slashers		M	
Soapers		M	
Spinners		M	
Tenter frames		M	
Washers		M	
Winders		M	
Windlass*		M	

*Refer to gear manufacturer.
†To be selected on basis of 24-hour service only.

average horsepower that can be transmitted continuously without creating a dangerous rise in temperature and without necessitating auxiliary cooling of the unit.

Install Gear Drives Carefully

When installing gear-drive units, be sure that they are well supported, accurately aligned, and securely anchored to prevent misalignment of gears or shafts. Consult the installation and maintenance instructions furnished by the manufacturer.

TABLE 9.2 Service Factors for Double-Helical, Helical, and Spiral Bevel Gear Units

	Electric-motor or steam-turbine drive				Multicylinder internal-combustion engine				Single-cylinder internal-combustion engine			
Character of load on driven machine	8–10 hr per day	24 hr per day	Inter-mittent 3 hr per day	Occa-sional ½ hr per day	8–10 hr per day	24 hr per day	Inter-mittent 3 hr per day	Occa-sional ½ hr per day	8–10 hr per day	24 hr per day	Inter-mittent 3 hr per day	Occa sional ½ hr per day
Uniform	1.0	1.25	0.8	0.5	1.25	1.05	1.0	0.8	1.5	1.75	1.25	1.0
Moderate shock	1.25	1.5	1.0	0.8	1.5	1.75	1.25	1.0	1.75	2.0	1.5	1.25
Heavy shock	1.75	2.00	1.5	1.25	2.0	2.25	1.75	1.5	2.25	2.5	2.0	1.75

1. Ratings shown in horsepower tables are based on a service factor of 1. For service factors other than 1 it is necessary to multiply the actual running horsepower required under normal full load by the service factor. The product of these two, which may be called the *equivalent horsepower,* is to be used when making reducer selection from horsepower table.

2. The horsepower tables permit a maximum momentary or starting load of 200 percent normal (100 percent overload). If peak load on driven machine exceeds twice the normal running horsepower, divide the peak horsepower by 2 and compare with the equivalent obtained by item 1. If larger, use it instead of item 1 in selecting reducer from tables.

3. Extreme repetitive shock and applications where exceedingly high energy loads must be absorbed, as when stalling, require special consideration and are therefore not covered by service factors given in the table.

4. In selecting a unit, the horsepower required should not exceed the thermal rating of the unit. Thermal rating indicates the amount of power that can be delivered through the unit without overheating.

TABLE 9.3 AGMA Standard Practice for Single and Double-Reduction Cylindrical-Worm and Helical-Worm Speed Reducers

		Driven machine load classifications		
Prime mover	Duration of service per day	Uniform	Moderate shock	Heavy shock
Electric motor	Occasional ½ hr	0.80	0.90	1.00
	Intermittent 2 hr	0.90	1.00	1.25
	10 hr	1.00	1.25	1.50
	24 hr	1.25	1.50	1.75
Multicylinder internal combustion engine	Occasional ½ hr	0.90	1.00	1.25
	Intermittent 2 hr	1.00	1.25	1.50
	10 hr	1.25	1.50	1.75
	24 hr	1.50	1.75	2.00
Following service factors apply for applications involving frequent starts and stops				
Single-cylinder internal combustion engine	Occasional ½ hr	1.00	1.25	1.50
	Intermittent 2 hr	1.25	1.50	1.75
	10 hr	1.50	1.75	2.00
	24 hr	1.75	2.00	2.25
Electric motor	Occasional ½ hr	0.90	1.00	1.25
	Intermittent 2 hr	1.00	1.25	1.50
	10 hr	1.25	1.50	1.75
	24 hr	1.50	1.75	2.00

Notes: 1. Time specified for intermittent and occasional service refers to total operating time per day.
2. Term *frequent starts and stops* refers to more than 10 starts per hour.

TABLE 9.4 Service Factors for Gearmotors

The three classes of gearmotors and shaft-mounted reducers as defined by the American Gear Manufacturers Association are as follows:

Class I:	For steady loads not exceeding normal rating of motor and 8 hr a day service. Moderate shock loads where service is intermittent. Service factor equals 1.
Class II:	For steady loads not exceeding normal rating of motor and 24 hours a day. Moderate shock loads for 8 hours a day. Service factor equals 1.4.
Class III:	For moderate shock loads for 24 hours a day. Heavy shock loads for 8 hours a day. Service factor equals 2

Good-quality mechanical couplings suitable for the application should be used to couple the shafts of driving and driven units.

Slight angular or linear shaft misalignments may be accommodated by using flexible-type mechanical couplings. In some cases, torsional stresses at starting or during momentary overloads can be compensated for by use of certain types of flexible mechanical couplings.

Proper loading of gear-drive units is essential for a long and trouble-free service life. Assuming that gear drives are properly rated for the particular applications and are properly installed, it is important that they should not be subjected to extreme or sustained overloads.

Torque limit switches are available as optional equipment on the gear drives of some manufacturers. Where the possibility of overloading or machinery jamming (which might produce an overload on the drive unit) is present, it is wise to insist on torque-limiting devices.

If you have any questions about the load capacities of gear drives on your machinery, do not hesitate to consult the manufacturer of the drive units.

Gear-drive housings are usually designed for proper heat dissipation under normal operating conditions. Do not allow units to operate where oil temperatures exceed those recommended by the manufacturer. Where surrounding atmospheric conditions might reduce normal heat dissipation, consult the drive manufacturer for his recommendations.

Start-Up

Some manufacturers ship new gear-drive units with internal parts protected by a polar-type rust-preventive film. There is no necessity to flush out this film, since it is usually soluble in the lubricant. (Consult the supplier of your particular gear-drive units for confirmation of this fact.) Merely fill the case with the recommended lubricant to the proper oil level. Always check to see if gear-drive units are shipped with or without oil from the factory. Units having bearings requiring grease must be checked and greased as required.

When units furnished with forced-feed lubrication are first put into service, they should be checked to observe that oil is being pumped. When a pressure gauge is furnished with the unit, gauge pressure should be as specified by the manufacturer, or if not specified, the pressure should be approximately 15 to 30 psi with the sump oil temperature at approximately 160°F. Adjust the relief valve if necessary to obtain the pressure specified in manufacturer's service manual.

Each unit is usually given a short run-in at the factory as part of the inspection procedure. However, for complete run-in under operating conditions, it is recommended that the unit be operated at partial load for 1 or 2 days to allow final wearing in of the gears. After this period, the load should be gradually increased to rated value.

After the unit has been operated under rated load for 2 weeks, it should be shut down in order to drain the oil and flush the housing. If desired, the original oil may be filtered, tested, and replaced. Filters finer than 25 microinches may filter out the additives. After the original oil has been drained, fill the case to the indicated level with SAE 10 straight-run mineral flushing oil containing no additives. The unit should be started, brought up to speed, and shut down immediately as a flushing procedure. Drain off flushing oil, and fill with recommended lubricant to the proper level.

After this initial oil change, an oil change is recommended after every 2500-hour or 6-month period of normal operation, whichever occurs first, unless there are unusually high temperature conditions combined with intermittent high loads where the temperature of the gear case rises rapidly and then cools off quickly. This condition may cause sweating on the inside walls of the unit, thus contaminating the oil and forming sludge. Under these conditions, or if the oil temperature is continuously above 150°F, or if the unit is subjected to an unusually moist atmosphere, oil changes may be necessary at 1- or 2-month intervals, as determined by field inspection of the oil. Synthetic oils, particularly hydrocarbons, may be used to improve oil life. Consult the manufacturer for recommended actions.

Lubrication

Lubricating oils for use with enclosed gears and gear units should be high-grade, high-quality, well-refined, straight mineral petroleum oils, within the recommended viscosity ranges as shown in Table 9.5. They must not be corrosive to gears or ball or roller bearings. They must be neutral in reaction. They should have good defoaming properties. No grit or abrasives should be present.

TABLE 9.5 AGMA Lubricant Number Recommendations for Enclosed Helical, Herringbone, Straight Bevel, Spiral Bevel, and Spur Gear Drives

	AGMA lubricant number[b,c,]	
	Ambient temperature[d,e]	
	−10 to +10°C	10 to 50°C
Type of unit[a] (low speed center distance)	(15 to 50°F)	(50 to 125°F)
Parallel shaft (single reduction)		
Up to 200 mm (to 8 in.)	2–3	3–4
Over 200 mm, to 500 mm (8 to 20 in.)	2–3	4–5
Over 500 mm (over 20 in.)	3–4	4–5
Parallel shaft (double reduction)		
Up to 200 mm (to 8 in.)	2–3	3–4
Over 200 mm (over 8 in.)	3–4	4–5
Parallel shaft (triple reduction)		
Up to 200 mm (to 8 in.)	2–3	3–4
Over 200 mm, to 500 mm (8 to to 20 in.)	3–4	4–5
Over 500 mm (over 20 in.)	4–5	5–6
Planetary gear units (housing diameter)		
Up to 400 mm (to 16 in.) O.D.	2–3	3–4
Over 400 mm (over 16 in.) O.D.	3–4	4–5
Straight or spiral bevel gear units		
Cone distance to 300 mm (to 12 in.)	2–3	4–5
Cone distance over 300 mm (over 12 in.)	3–4	5–6
Gearmotors and shaft-mounted units	2–3	4–5
High-speed units[f]	1	2

[a]Drives incorporating overrunning clutches as backstopping devices should be referred to the gear drive manufacturer as certain types of lubricants may adversely affect clutch performance.
[b]Ranges are provided to allow for variations in operating conditions such as surface finish, temperature rise, loading, speed, etc.
[c]AGMA viscosity number recommendations listed above refer to R&O gear oils. EP gear lubricants in the corresponding viscosity grades may be substituted where deemed necessary by the gear-drive manufacturer.
[d]For ambient temperatures outside the ranges shown, consult the gear manufacturer. Some synthetic oils have been used successfully for high- or low-temperature applications.
[e]Pour point of lubricant selected should be at least 5°C (9°F) lower than the expected minimum ambient starting temperature. If the ambient starting temperature approaches lubricant pour point, oil sump heaters may be required to facilitate starting and ensure proper lubrication.
[f]High-speed units are those operating at speeds above 3600 rpm or pitch line velocities above 25 m/s (5000 fpm) or both. Refer to Standard AGMA 421, "Practice for High Speed Helical and Herringbone Gear Units," for detailed lubrication recommendations.

For high operating temperatures, good resistance to oxidation is needed. For low temperatures, an oil having a low pour point to meet the lowest temperature expected is needed. When the operating temperature varies over a wide range, an oil having a high viscosity index is desirable.

When the gears are subject to heavy shock or impact loading or when the unit is subject to extremely heavy duty, an extreme-pressure (EP) lubricant should be used.

The EP lubricant must meet the general specifications listed above for straight mineral oil. For severe conditions, synthetic lubricants offer higher viscosity index and extended temperature operating range. All lubricants should meet the recommendations of the gear manufacturer.

The viscosity of the EP lubricant should be approximately the same as that of the recommended AGMA lubricants given in Tables 9.5 and 9.6.

On many types of gear-drive units, pressure fittings are supplied for the application of grease to bearings that are shielded from the oil.

Sufficient grease to form a film over the rollers and races of the bearing is all that is actually required for lubrication of roller bearings; however, ample reservoir space for grease is usually provided.

Gear-drive units are usually shipped from the factory with grease applied. Bearings and seals should be lubricated at definite intervals. Study will be required to determine how frequently this should be done for a particular operation to ensure a proper supply of lubricant to the bearing and seal areas.

Many greases are suitable as lubricants for bearings and seals in gear-drive units. The grease should be high-quality, nonseparating, ball-bearing grade suitable for the operating temperature.

The lubricant should not be corrosive to gears or to ball or roller bearings; must be neutral in reaction; should have no grit, abrasive, or fillers present; should not precipitate sediment; should not separate at temperatures up to 300°F; and should have moisture-resisting characteristics. The lubricant must have good resistance to oxidation.

Every precaution should be taken to prevent any foreign matter from entering the gear case. Sludge is caused by dust, dirt, moisture, and chemical fumes. These are the biggest enemies of proper and adequate lubrication in gear-drive units.

Good Maintenance Practice

During normal periods of operation, gear-drive units should be given daily routine inspection, consisting of visual inspection and observation for oil leaks or unusual noises. If oil leaks are evident, the unit should be shut down, the cause of the leakage corrected, and the oil level checked. If any unusual noises occur, the unit should be shut down until the cause of the noise has been determined and corrected. Check all oil levels at least once a week. The operating temperature of the gear-drive unit is the temperature of the oil inside the case. Under normal conditions, the maximum operating temperature should not exceed 180°F. Generally, pressure-lubricated units are equipped with a filter which should be cleaned periodically.

Shutdown

If it becomes necessary to shut down the unit for a period longer than 1 week, the unit should be run at least 10 min each week while it is idle. This short operation will keep the gears and bearings coated with oil and help prevent rusting due to condensation of moisture resulting from temperature changes.

Troubleshooting Gears

Someone has observed that "gears wear out until they wear in…and then they never wear out." The AGMA describes this phenomenon more precisely as follows:

> It is the usual experience with a set of gears in a gear unit…assuming proper design, manufacture, application, installation, and operation…that there will be an initial "running-in" period during which, if

TABLE 9.6 AGMA Lubricant Number Recommendations for Enclosed Cylindrical and Double-Enveloping Worm Gear Drives

Type, worm gear drive	Worm speed‡ up to, rpm	AGMA lubricant numbers* Ambient temperature†		Worm speed‡ above, rpm	AGMA lubricant numbers* Ambient temperature†	
		−10 to +10°C (15 to 50°F)	10 to 50°C (50 to 125°F)		−10 to +10°C (15 to 50°F)	10 to 50°C (50 to 125°F)
Cylindrical worm§						
Up to 150 mm (to 6 in.)	700	7 Comp, 7 EP	8 Comp, 8 EP	700	7 Comp, 7 EP	8 Comp, 8 EP
Over 150 mm, to 300 mm (6 to 12 in.)	450	7 Comp, 7 EP	8 Comp, 8 EP	450	7 Comp, 7 EP	7 Comp, 7 EP
Over 300 mm, to 450 mm (12 to 18 in.)	300	7 Comp, 7 EP	8 Comp, 8 EP	300	7 Comp, 7 EP	7 Comp, 7 EP
Over 450 mm, to 600 mm (18 to 24 in,)	250	7 Comp, 7 EP	8 Comp, 8 EP	250	7 Comp, 7 EP	7 Comp, 7 EP
Over 600 mm (over 24 in.)	200	7 Comp, 7 EP	8 Comp, 8 EP	200	7 Comp, 7 EP	7 Comp, 7 EP
Double-enveloping worm§						
Up to 150 mm (to 6 in.)	700	8 Comp	8A Comp	700	8 Comp	8 Comp
Over 150 mm, to 300 mm (6 to 12 in.)	450	8 Comp	8A Comp	450	8 Comp	8 Comp
Over 300 mm, to 450 mm (12 to 18 in.)	300	8 Comp	8A Comp	300	8 Comp	8 Comp
Over 450 mm, to 600 mm (18 to 24 in.)	250	8 Comp	8A Comp	250	8 Comp	8 Comp
Over 600 mm (over 24 in.)	200	8 Comp	8A Comp	200	8 Comp	8 Comp

*Both EP and compounded oils are considered suitable for cylindrical worm gear service. Equivalent grades of both are listed in the table. For double-enveloping worm gearing, EP oils in the corresponding viscosity grades may be substituted only where deemed necessary by the worm gear manufacturer.

†Pour point of the oil used should be less than the minimum ambient temperature expected. Consult gear manufacturer on lube recommendations for ambient temperatures below −10°C (14°F).

‡Worm gears of either type operating at speeds above 2400 rpm or 10 m/s (200 fpm) rubbing speed may require force-feed lubrication. In general, a lubricant of lower viscosity than recommended in the above table shall be used with a force feed system.

§Worm gear drives also may operate satisfactorily using other types of oils. Such oils should be used, however, only upon approval by the manufacturer.

the gears are properly lubricated and not overloaded, the combined action of rolling and sliding of the teeth may smooth out the manufactured surface and give the working surface a high polish. Under continued proper conditions of operation, the gear teeth will then show little or no sign of wear.

Despite the truth of this statement, failure of metallic gear teeth may occur as a result of excessive deterioration of the working surfaces of the teeth or as actual tooth breakage. In many such situations, early recognition of possible trouble may suggest a remedy before extensive damage occurs.

GEAR-TOOTH WEAR AND FAILURE

Experience indicates that the vast majority of gear-tooth wear and failure types may be summed up under nine basic headings in two classifications:

Classification A: Surface Deterioration

1. Wear
2. Plastic flow
3. Scoring
4. Surface fatigue
5. Miscellaneous tooth-surface deteriorations

Classification B: Tooth Breakage

6. Fatigue
7. Heavy wear
8. Overload
9. Cracking

The following discussion (which conforms to AGMA Standard 110.04 nomenclature) may be used as a guide to identification of gear-tooth trouble. If discovered early enough, many gear-tooth failures can be avoided through proper corrective maintenance as indicated. (The illustrations were prepared by the AGMA, which has given permission for their use.)

Surface Deterioration

Wear. *Wear* is a general term describing loss of material from the contacting surfaces of gear teeth.
Normal wear is the slow loss of metal from the contacting surfaces at a rate that will not affect satisfactory performance within the expected life of the gears. (See Fig. 9.13.)

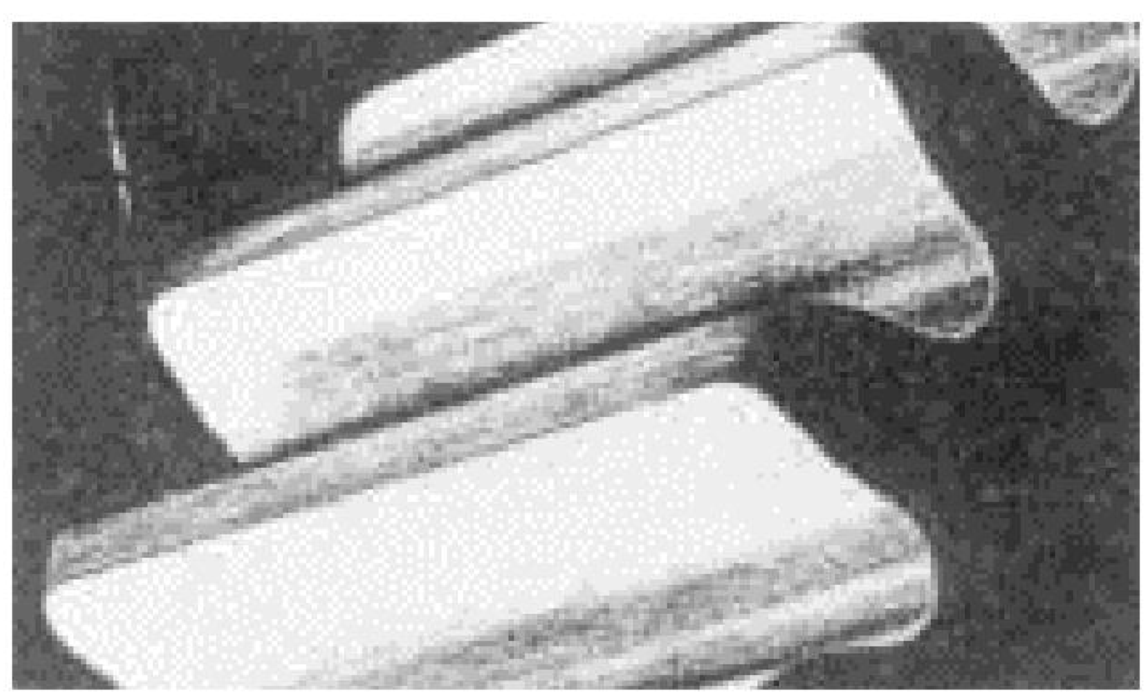

FIGURE 9.13 Normal wear.

Maintenance Procedure. A certain amount of smoothing and polishing is expected during "running-in" of new gearsets. This type of wear is less noticeable where gears have been shaved or finished-ground during manufacture. Before gears are run at all, they should be checked for proper installation and to ensure that loading is controlled within rating limits as set by the manufacturer. The use of recommended lubricant and filters should eliminate excessive gear-tooth wear during the running-in period. Most manufacturers of assembled gear drives recommend flushing the gear case frequently to remove any metallic particles and to eliminate the possibility of foreign objects circulating through the gear mesh.

Abrasive wear is surface injury caused by fine particles passing through the gear mesh. These particles may be dirt not completely removed, sand or scale from castings, impurities in the oil or from the surrounding atmosphere, or metal detached from the tooth surfaces or bearings. (See Fig. 9.14.)

Maintenance Procedure. Whenever abrasive wear is detected, the unit should be stopped immediately. Oil should be drained. The inside of the housing, gear teeth, and oil passages should be thoroughly scraped, flushed, and wiped down. A light flushing oil should be used for a short time and then drained before the oil reservoir is refilled with clean oil of proper grade. In contaminated atmospheres, special air breathers, oil seals, and filters should be considered as means of eliminating the infiltration of foreign particles into the gear case.

Scratching is a severe form of abrasive wear, characterized by short, scratchlike lines or marks on the contracting surfaces in the direction of sliding. It may be caused by burrs, projections on the tooth surface or material embedded in the tooth surface, or hard foreign pieces passing through the gear mesh. Scratching should not be confused with scoring because the two effects differ by definition, and scratching damage usually is light and does not result in progressive destruction, provided that the cause is removed. (See Fig. 9.15.)

Maintenance Procedure. Since scratching is an accentuated type of abrasive wear, with comparatively deep and widespread grooves up and down the tooth profile, the maintenance procedure is identical with that for simple abrasive wear. Make sure that the case, gears, and lubrication channels are completely free of foreign matter. Protect against recontamination by use of special filters, breathers, and oil seals where conditions indicate.

Overload wear is a form of wear experienced under conditions of heavy load and low speed, in both hardened and unhardened gears. Metal seems to be removed progressively in thin layers or flakes, leaving surfaces that appear somewhat as if etched. (See Fig. 9.16.)

Maintenance Procedure. The only permanent remedy for overload wear is to reduce the rate of wear. Care should be exercised in selecting extreme-pressure lubricants which are free from corrosive substances. Check the manufacturer for recommendations.

Plastic Flow. *Plastic flow* is the surface deterioration resulting from the yielding of the surface metal under heavy loads. It is usually associated with the softer metals but may occur in through-hardened and case-hardened steels.

Ridging is a particular form of plastic flow occurring on the tooth surfaces of case-hardened hypoid pinions and bronze worm gears. It usually appears as diagonal lines or ridges across the tooth surface

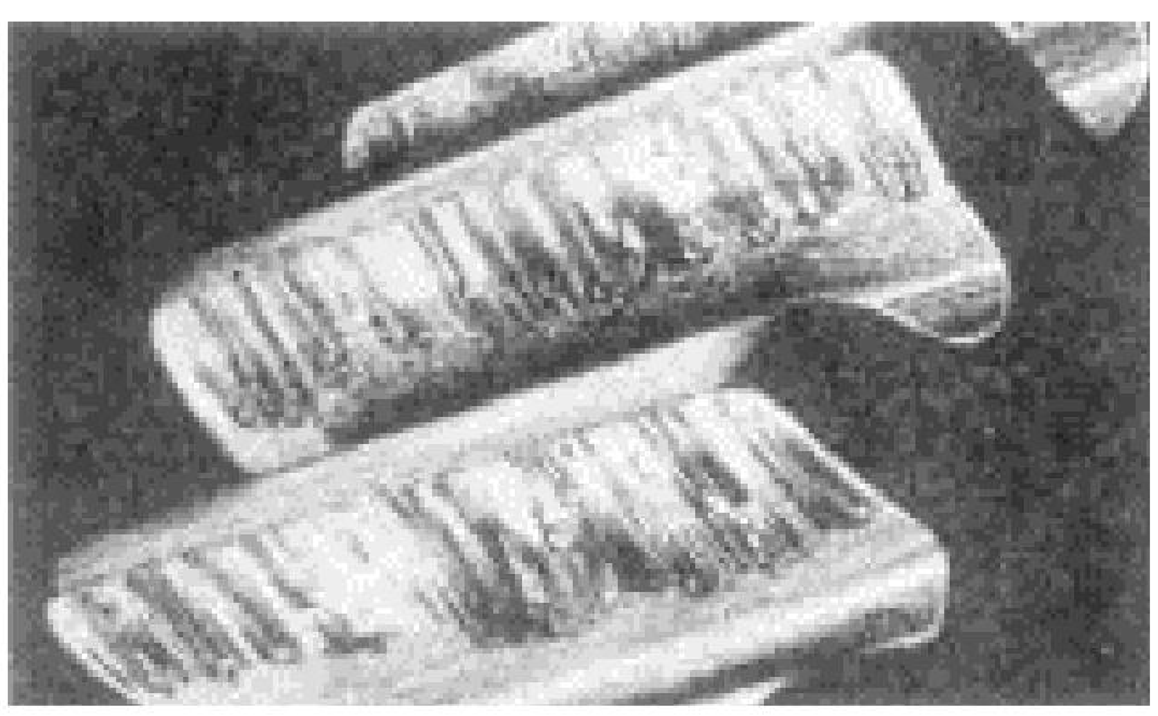

FIGURE 9.14 Abrasive wear.

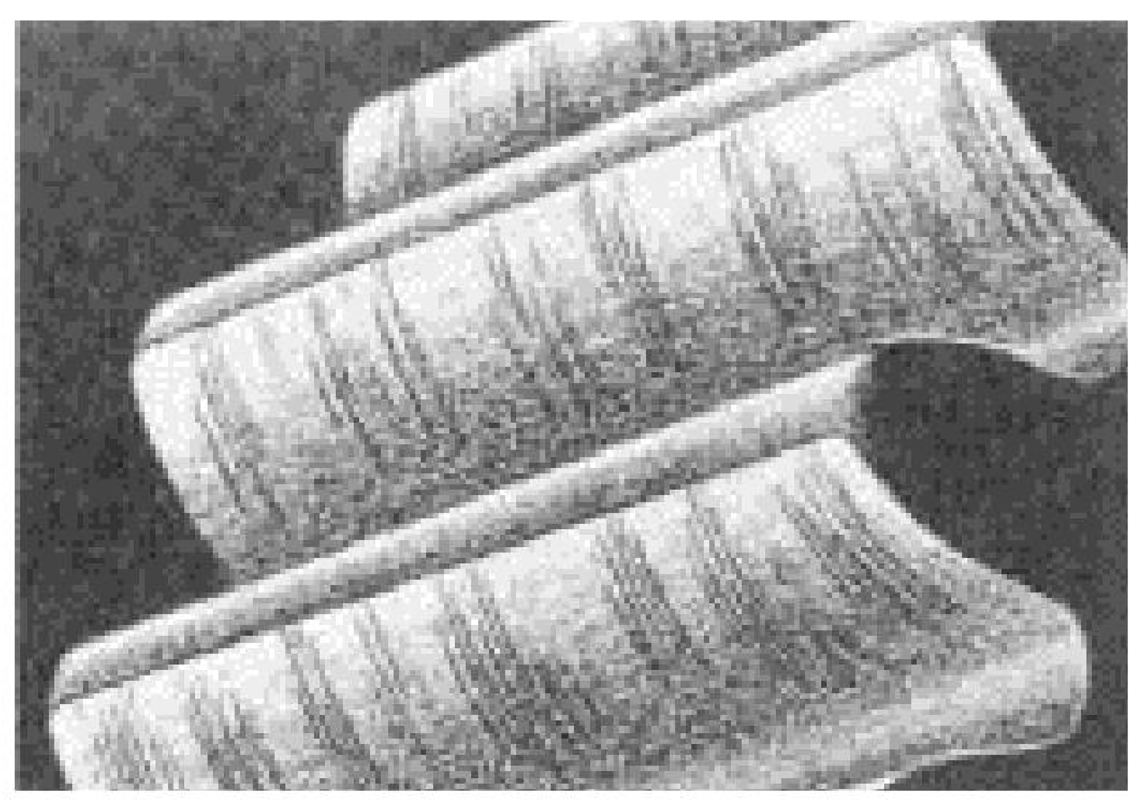

FIGURE 9.15 Scratching.

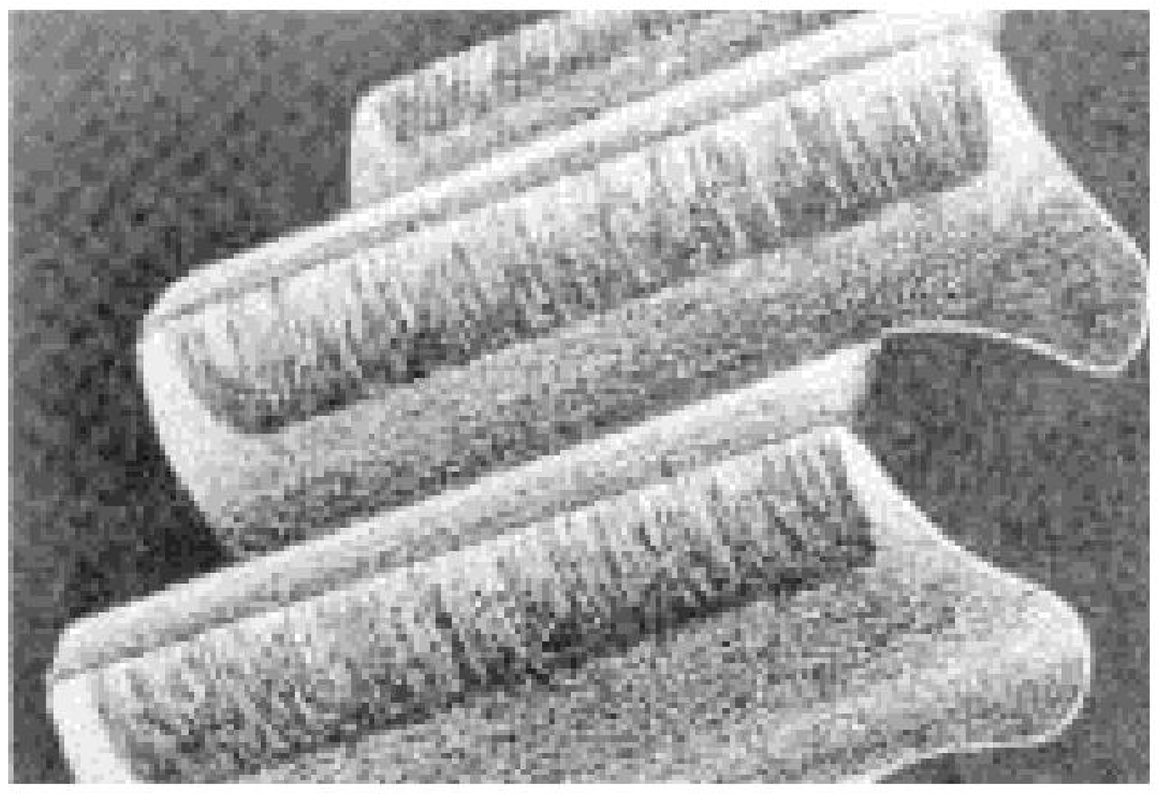

FIGURE 9.16 Overload wear.

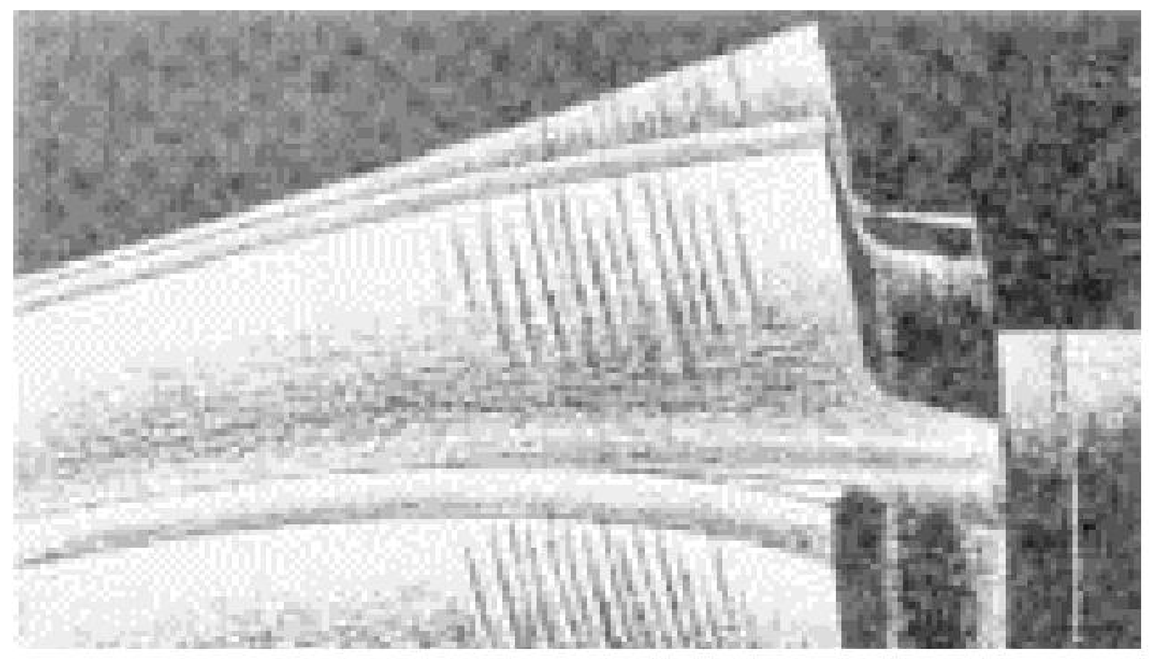

FIGURE 9.17 Ridging.

but may be characterized by a herringbone or fishtail pattern, both occurring in the direction of sliding. Ridging is generally associated with excessive loads or inadequate lubrication, and usually complete failure results unless the material has a great capacity for work hardening. (See Fig. 9.17.)

Maintenance Procedure. Since ridging usually results from localized loading, wherever possible, gears should be adjusted to distribute the load more evenly over the full tooth surface. In the case of bevel gears, backlash should be altered to reduce impact loading. In some cases, the use of extreme-pressure lubricants may help to reduce the rate of tooth-surface deterioration.

Rolling and peening almost always occur together as the result of the sliding action under excessive loads and the impact loading from improper tooth action. They are characterized by fins at the top edges or ends of the teeth (not to be confused with burrs from cutting or shaving), by badly rounded tooth tips, or by a depression in the surface of the driving gear at the start of single-tooth contact, with a raised ridge near the pitch line of the mating or driven gear. The remaining portions of the profiles are usually deformed to a considerable degree long prior to complete destruction. (See Fig. 9.18.)

Maintenance Procedure. Often, peening can be checked by reducing the backlash of gear teeth. Occasionally, the addition of a flywheel to the gear shaft will serve to smooth out the hammerblow effects of spur gear teeth entering and leaving the mesh. Since peening is "localized" surface deterioration, extreme-pressure lubricant sometimes is effective in reducing this destructive type of plastic flow.

Rippling is a wavelike formation on the surface at right angles to the direction of sliding. It is characterized by a fish-scale appearance, occurs mostly on case-hardened hypoid pinions, and does not constitute failure unless allowed to progress. It may be caused by surface yielding due to "slip-stick" friction resulting from inadequate lubrication, heavy loads, or vibration. (See Fig. 9.19.)

Scoring. The term *scoring* has been selected as preferable to such other terms as *scuffing, seizing,* and *galling*. It is the rapid removal of metal from the tooth surfaces caused by the tearing out of small contacting particles that have welded together as a result of metal-to-metal contact, and the scored surface is characterized by a torn or dragged and furrowed appearance with markings in the direction of sliding. Sometimes surface roughness or foreign matter passing through the mesh will cause

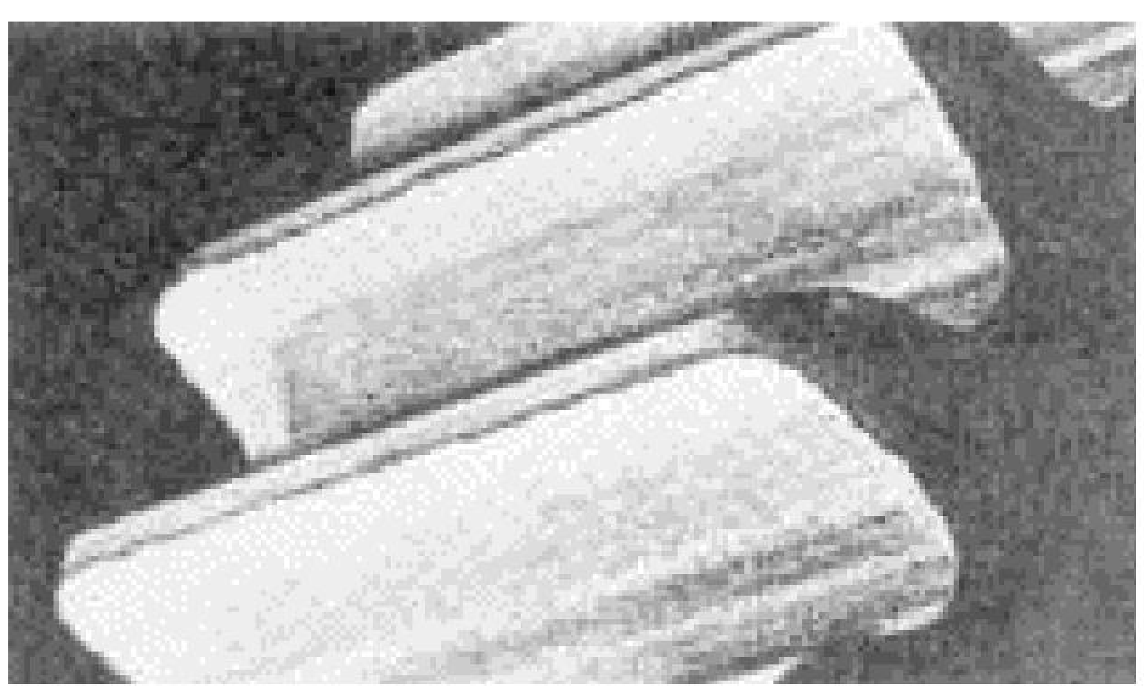

FIGURE 9.18 Rolling.

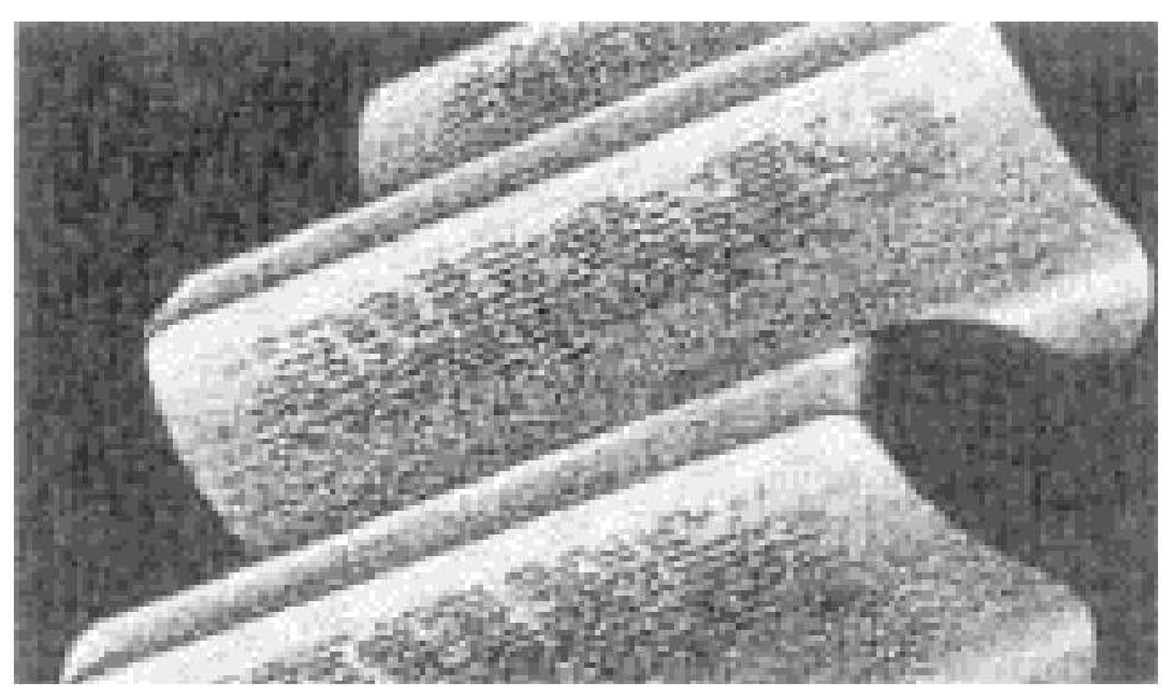

FIGURE 9.19 Rippling.

localized yielding on the mating profile, without tearing as such, with a similar furrowed appearance as a result of the "plowing" action. It may be localized initially and spread if the causative condition is not corrected. Sometimes, particularly in the case of misalignment, the damage may cease and the surface becomes smoother as the contact area spreads and more load-carrying face is brought into contact. Scoring is usually caused by rupture of the oil film resulting from load concentration at localized contact areas. Excessive unit loading or an unsuitable lubricant has the same effect. Sometimes scoring can be arrested by smoothing up the roughened area by filing or stoning or by use of a different type or grade of lubricant.

Slight scoring is a minor impairment of the gear-tooth surface of a welding nature, showing slight tears and scratches in the direction of sliding. Scoring usually starts at a surface area where there is a combination of high surface stress and sliding velocity-generally occurring at or near the tip of the tooth. (See Fig. 9.20.)

Severe scoring is a more advanced degree of welding, showing deep scratches and adhesions and leading to rapid gear-tooth-surface deterioration. (See Fig. 9.21.)

Maintenance Procedure. Correction of slight surface scoring often can be accomplished through the use of an extreme-pressure lubricant. Consult your lubricant supplier for specific recommendations. In some cases it may be necessary to polish tooth surfaces in addition to using an extreme-pressure lubricant. Where scoring persists, gear teeth may be metallurgically hardened (after polishing damaged areas) to resist further damage.

Surface Fatigue. Surface fatigue is the failure of the material as a result of repeated surface or subsurface stresses that are beyond the endurance limit of the material. It is characterized by the removal of metal and the formation of cavities. These cavities may be small and remain quite small; they may be small initially and then combine or increase in size by continued fatigue; or they may be of considerable size at the start.

Initial pitting is the type of surface fatigue which may occur at the beginning of operation and continue only until the overstressed local high areas of the surface have been reduced, thus obtaining sufficient area of contact to carry the load without further impairment. It usually occurs in a narrow band just below or at the pitch line. Such pitting is not serious, since it is corrective and nonprogressive. (See Fig. 9.22.)

Maintenance Procedure. Usually, initial pitting is observed as tiny cavities at scattered spots on the surfaces of the gear teeth. In most cases, running-in of gears will tend to polish down surface irregularities, and pitting will cease. Where pitting continues, metallurgical surface hardening of the gear teeth may be necessary. On occasion, grinding and/or polishing of tooth-bearing surfaces will help.

Destructive pitting is a form usually starting below the pitch line, progressively increasing size and number of pits until smoothness of operation is impaired. The remaining surface fails in a similar manner, and finally the tooth shape is destroyed. The pits constitute stress raises which may lead to failure by fatigue breakage. Large pits formed by the joining of smaller adjacent pits are due to failure of the material between them and constitute a form of spalling. (See Fig. 9.23.)

Maintenance Procedure. Destructive pitting may be checked by grinding and polishing gear-tooth surfaces. If polishing fails to retard destruction, metallurgical surface hardening often will eliminate further damage. In some cases, use of extreme-pressure lubricants has met with success.

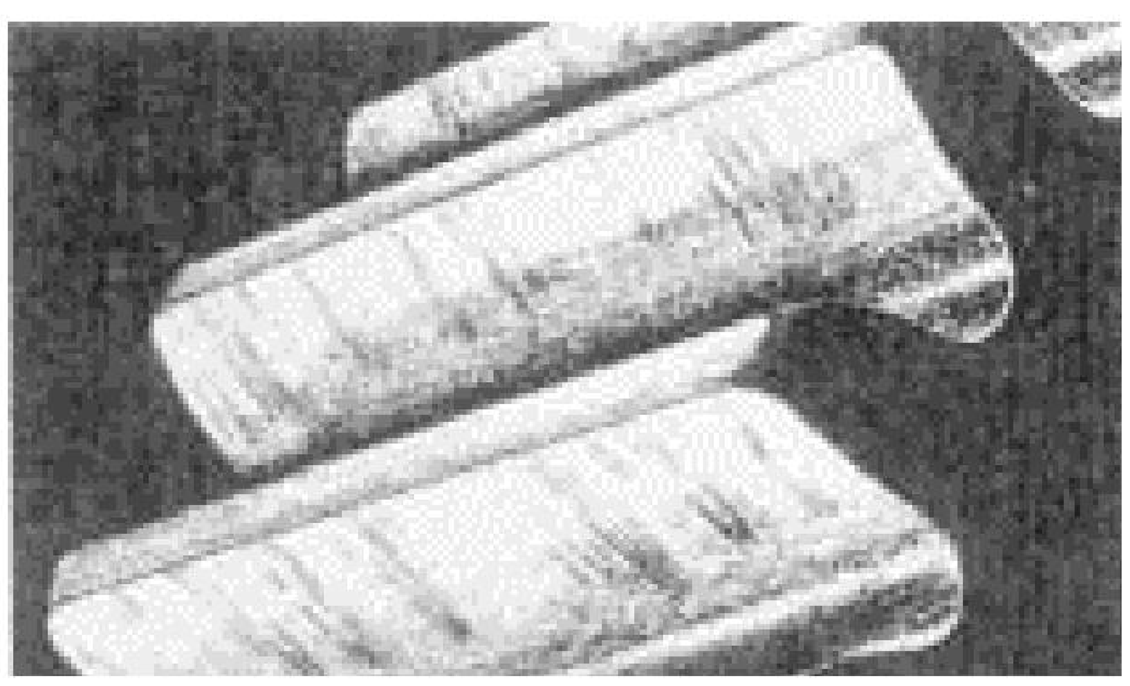

FIGURE 9.20 Slight scoring.

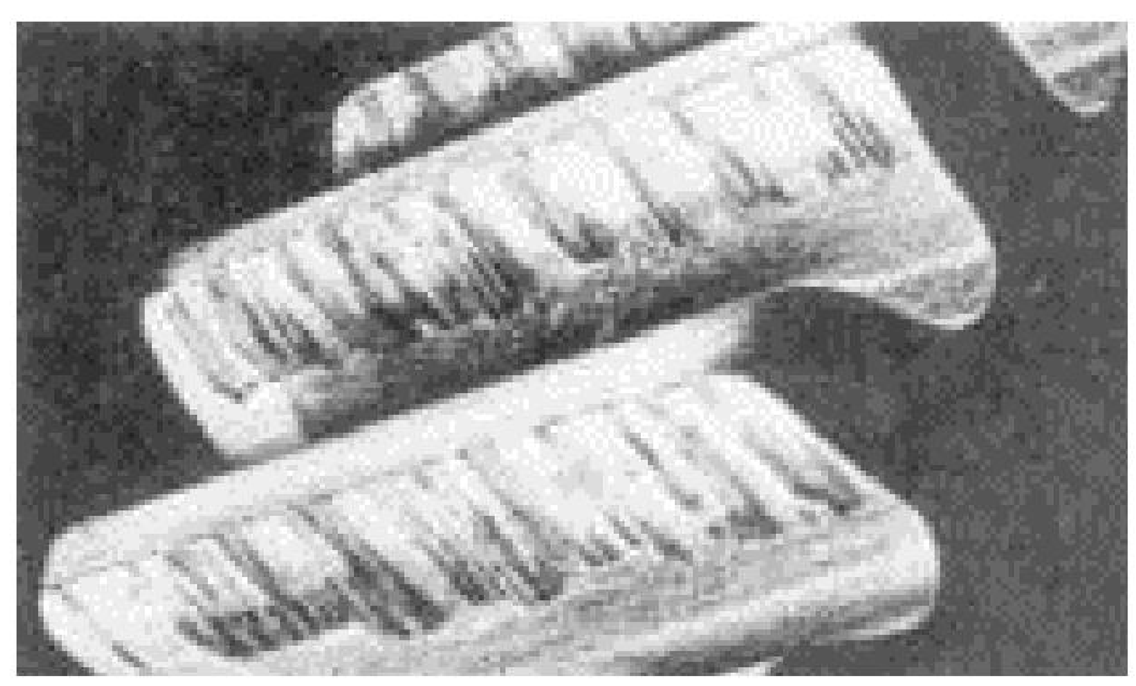

FIGURE 9.21 Severe scoring.

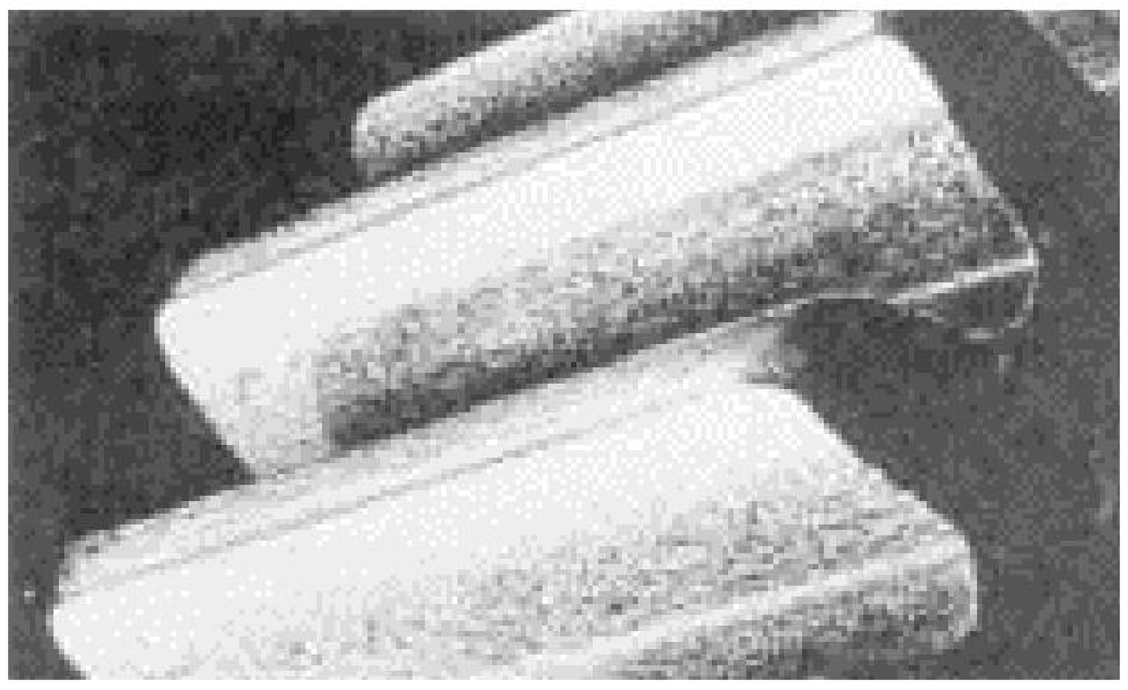

FIGURE 9.22 Initial pitting.

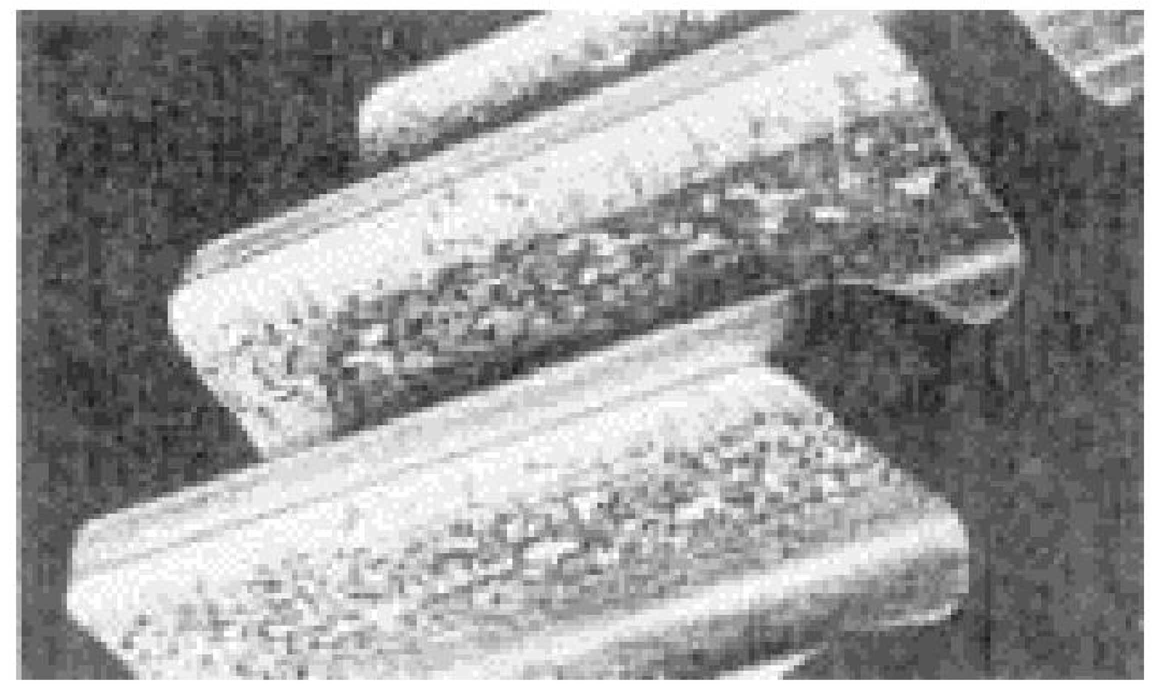

FIGURE 9.23 Destructive pitting.

Spalling of the more usual type is a sporadic fatigue failure, occurring only in fully hardened or, more usually, case-hardened steel, originating with a surface or subsurface defect or from excessive internal stresses due to heat treatment. It is characterized by large particles or chips which spall or flake out the tooth surfaces, usually along the top edges or ends. The cavities are larger, deeper, and of a cleaner break than pits, although the distinction is primarily one of degree. Frequently, it is not a fatigue failure of the usual variety, since it occurs after a relatively few cycles as a result of excessive internal stresses. The joining of several smaller pits by failure of the metal between them is a form of spalling. (See Fig. 9.24.)

Maintenance Procedure. If damage from spalling is not too extensive, use of an extreme-pressure lubricant may retard further damage. In some cases, surface polishing will provide more even distribution of the load across the gear-tooth surface and will relieve excessive pressure at the point where spalling has occurred. If tooth destruction continues, consult the gear manufacturer.

Miscellaneous Tooth-Surface Deterioration. Since corrosive wear, burning, interference, and grinding cracks in tooth-surface deterioration are independent sources of trouble, not closely related to one another or to the foregoing groups, they are treated independently.

Corrosive wear is surface deterioration from the chemical action of acid, moisture, or contamination of the lubricant. It may occur under several different circumstances. If the lubricant becomes contaminated with foreign acid, the teeth may become lightly pitted. The wiping action during contact may continually remove all evidence of this, but the rate of wear is excessive. Rusting as a result of contamination with water from condensation, excessive humidity, etc., will produce similar results. If corrosion or rusting is taking place, evidence also should appear on other surfaces besides the active tooth faces. In addition, corrosive wear can occur as a result of highly active EP ingredients in the lubricant. Under heavy loads, EP oils may react with the metal, permitting operation without scoring but with a uniform and low rate of wear under load conditions that could not otherwise be tolerated. Gear teeth that are wearing as a result of EP activity usually have a smooth appearance. If the oil temperature becomes excessive, more active reaction of the EP materials with the metal can take place, resulting in accelerated high-temperature corrosive wear. (See Fig. 9.25.)

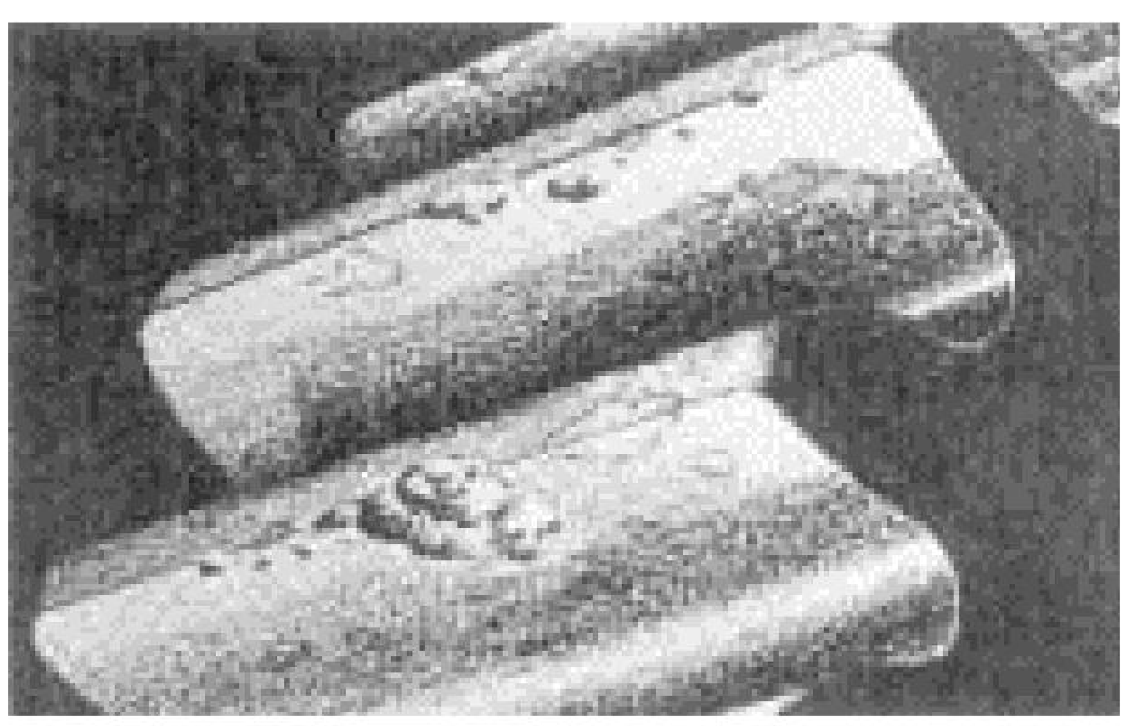

FIGURE 9.24 Spalling.

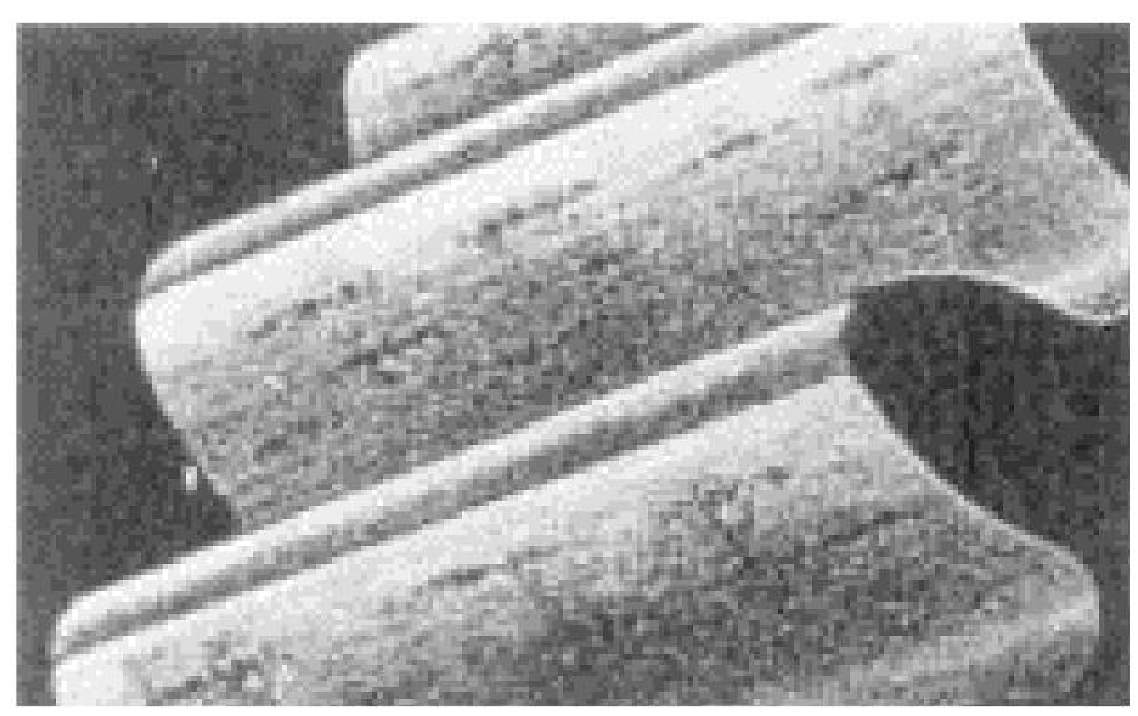

FIGURE 9.25 Corrosive wear.

Maintenance Procedure. Drain and flush gear case and gears to remove the source of existing contamination. Be sure that new lubricant is clean, of high quality, and uncontaminated. If corrosive wear persists, consult the manufacturer for recommendations as to special breathers and oil seals.

Burning can result in severe wear and surface deterioration of the previously described types owing to loss of hardness from high temperatures. The fatigue life also may be adversely affected—depending on the degree and location of the burn. It is characterized by temperature discoloration of the contacting and/or adjacent surfaces and is the result of excessive temperature, from either external sources or the excessive friction from overload, overspeed, or inadequate lubrication. On gears that have not been put into service, the same discoloration would indicate improper grinding, but generally, grinding burns can be detected only by etching. (See Fig. 9.26.)

Maintenance Procedure. To reduce friction, look first to the lubricant. In many cases, extreme-pressure lubricants will eliminate gear-tooth burning. Be sure gears are not being run in excess of their rated load and speed capacities. If burning persists, request the gear manufacturer to test for proper backlash and gear-tooth spacing.

Interference wear occurs when improper or premature contact concentrates the entire load at the point of engagement of the driving flank with the mating tip or at the disengagement of the driven flank and mating tip. It may range from a light line of wear or pitting of no serious consequence other than noisy operation to a more severe damage in which the flank is gouged out and the tip of the mate heavily rolled over, usually resulting in complete failure of the pair. (See Fig. 9.27.)

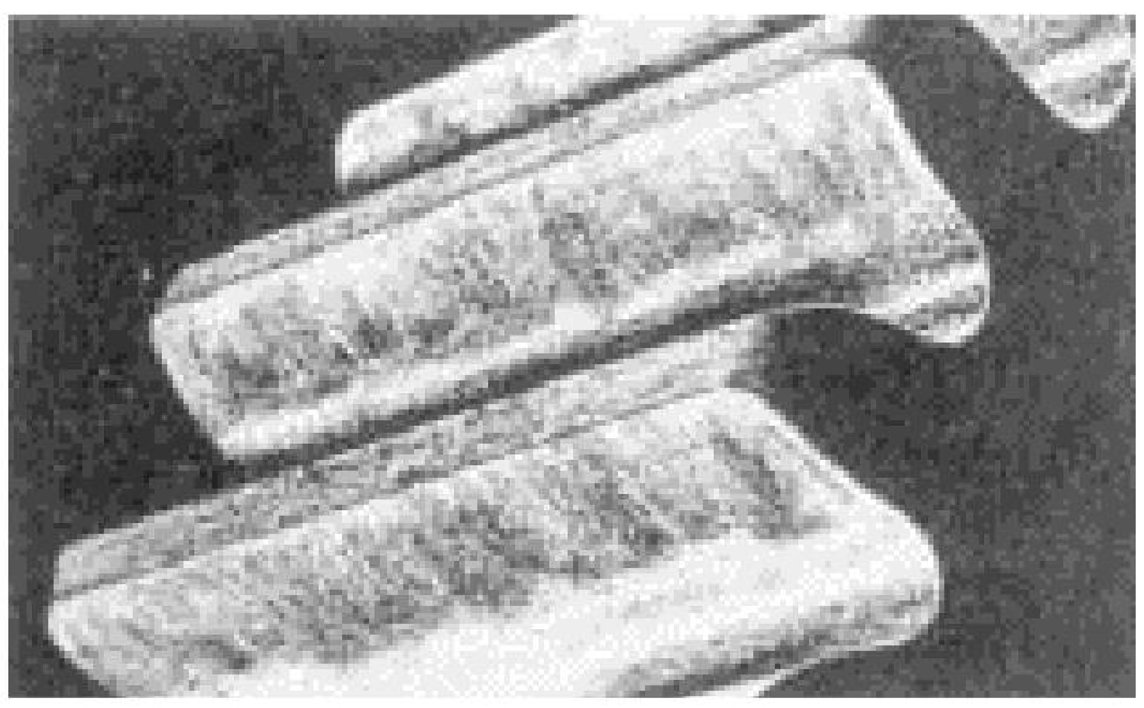

FIGURE 9.26 Burning.

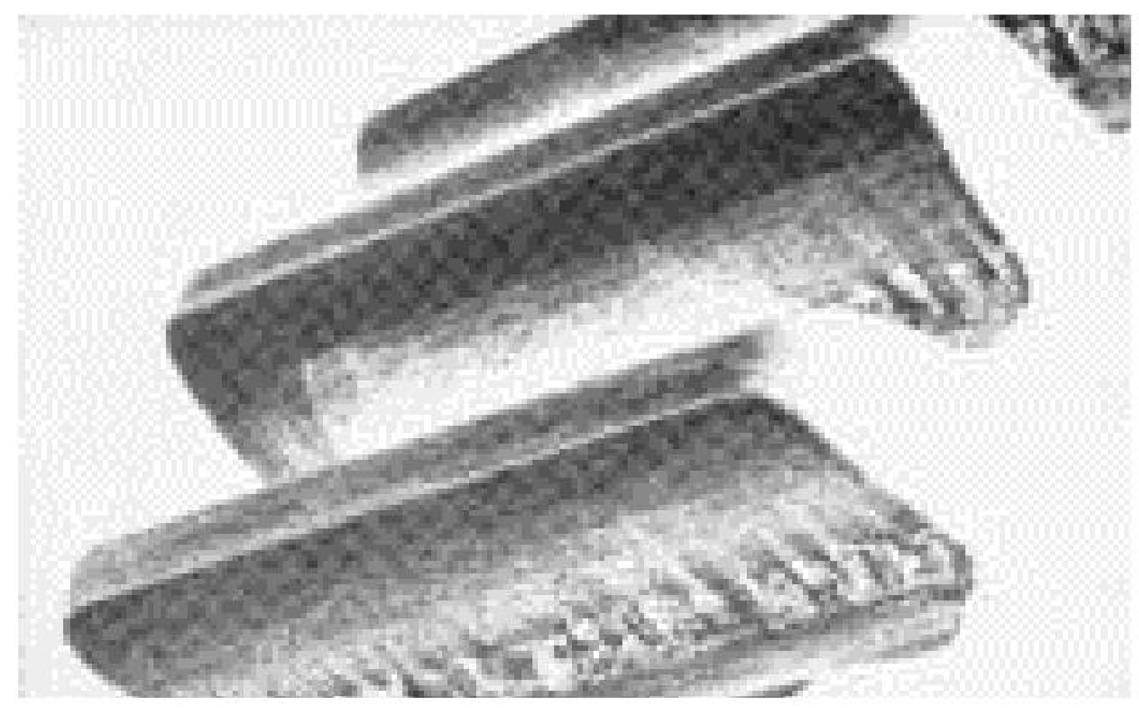

FIGURE 9.27 Interference.

Maintenance Procedure. Since interference usually is the result of improper gear design or manufacture, or of deflection, or of assembling the gears at too close a center distance for the profile shapes existing on the teeth, remedy for the situation should be left to the gear manufacturer.

Grinding cracks are fine surface cracks developed in grinding, usually in a definite pattern or network, caused by improper grinding technique or heat treatment or both. Sometimes they do not appear until the surface has been subject to load. Such cracks can be originating points for fatigue breakage, although sometimes the failure may be of the surface alone with large areas spalling out. (See Figs. 9.28 to 9.31.)

Maintenance Procedure. In some cases, grinding cracks will not cause serious gear-tooth deterioration if gears are properly lubricated. Where overloading, high operating velocities, or high-temperature service cause grinding cracks to enlarge, magnetic inspection and polishing may be useful in overcoming the trouble. If damage continues, consult the gear manufacturer.

Tooth Breakage

Tooth cracking or actual breakage is the end result of gear-tooth deterioration. These conditions are listed for identification purposes only, since their existence indicates a situation already beyond the ability of maintenance procedures to retard.

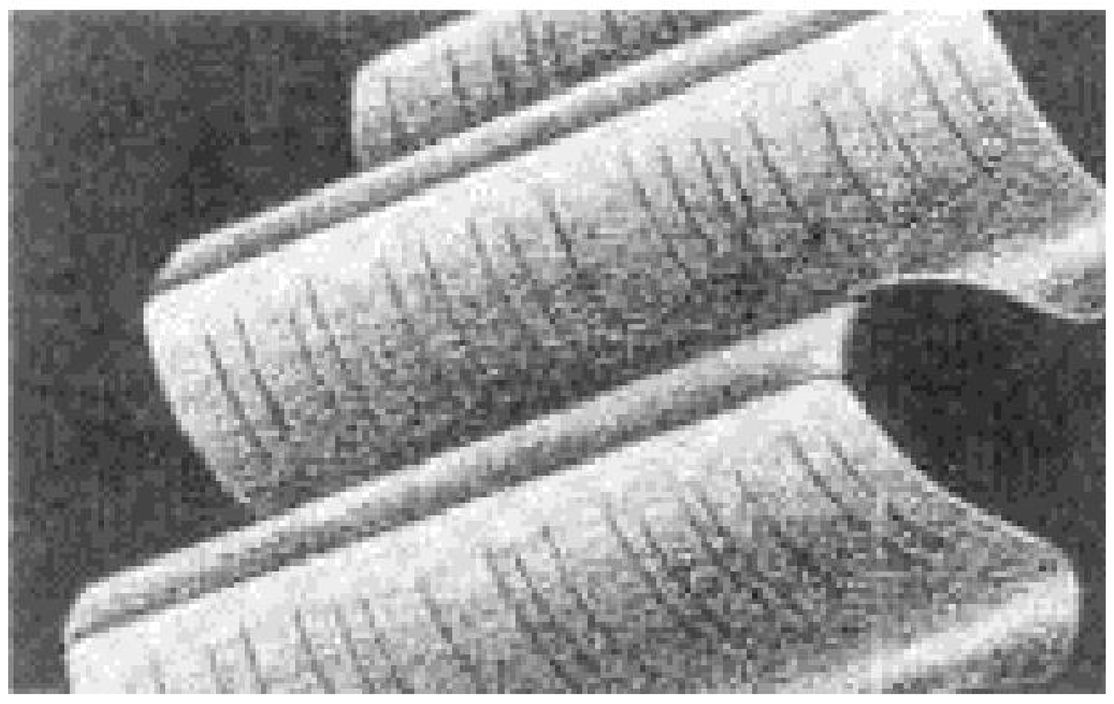

FIGURE 9.28 Grinding checks.

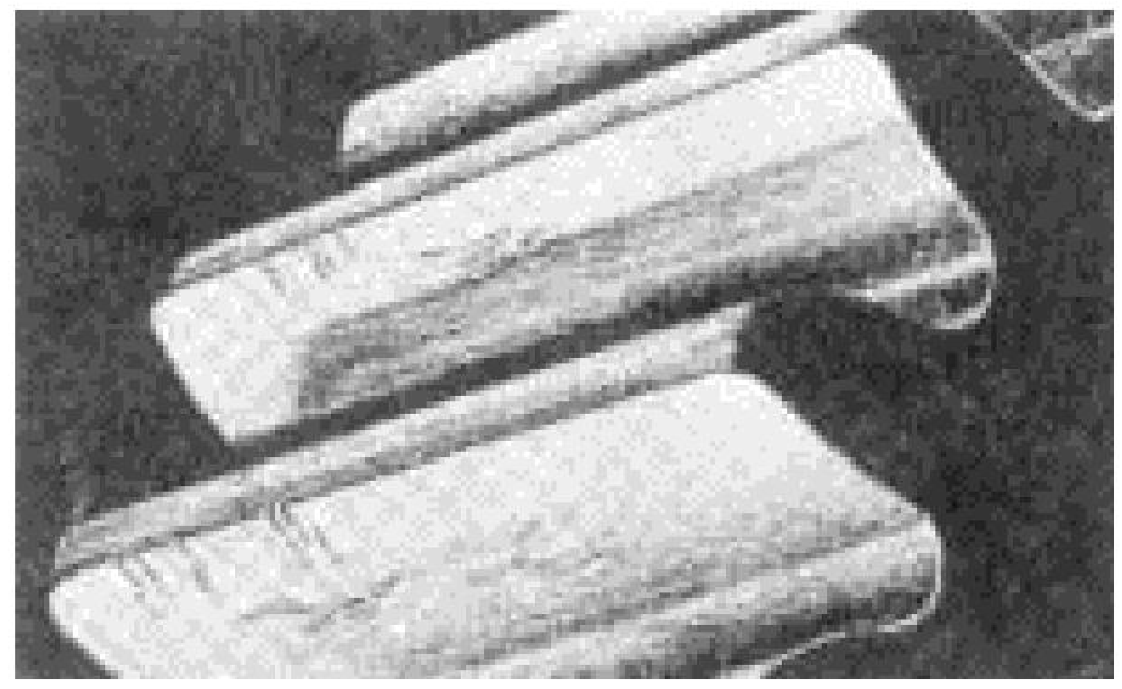

FIGURE 9.29 Cracking.

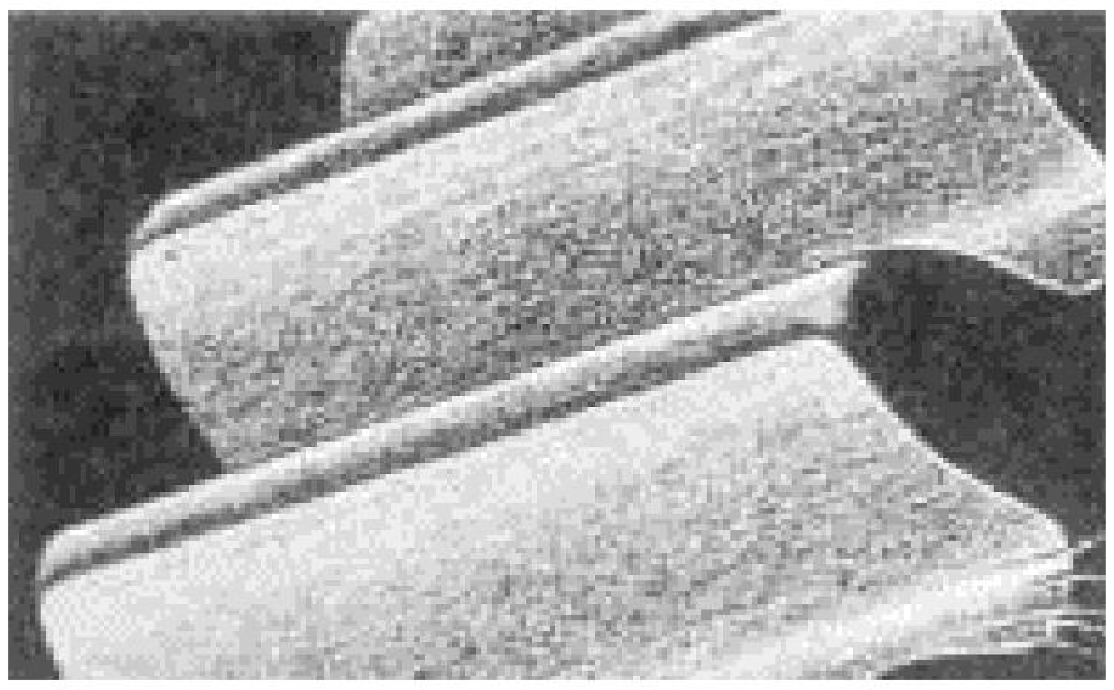

FIGURE 9.30 Quenching cracks.

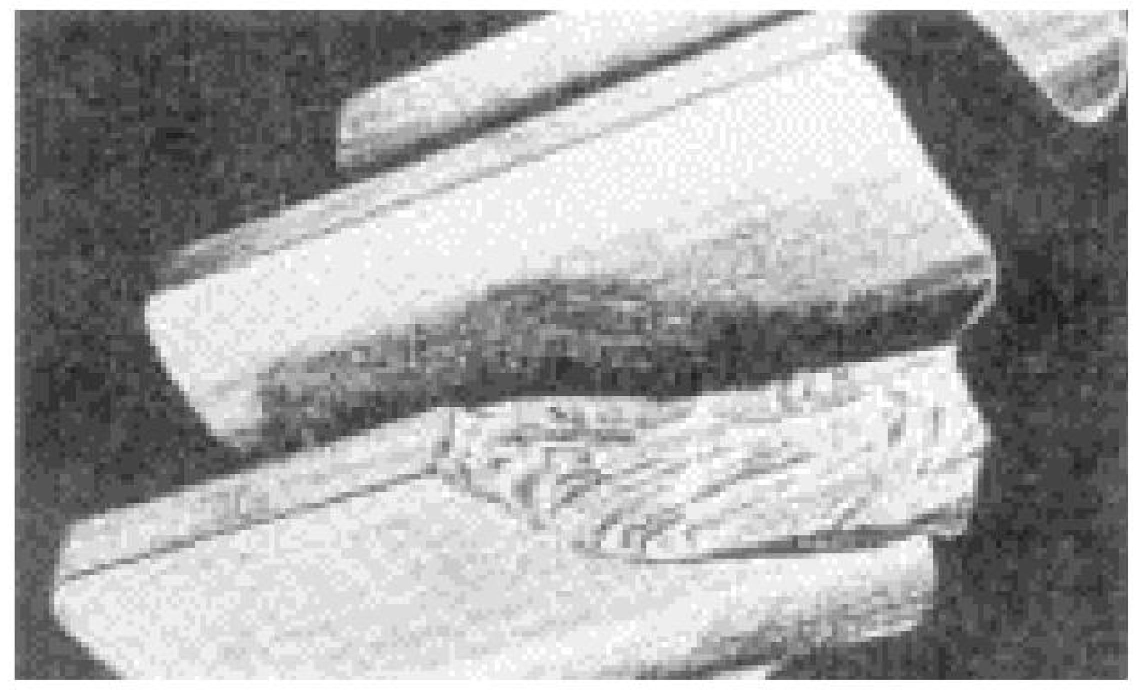

FIGURE 9.31 Overload breakage.

Fatigue breakage is the most common type of failure by breakage. It results from repeated bending stresses that are above the endurance limit of the material. Such stresses can result from poor design, overload, misalignment, or inadvertent stress raisers such as notches or surface or subsurface defects. It originates as a crack on the loaded side, usually in the fillet at the edge of the face, and progresses to complete failure either along the root or diagonally upward across the tooth. Fatigue fractures are usually characterized by a series of contour lines and a focal point in an area that is smooth by comparison. In the case of a subsurface point of origin, the eye (focal point) at the bottom of the cavity is highly polished. (See Fig. 9.32.)

Breakage from heavy wear is a secondary type, since it is a result of another kind of failure or wear. For instance, severe pitting, spalling, or heavy abrasive wear can remove enough metal to reduce the strength of the tooth below the breaking point.

Overload breakage is a rather common type of failure resulting from sudden shock overload and does not show progression of the crack as in fatigue. The fracture will have a silky appearance in the harder and more brittle materials, and a fibrous and torn appearance without a definite pattern in the more ductile metals. Misalignment which concentrates the load at one end of the face is usually

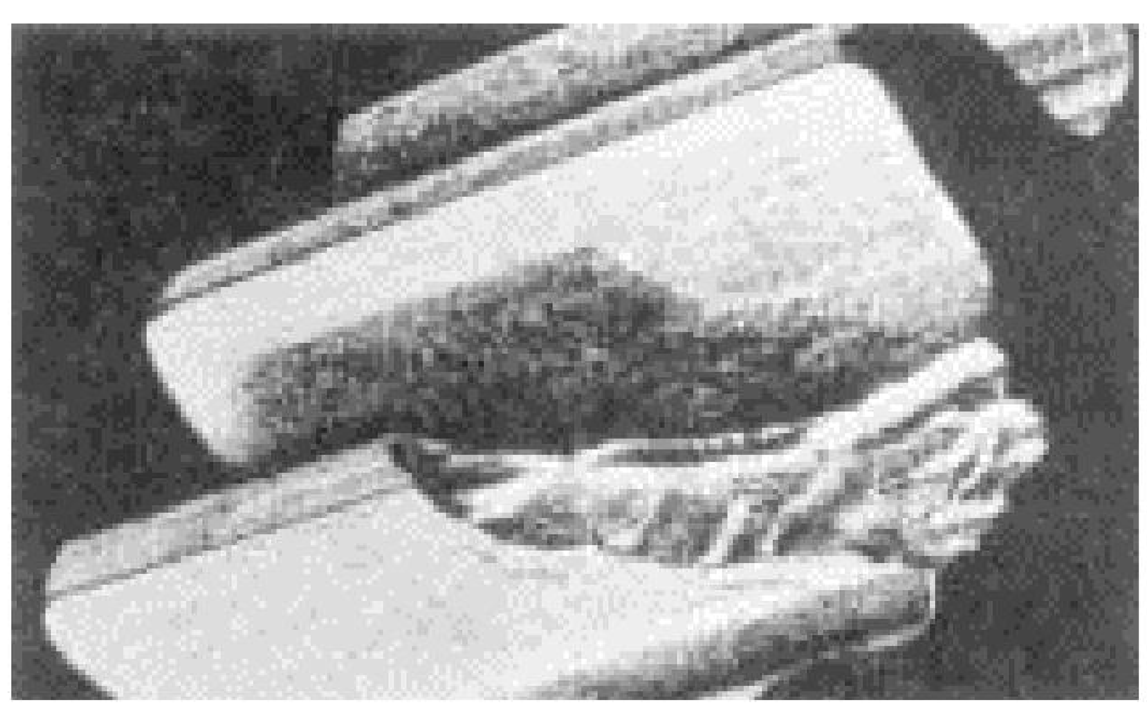

FIGURE 9.32 Fatigue breakage.

the cause, but overload breakage also may be caused by welding of the teeth due to bearing failure, bent shafts, or large pieces of foreign matter entering the mesh.

Quenching cracks result from excessive internal stresses developed by heat treatment and can be originating points for failure breakage. Usually they are visible hairline cracks. They may run across the top land or be radial in direction in the fillet region or be at random direction at the ends of the teeth. If large, the cracks may result in a failure similar to overload breakage after relatively few cycles. In either case, the initial portion of a break will be discolored from rusting or oxidation.

CONCLUSION

Maintenance of gear drives involves proper selection, proper installation, proper loading of the unit, proper lubrication, and periodic inspection. Metallic gears have tremendous service life when properly used and cared for.

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CHAPTER 10
RECIPROCATING AIR COMPRESSORS

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An adequate and dependable supply of air is always necessary for continuous and economical operation of air tools. A specific compressor-maintenance program will go a long way toward obtaining the maximum efficiency from a compressor and eliminating unnecessary shutdown periods. The modern compressor is a precision-built machine, and it should be operated and maintained as such. Too many compressors are installed in out-of-the-way locations and are practically forgotten until trouble develops.

Each major air compressor manufacturer furnishes an installation, operation, and service instruction book with each unit. Many hours of preparation and years of experience are represented in these books. They are included with the compressors so that owners and operators will have sufficient information to install, operate, and maintain the equipment for maximum efficiency. Read the instruction book carefully and become familiar with the compressor construction so that minor adjustments and emergency repairs can be made. Also know who to contact should serious difficulty develop.

Location. For good maintenance a clean, well-lighted location should be selected with enough space allowed to dismantle any parts that may need to be removed for servicing. Too often compressors are located so that it is impossible to remove the pistons, rods, or cylinders without breaking through a wall or moving the compressor. Outline and foundation drawings show the necessary service clearance. Additionally, light overhead lifting capability is highly advantageous when overhaul is required. Near access to clean, cool, outside air will reduce cost of suction air piping. Maintenance and costs are materially reduced where these recommendations are followed.

Foundation. An adequate compressor foundation (Fig. 10.1) is a necessity for satisfactory operation and maintenance of a compressor. A foundation that is designed without sufficient mass and bearing surface will cause vibration of the compressor, resulting in discharge-, suction-, and waterline breakage and excessive wear of compressor parts.

For compressors requiring concrete foundations, the compressor vendor furnishes prints showing the foundation above the floor line plus the weights of the parts to be mounted on the foundation, also the out-of-balance forces that must be absorbed by the foundation. The amount of foundation will

[1] The author is now retired.

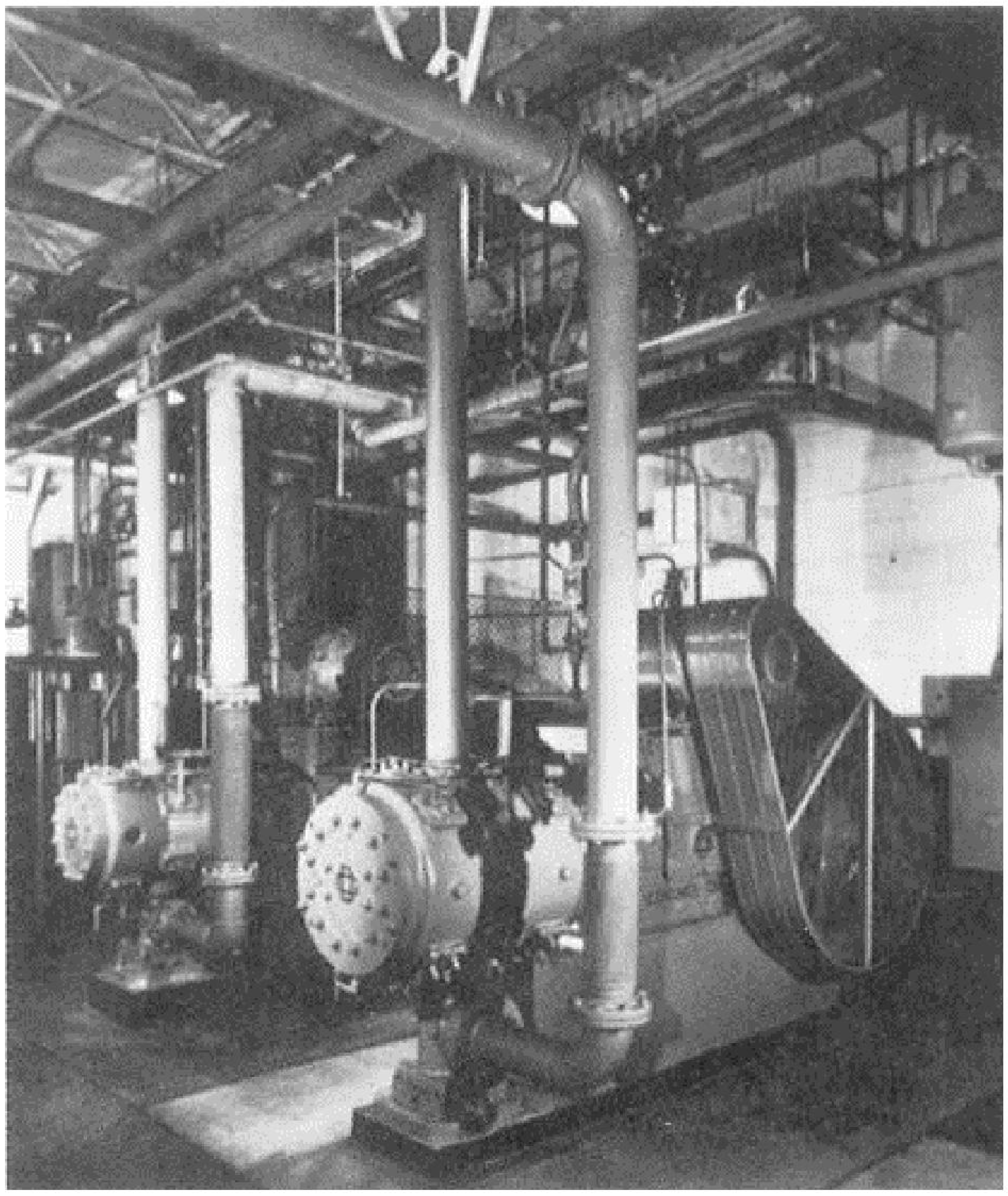


FIGURE 10.1 Compressors on proper foundation.

depend on the type of soil upon which it is being set. To determine the depth and size of the foundation below the floor line, a competent foundation engineer should be consulted who will take test cores and from these calculate the soil carrying capacity. With this information, along with the weights and out-of-balance forces, a foundation can be designed for satisfactory compressor operation.

Many small vertical compressors are installed on existing concrete floors and usually operate very well this way, as the large area of the floor forms a more than sufficient mass to offset any out-of-balance forces of the compressor.

At some locations it is impossible to set the compressor on a foundation or concrete floor that is poured on the ground. It must be located on a floor that does not have a solid base under it. For this type of installation, isolation dampers are used under the base supporting the compressor and its driver. Suction, discharge, and water lines should be attached with good flexible connections to prevent vibration and noises from being carried through the building. There are many manufacturers of isolation dampers, and their engineering should be consulted for recommendations for this type of application.

Air Filters and Suction Lines. Every compressor must be equipped with an air filter which should be the most efficient type made for the service it is applied to. The air filter must be located so that an adequate supply of cool, clean, and acidfree air will be had at all times, with explicit instructions for servicing the air filter posted where the maintenance personnel will always be reminded of the regular servicing required for good maintenance (Fig. 10.2).

At some locations it is necessary to place the air filter away from the compressor because of unfavorable surrounding conditions. Care must be used in providing a suction line to a compressor. It must be tight, free of dirt, chips, and scale, corrosion-resistant, and of adequate size for the length necessary to connect the air filter to compressor suction. PVC piping can provide a good solution to most

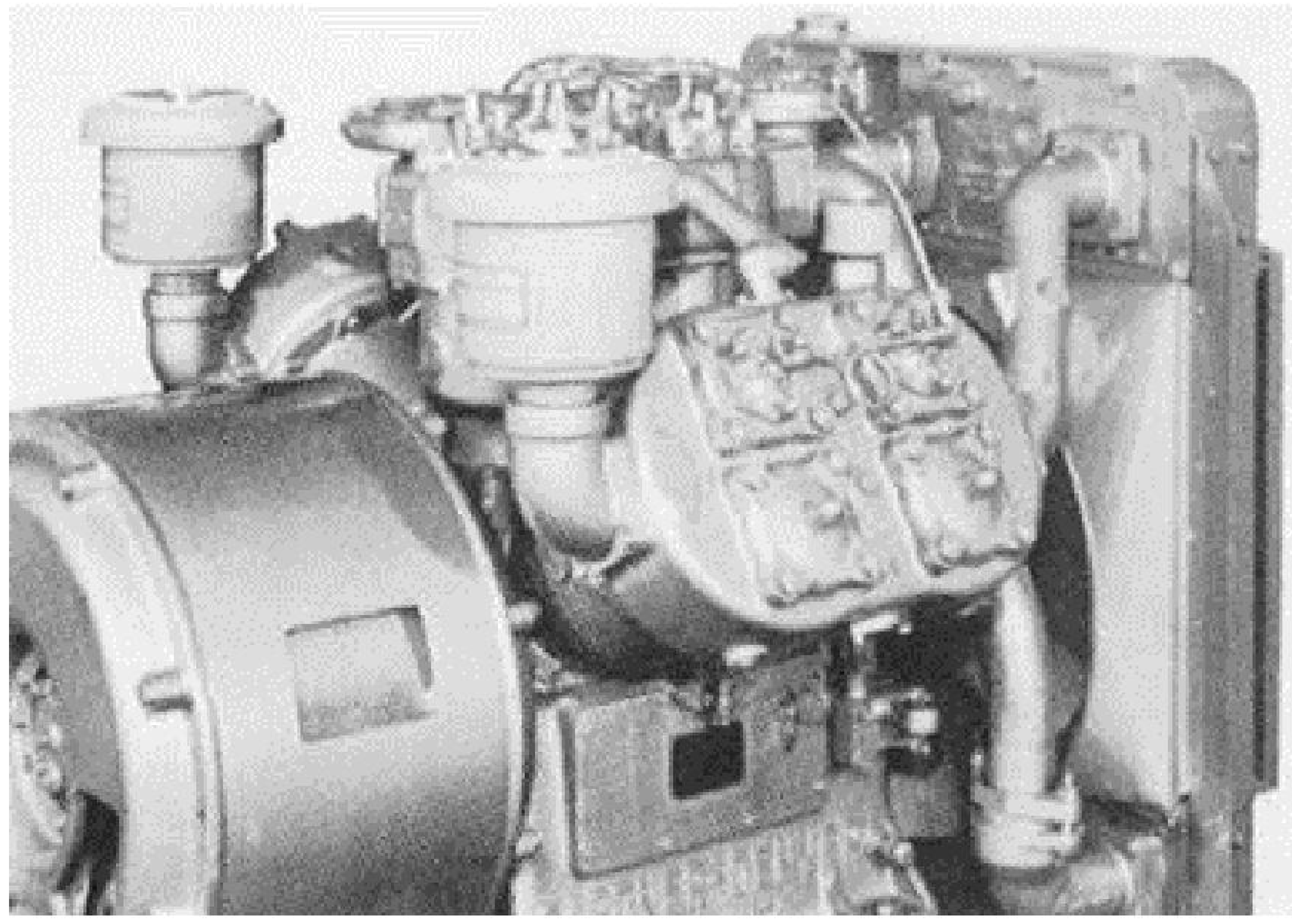


FIGURE 10.2

of these requirements. Care should be taken to avoid acoustically resonant pipe lengths, which can result in excessive pulsations within the piping. The compressor manufacturer should provide a list of acoustically resonant pipe lengths to avoid. Normally the shortest possible suction line is preferable.

The time interval for cleaning an air filter depends on the type of filter and its location, and is best determined by the differential pressure across the filter. Reputable air filter manufacturers offer differential pressure devices which indicate when the cleaner requires servicing.

Air-Receiver Location and Capacity. Air receivers often are considered accessories to air compressors and, for many applications, are not correctly installed or properly sized. Proper installation and proper sizing are very important for both compressor and air-line systems. An air receiver absorbs pulsations in the discharge line from the compressor and smooths the flow of air to the service lines. It serves as a reservoir for the storage of compressed air to take care of sudden and unusual momentary demands in excess of the capacity of the compressor. Another of its functions is to precipitate moisture that may be condensed in the receiver and prevent it from being carried into the air-distribution system.

The preferable location for an air receiver is as near the compressor as possible so that the discharge line can be of minimum length, eliminating pressure drop between the receiver and compressor. Many receivers are located outside the compressor room and are exposed to the weather, offering difficulties when the temperature drops low enough to cause freezing. An ordinary top-outlet safety valve can be frozen shut, creating a hazard; the valve should be placed with opening down, thereby keeping water out and allowing the valve to function if necessary. Should the compressor be shut down, allowing no air to pass through the receiver, the drain valve or mechanism can freeze, resulting in possible breaking of the parts making up the drain.

The size of the receiver usually is recommended by the compressor vendor, who has charts listing the necessary receiver sizes for various compressor sizes. Start-and-stop compressors require larger receivers than do continuously operated compressors, to keep them from starting too often. Each start requires electrical inrush to the motor, which can cause expense by increasing electrical requirements beyond normal electrical demand.

Air from the compressors should flow into the receiver at the bottom and out at the top. Condensate is a troublesome factor in the system. Use an efficient water-cooled aftercooler and separator between the compressor and receiver. The aftercooler will condense the moisture and collect most of it in the separator, which can be drained manually or automatically. The aftercooler dries and cools air, which promotes efficiency and safety. Most aftercoolers will cool the air within 15°F of the incoming cooling water. Where water supply is short or expensive, air-cooled aftercoolers are available. They are not as efficient as the water-cooled but, if properly sized and of good quality, usually will cool to within 20 to 30°F of the ambient temperature.

Always consult the compressor vendor about receiver problems. Many states are exacting about pressure-vessel requirements; pressure vessel must meet the codes for safety and pass inspection by the insurance companies.

Starting a New Compressor. Before a new or repaired compressor is started, careful check must be made of the lubricating system, making certain all places needing lubrication have been oiled per manufacturer's requirements. On compressors having a forced mechanical lubricator, crank or pump by hand until it is certain the oil is getting to the parts requiring lubrication, as some initial lubrication is required before the unit is started. Tighten all bolts, nuts, and cap screws. Turn the compressor over by hand wherever possible to determine that there is no interference or binding of working parts.

In the case of compressors requiring cooling water from a water main, turn on the water and check for leaks and for circulation through all parts requiring cooling water. For compressors having a self-contained water-cooling system, fill it and check to see that all air is out of the cooling system.

Check the discharge line from the compressor to the receiver, and if there are any globe, gate, or check valves anywhere between the compressor and receiver, be sure the valves are open and that there is a safety valve between the compressor and valves. The safety valve is a necessity, as it is possible that a valve might be left closed when the compressor started, resulting in an explosion should there be sufficient power in the driver, or should the overload protection fail to act.

If all points have been checked, apply driving power momentarily and let the machine coast to rest. Close observation during the coasting period will reveal any excessive tightness and will confirm proper direction of rotation. The time that the unloaded machine continues to roll after driving power has been removed gives a fair indication of no-load friction; if direction of rotation is correct and no other trouble is evident, the unit can be run without load. On units with a pressurized lubrication system for the crankcase, check immediately after start-up for proper oil pressure; if adequate oil pressure is not attained within about 10 sec after start-up, shut the unit down and determine the cause. While running unloaded, check any mechanical forced feed lubricators for proper drops-per-minute feed rates to cylinders and piston packings as specified by the operator's manual.

Operating a water-cooled compressor with too much cooling water through the system will cause excess condensate and cylinder wear because a cold cylinder will not lubricate properly; and because lubrication is affected, excess horsepower is required, adding to both maintenance and operating costs. A good rule is to hold the outlet temperature of the water between 110 and 120°F. This range will allow for good cooling and lubrication and also will keep condensation in the cylinder to a minimum. The introduction of cold water into cylinder water jackets must also be avoided in order to prevent condensation from occurring on interior cylinder walls. Usual practice is to circulate the cold supply water through aftercoolers or other water-cooled heat exchangers nearby and then direct the warmed water to the compressor cylinder jackets. In the case of multistage compressors, the cooling water is initially introduced into the interstage intercoolers and then directed to the cylinder jackets. It is good practice to have the temperature of the cooling water entering the cylinder jackets at least 10°F above the temperature of the air entering the cylinder to preclude condensation.

After running from 1 to 2 hr unloaded, with periodic stops to check for any heating of bearings or other working parts, apply partial load and build up to maximum load and pressure gradually. The entire breaking-in period should consume a minimum of 4 hr.

The importance of a break-in run cannot be stressed too strongly. The time and care spent in giving the running surfaces a polished finish pay dividends by increasing compressor life. After the initial run, compressor operation resolves itself into maintaining a clean air supply, feeding sufficient cooling water, and supplying adequate lubrication.

All the foregoing requirements are necessary to get a compressor ready for efficient operation and to hold maintenance costs to the minimum. Routine maintenance must now be set up and a definite pattern followed.

Lubrication. The most important check for any compressor is the lubrication system (Fig. 10.3). Keep the compressor well lubricated; check the oil level at least once every 8 hr of operation. Use

only oil and greases as recommended by the compressor manufacturer. The oil used should have a low carbon-forming tendency and sulfur content and contain an oxidation inhibitor. It is important to use the correct viscosity oil, consistent with existing temperatures. The instruction book lists these conditions.

Because dust, dirt, and atmospheric conditions are different at various locations, it is not practical to state definitely how often the oil should be changed in the crankcase of an air compressor. Oil will become contaminated with foreign materials held in suspension and will also oxidize. The time for oil changes is regulated by local conditions and must be determined by the discoloration and physical condition of the oil. Convenient lab services are available to analyze mail-in oil samples to assist in determining proper oil change periods and to provide warning of excessive frictional surfaces wear or other unusual foreign material contamination.

When oil changes are made, it will always pay to remove a handhole or cover plate and wipe the inside of the crankcase or power end clean with lint-free rags. If impossible to wipe out, use a good grade of flushing oil to remove any particles that may have settled on the crankcase floor. On units with pressurized crankcase lubrication, clean the oil pump suction strainer located in the lower part of the oil sump. Oil filter elements should also be replaced or cleaned (if applicable). When refilling the compressor oil sump, be certain the filling container is free of all dirt, grit, or dust. This simple point is often overlooked.

Nonlubricated Cylinders. A significant portion of double-acting reciprocating compressors being installed, or currently working, are equipped with nonlubricated cylinders. These cylinders are designed to avoid any metal-to-metal contact between pistons and cylinder bores, and are usually equipped with some type of filled-PTFE (polytetrafluoroethylene) piston rings and rider rings. Piston rod packings are also nonlubricated and usually equipped with either carbon or filled-PTFE packing rings. Nonlubricated, double-acting cylinders are also usually equipped with extended length piston rods and distance pieces so that no portion of the piston rod entering the lubricated crankcase can enter the nonlubricated piston rod packing. Additionally, the piston rod is usually equipped with a baffle ring to prevent creepage of lubricant from the lubricant-wetted portion of the rod to the portion entering the rod packing.

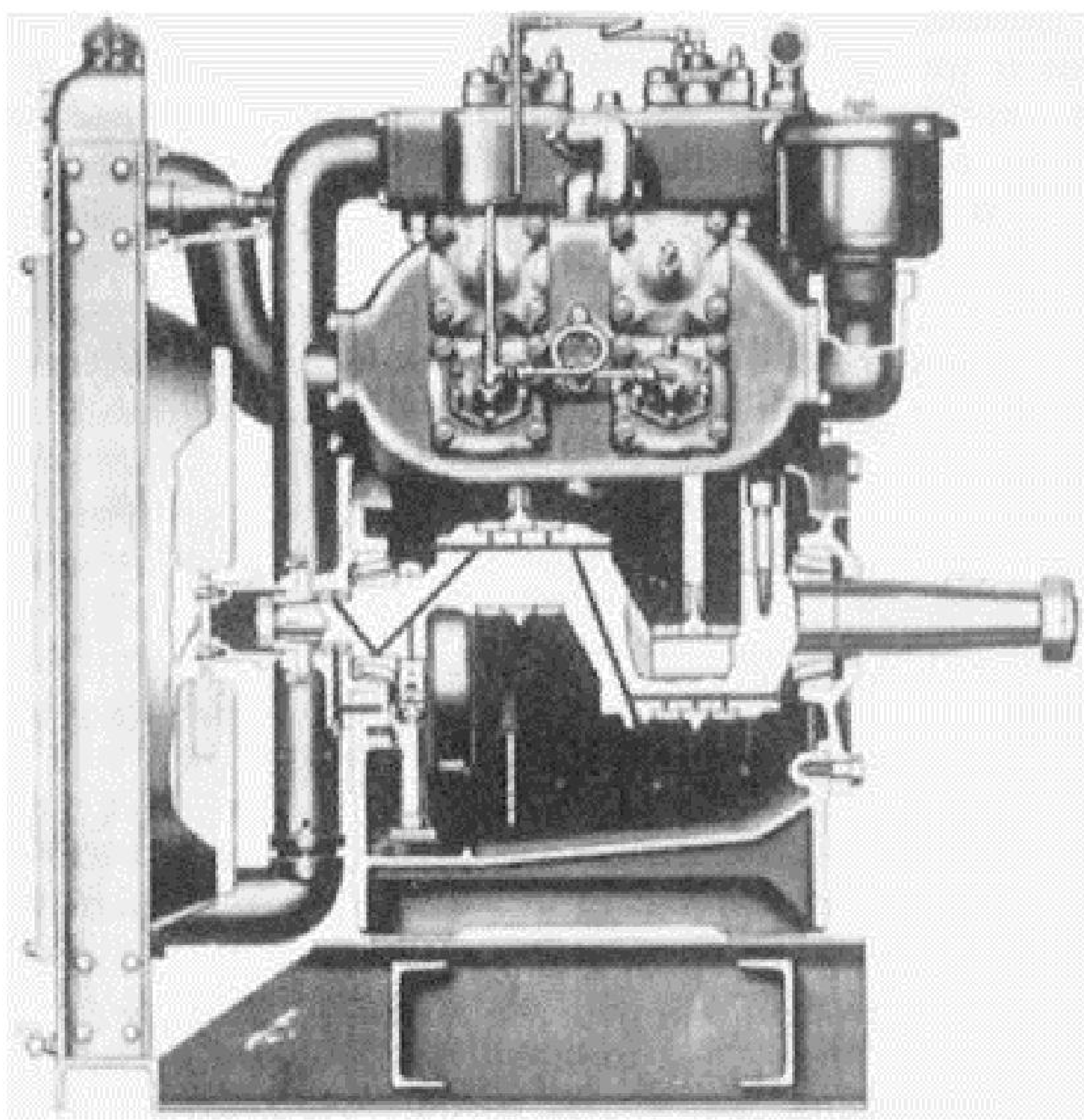
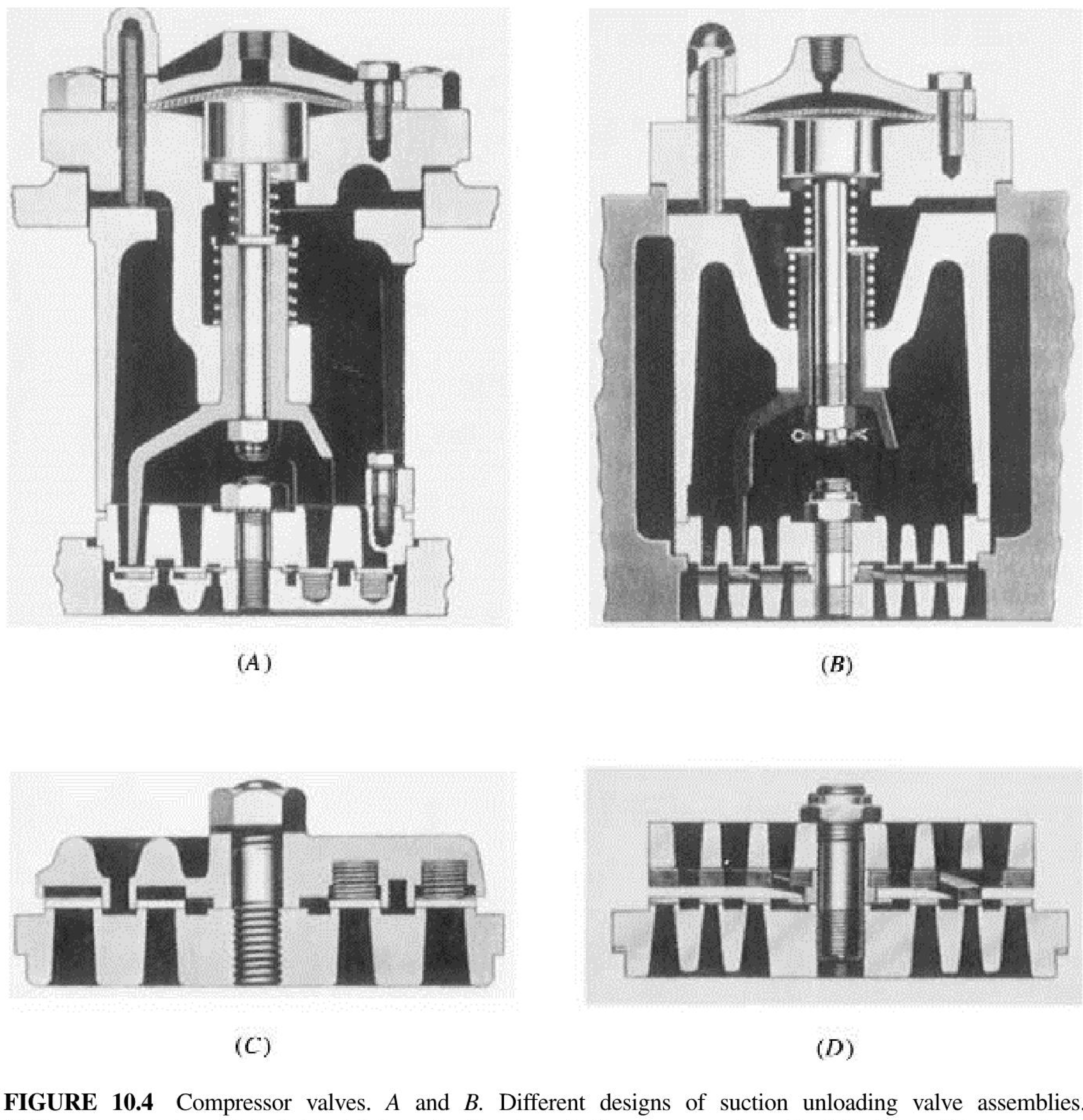


FIGURE 10.3 Pressurized lubrication system.

It is critically important to the expected service life and reliability of nonlubricated cylinders that due care be taken to prevent the intake of particulate and liquid contaminants into the cylinder, and the formation of condensate within the cylinder. Start-up procedures are the same as that described for lubricated cylinder units, with the exception of comments related to cylinder forced feed lubricators.

Valves. Reciprocating compressor valves must be kept in first-class operating condition, as leaking or broken valves cause excessive operating temperatures and loss of air delivered. It is therefore important to check the valves periodically and be certain they are always in good operating condition (Fig. 10.4).

The checking time for valves depends on several conditions, such as efficiency of the air cleaner, carbon-forming tendency of oil used, and the overall condition of the compressor. If the air cleaner is efficient and regularly serviced, excess dirt will be kept out of the airstream and dirt will not lodge in the valves. By using low-carbon-forming oil, the carbon buildup on the valves is held to a minimum. Synthetic lubricants such as diesters, polyol esters, and polyalphaolefins are commonly available which have characteristics highly desirable for air compressor cylinder lubrication. Although higher in cost than mineral oils, the synthetic oils result in cleaner-running valves and significant reductions in deposits formed in all hot areas of cylinder air passages and air piping. Most currently



(*A*) (*B*) (*C*) (*D*)

FIGURE 10.4 Compressor valves. *A* and *B*. Different designs of suction unloading valve assemblies. *C*. Compressor valve with individual disks and coil springs. *D*. Compressor valve with plate disk and finger springs.

produced compressors are compatible with the above synthetic oils; however, compatibility with existing and older compressors should be confirmed with the manufacturer. Additionally, prior to conversion of any existing and older compressor and associated air distribution systems from mineral oil to synthetic oil lubrication, due care must be given concerning the solventlike action of most synthetic oils on hydrocarbon deposits within the compressor and air-distribution system. Proper procedures are available from the synthetic oil vendor and compressor manufacturer. For single-acting vertical compressors, the pistons, rings, and cylinder walls should be kept in good condition so that excess oil will not pass these parts. Low oil consumption adds to valve life by eliminating unnecessary carbon deposit. No set checking time can be recommended; it will need to be determined by actual investigation by the maintenance personnel. On a new unit the valves should be checked after 200 hr of operation.

Many compressor owners have found it helpful to have a spare set of valves so that a change of valves can be made immediately, and the replaced set reconditioned when time allows.

When valve troubles occur, there are several means of locating the valve or valves causing the difficulty. The first symptoms usually are low net air delivery and heating around the valve compartments. On a single-stage compressor, the usual method used is to check the temperature of the valve cover plates and examine the valve under the cover plate that is the hottest. If suction valves are leaking, a definite blow-back noise can be heard in the air cleaner when the compressor is operating under load.

On two-stage compressors, the intercooler pressure gage is used as a guide to locate defective valves. When low intercooler pressure occurs, examine the valves on the low-pressure cylinders, and when high intercooler pressure is found, examine those on the high-pressure cylinders. By checking the temperature of the valve cover plates, the defective valve can be located under the cover plate that is the hottest. If high-pressure suction valves are leaking, the intercooler-gage hand will fluctuate above normal intercooler pressure and the intercooler safety valve will pop. If high-pressure discharge valves are leaking, the intercooler-gage hand will rise steadily and pressure will build up in the intercooler until the intercooler safety valve will release it.

When low-pressure suction valves leak, the air will blow back through the suction line and air cleaner if the compressor is operating under load. Leaking low-pressure discharge valves will cause the intercooler pressure gage to fluctuate below normal intercooler pressure.

Since the valves are such an important part of the compressor, the information given in the instruction book must be followed when removing and installing them.

Wear between the valve disks or plate and the valve seat appears as indentations in the valve disks or plate, leaving a shoulder. The valve disks or plate are normally replaced if they show any amount of wear or the presence of any nicks, chips, or cracks.

Most worn valve seats can be resurfaced. On some types of valves, it is necessary to check the lift of the valve after resurfacing the seat, and if found to be more than recommended by the vendor, the bumper will need to be cut down to get the correct lift. Too much lift causes rapid wear and breakage.

Most valves usually have raised valve seats, and when the seat is refinished it is not necessary to do anything to the bumper, as the lift will still be to manufacturer's specifications.

Whenever a valve has been overheated, replace all the valve disks or plates and springs, because excessive temperature resulting from this heat will reduce the life of these parts and may result in breakage, causing damage to the compressor.

Most compressor valves have a gasket under the seat. This gasket must be in first-class condition; should it show any imperfection, replace it, as a leaking valve-seat gasket will eventually blow out.

The cover-plate gaskets are also important, and when installing valve cover plates, be sure the gaskets are in good condition. It is imperative that the valve cover-plate nuts or cap screws be pulled down evenly. Do not completely tighten one side and then the opposite side, as this will cause uneven gasket pressure, resulting in leads or sprung cover plates.

Several types or designs of valves are used by different compressor manufacturers, and in order to get the proper installation in the compressor, refer to the instruction book that was furnished with the compressor. Too much care cannot be used when maintaining and installing the valves and the component parts.

Piston Rings. Valves often are the cause for lost compressor efficiency, but should the valves be known to be satisfactory, the lost efficiency could well be in the piston rings. Piston-ring wear usually is very slow when the rings are properly lubricated, but operating time will eventually wear them so that the gap increases and the piston-ring lands wear to the point where some of the ring valving action is lost, allowing for blowby through the gap and around in back of the ring.

When excess blowby is suspected, remove the pistons and check the piston-to-cylinder wall clearance and the piston rings for the amount of wear, determining the parts that will need to be replaced. Scored cylinders always will allow excess blowby, adding to operating costs due to lost horsepower and fast wear.

Compressors having automotive-type pistons should have the wrist-pin fits checked when new piston rings are installed, and if found loose, the pin bushings should be replaced. Often the added drag on the cylinder walls caused by new piston rings will result in a pin knock when too much clearance is allowed.

Piston Rings. In some cases it is possible to rebore cylinders and obtain pistons and rings for certain oversize conditions, such as 0.005, 0.010, 0.020, or 0.030 in. oversize. In still other cases, compressors are provided with replaceable cylinder liners. The compressor operator's maintenance manual should indicate the repair options available.

In the case of nonlubricated cylinders, particular care should be taken to see that piston rider rings do not wear to the point that allows piston-to-bore contact. As might be expected, horizontal and angular-mounted cylinders are subject to higher rates of rider ring wear than vertically mounted cylinders. Each time that valves are removed for inspection or repair, a feeler gage check should be made of the minimum piston-to-bore clearance. If the minimum clearance is 0.015 in. or less, it is time to take action. When replacing the rider ring, carefully examine the compression rings for wear and end gap; follow the compressor manufacturer's recommendations as to when rings should be replaced.

Bearings. Crankpin bearings are usually the automotive-insert type (Fig. 10.5). To correct problems with the insert type, the installation of new inserts will serve. Should the crankpin surface be damaged, it can be ground undersize, built up by plating, metallizing, or plasma arc processes, and finish ground to original new condition dimensions. This allows the use of standard dimension replacement inserts. In certain cases, depending upon availability and cost of undersize inserts, it may be more feasible to grind the damaged crankpin undersize to fit available undersize inserts and omit the process of restoring the crankpin to original new dimensions.

Double-acting compressors have crosshead pins which operate in crosshead-pin bushings which have no adjustment. If a failure occurs, the crosshead-pin bushings must be replaced. Because of different fit requirements for the various compressor manufacturers, the running fit must be obtained from the compressor instruction book (Fig. 10.6).

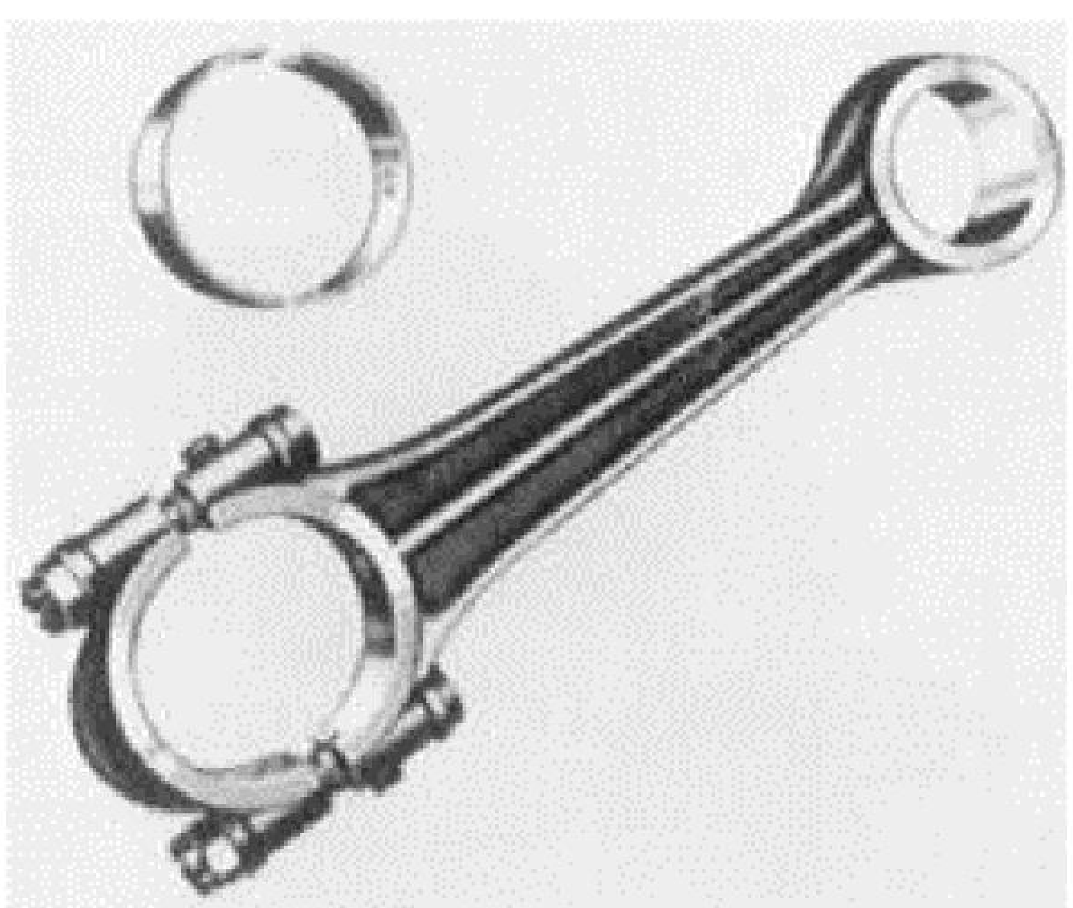


FIGURE 10.5 Automotive insert crankpin bearing with connecting rod.

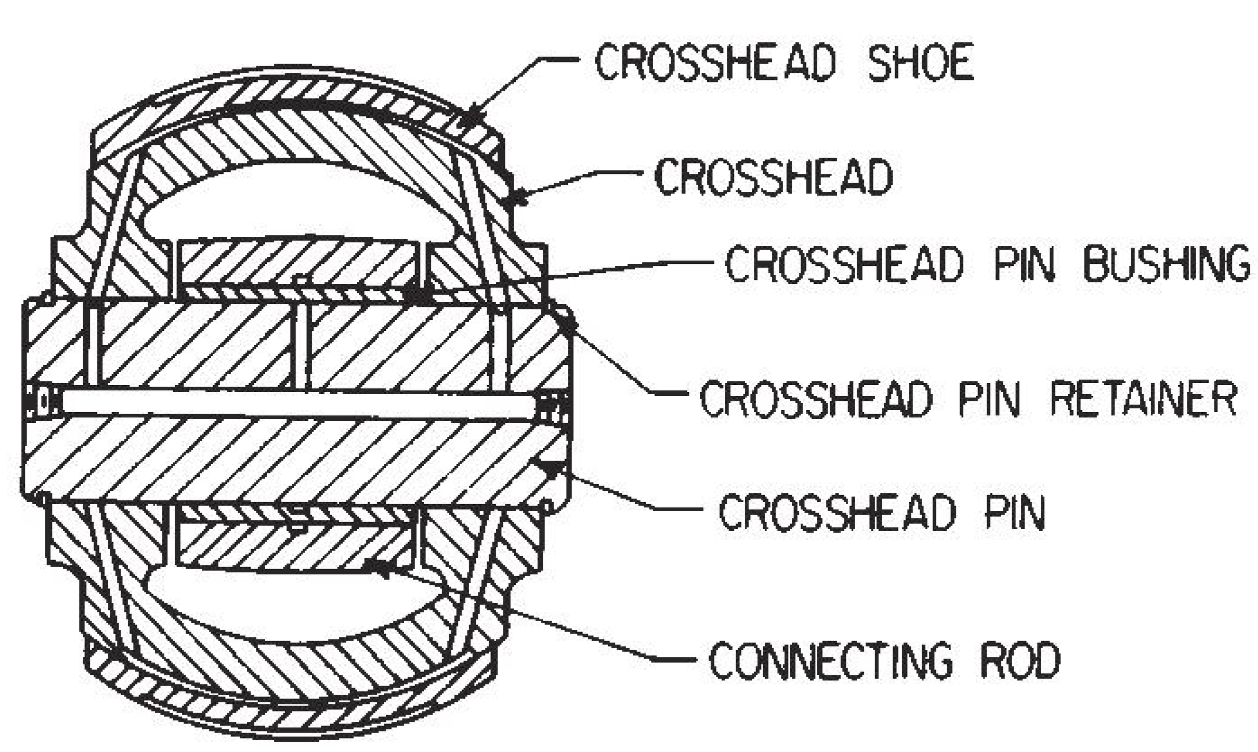


FIGURE 10.6 Crosshead pin bearing and pin.

Many different constructions are used for the main bearings in both single-acting and double-acting compressors, no matter whether they are sleeve or antifriction bearings. Antifriction, single-row, tapered-roller-bearing adjustment is made by removing or adding shims. Double-row tapered roller bearings have an adjusting nut locked on the shaft. Unlock the nut and turn it to move the cone in on the cup. For trial purposes use a feeler gage and get about 0.002 in. over the free rolls. Check bearings for heat and noise after starting, as it may be necessary to either tighten or loosen them slightly (Fig. 10.7).

Intercoolers and Aftercoolers. These are important compressor parts that often are neglected to the extent that they become inefficient. The most important maintenance is simple, and that is the proper draining of the moisture traps or compartments. Any type of cooler is a condenser, and the condensate, if not drained regularly, will build up until water is carried over to the high-pressure cylinders in the case of an intercooler, and on into the air receiver and air lines in the case of an aftercooler. Coolers should be drained regularly, according to existing humidity condition. The surest way to ensure draining is the use of automatic or timed drain traps on the intercooler, aftercooler, and air receiver.

Water-cooled intercoolers and aftercoolers are subject to buildup from the mineral content in water, which, if not removed, will eventually affect cooling; therefore, these coolers need inspection for deposit removal.

Air-cooled intercoolers and radiators must have the core sections cleaned on the outside, because dirt will lodge in the core, reducing heat dissipation. For removal of dust, air blown through in a direction opposite the usual flow will do; but in case the dirt is contaminated with oil, a solvent should be applied, allowed to soak for a while, and then blown clean.

Cleaning. An important item for proper compressor maintenance is keeping the compressor clean on the outside surfaces. Dirt and oil will make an insulation which hinders heat dissipation to atmosphere; this is especially true for an air-cooled compressor, which must depend on all heat dissipation through the cylinder and cylinder-head surfaces. When dirt is allowed to accumulate on the surfaces of a compressor, it is certain some will find its way into the working parts. A well-kept clean compressor will pay dividends with a good appearance plus reduced operating and maintenance costs.

Unloading. Practically every compressor manufacturer has his own type of air-unloading and control system; to cover all types would require complete data for each system. Some compressor vendors use several types; so for servicing and unloading system and its control, it is necessary to refer to the instruction book.

Some common unloading systems are suction unloading valves, suction closure device, centrifugal unloaders, and bypass systems. Most of these controls are operated by means of a pressure switch and a three-way valve actuated by a solenoid. Another means of actuating the unloading device is a pneumatic pilot, of which there are several types on the market.

Packing. Double-acting compressors using piston rods have oil-stop-head packing and cylinder-head packing which require periodic checking. The oil-stop-head packing is usually a set of metallic scraper

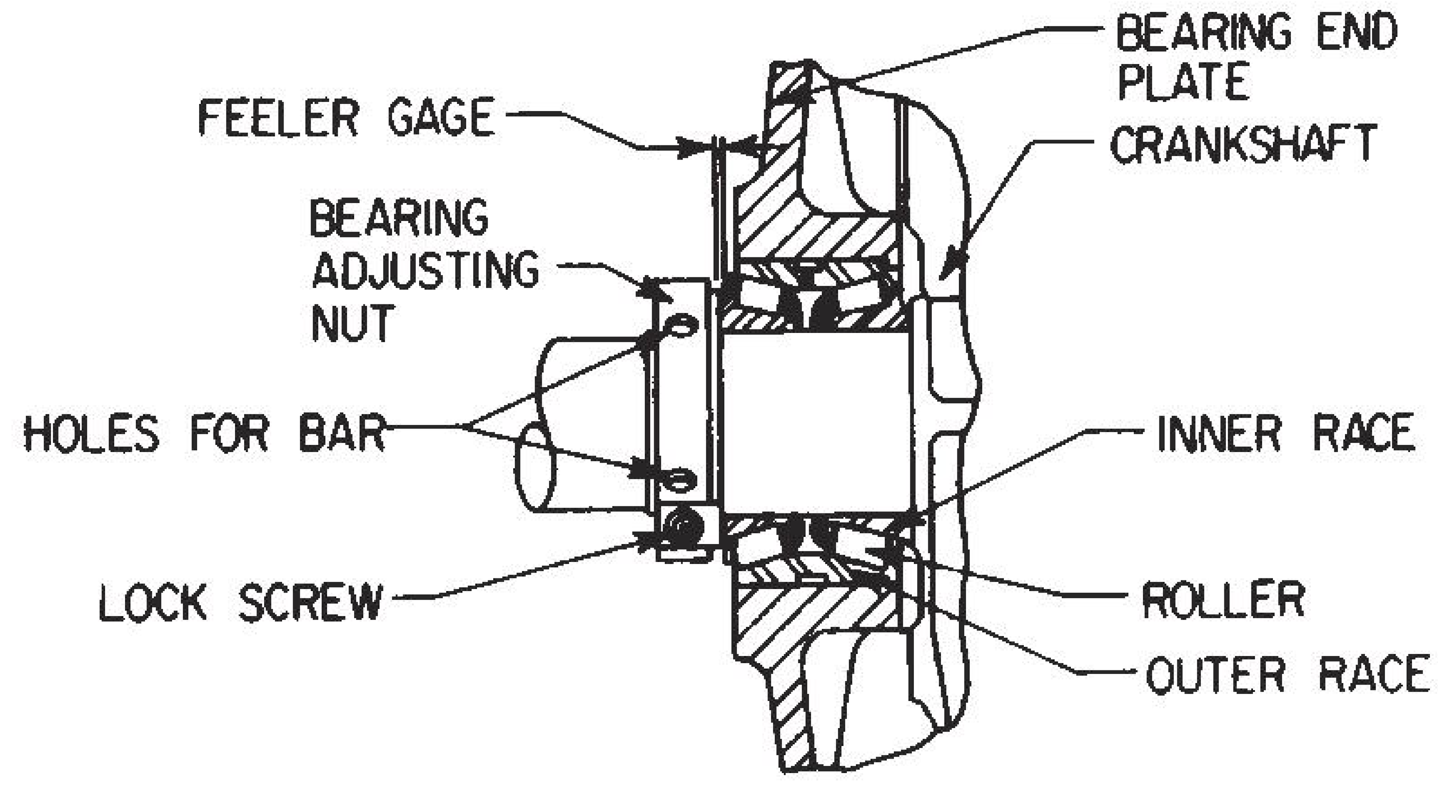


FIGURE 10.7 Crankshaft main-bearing adjustment for double-row tapered roller bearings.

rings. They require very little attention because they are designed to scrape oil off the rod, yet get excellent lubrication. Should the piston rod become damaged, the packing will be ruined and new packing required. Never put new scraper rings on a piston rod that is nicked, scratched, or worn (Fig. 10.8).

The cylinder-head packing usually is the full floating design with self-adjusting packing rings. The material, style, and quantity of the packing rings and number of lubrication lines depend on the type of gas being compressed and the discharge pressure. Some applications require special packing, such as vented and/or the elimination of nonferrous packing-ring and gasket materials (Fig. 10.9).

Metallic packing, after it is installed and worn in, requires little attention. However, the piston rod should be checked where it passes through the packing, and if any scratches are present, the packing must be removed and inspected for embedded material causing the scratches. As long as the packing does not leak or show any signs of marking the rod, it should not be disturbed, as the metal rings are self-adjusting for the slight wear that occurs under normal operation. Rod packings for nonlubricated cylinders function basically the same as for lubricated cylinders except that no lubricant is used and materials of construction are different. The packing ring sets are still of the segmental self-adjusting type but are, in most cases, of quite different material. Variations of carbon, filled-PTFE, and resin-bonded composites, among others, are employed for packing rings. Packing cups and glands are either made from corrosion-resistant material (stainless steels, brasses, bronzes, etc.) or are plated to resist corrosion.

The service check chart in Table 10.1 lists the common causes of malfunctions of mechanical parts of compressors.

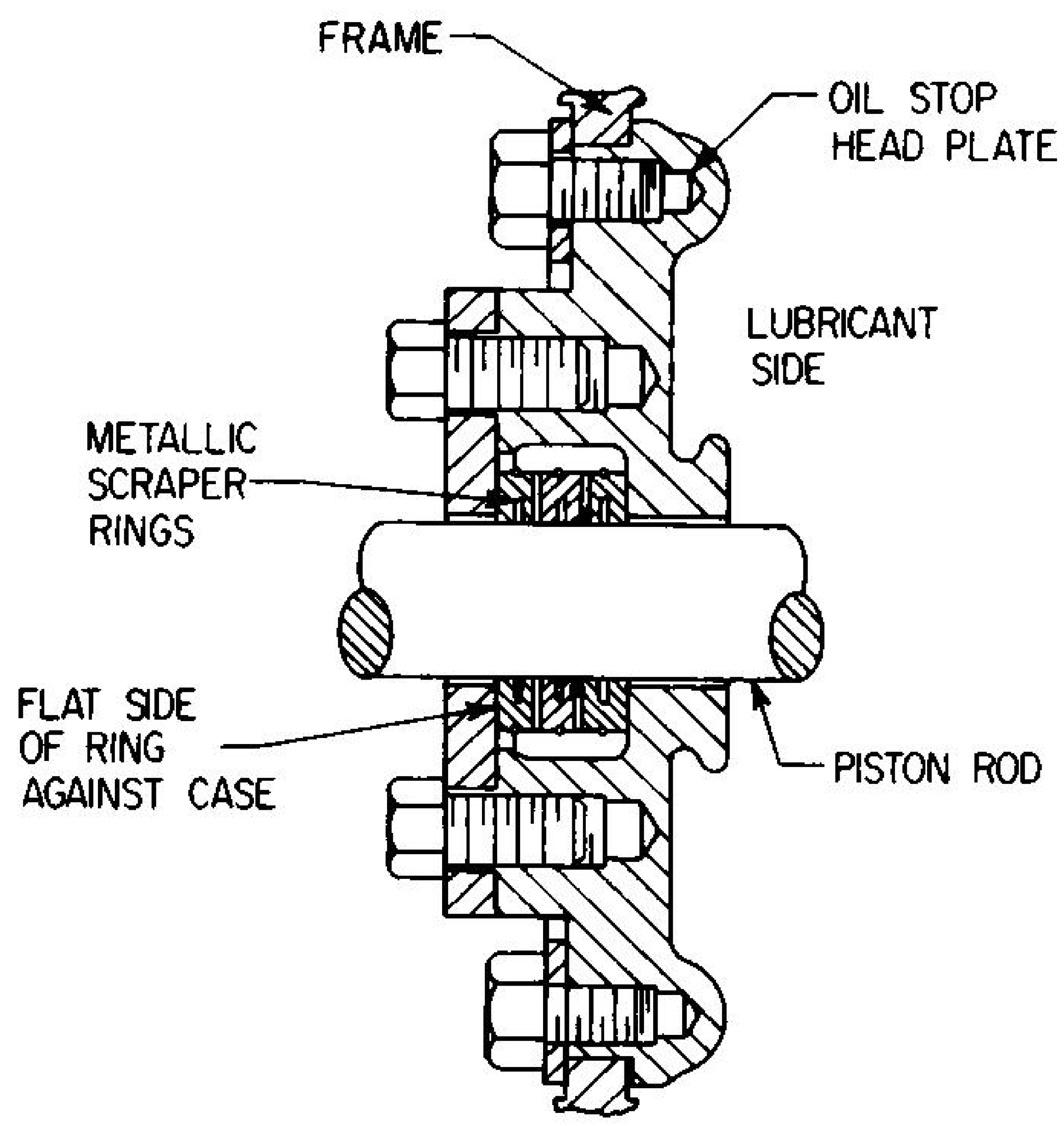


FIGURE 10.8 Oil-stop-head packing.

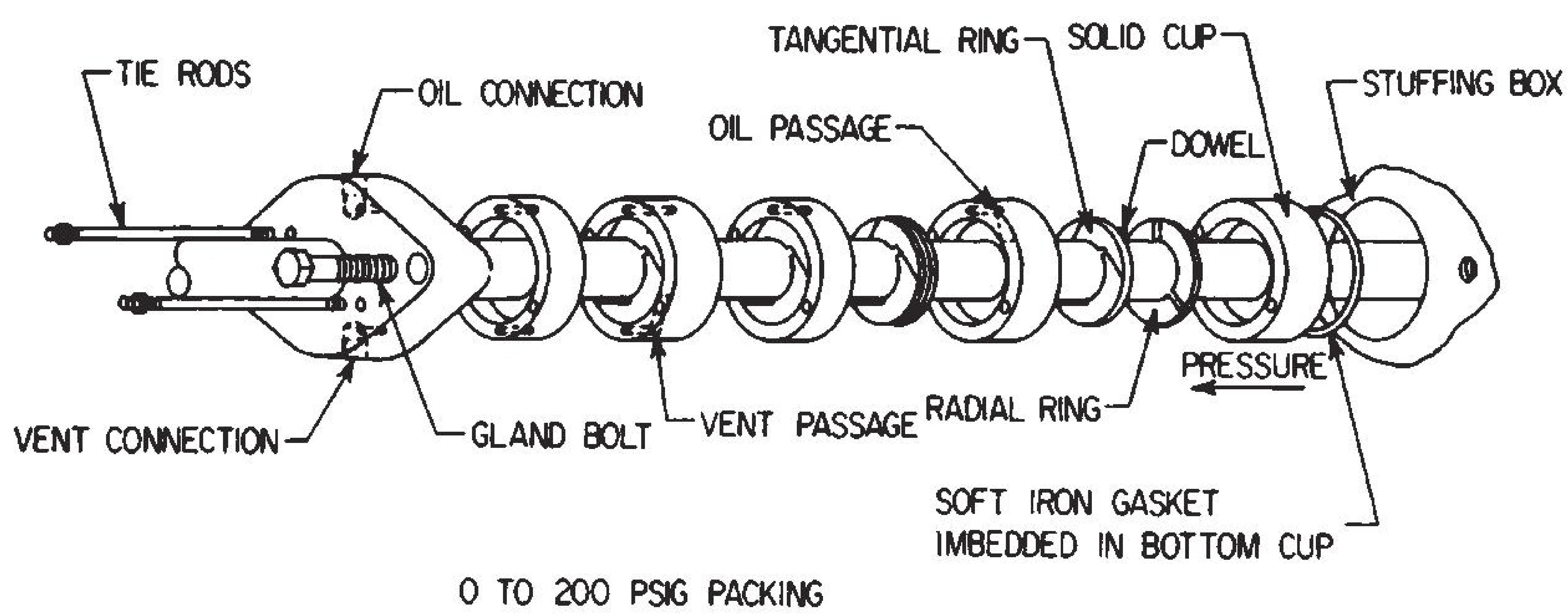


FIGURE 10.9 Cylinder-head packing.

TABLE 10.1 Service Check Chart, Mechanical Parts

1. Low oil pressure
- ***a.*** Low oil level
- ***b.*** Plugged oil-pump suction strainer
- ***c.*** Leaks in suction or pressure lines
- ***d.*** Worn-out bearings
- ***e.*** Defective oil pump
- ***f.*** Dirt in oil-filter check valve
- ***g.*** Broken oil-filter-check-valve spring
- ***h.*** Oil-pressure-bypass leaks

2. High oil pressure
- ***a.*** Plugged oil-pressure lines
- ***b.*** Defective oil-filter mechanism
- ***c.*** Excessive spring tension on filter check valves
- ***d.*** Excessive spring tension on oil-pressure adjusting mechanism

3. Incorrect delivery of mechanical lubricator
- ***a.*** Dirty or gummed pumps
- ***b.*** Broken spring in check valve at cylinder
- ***c.*** Leak in lines or sight feed
- ***d.*** Low oil level
- ***e.*** Plugged vent in lubricator reservoir

4. Overheated cylinder
- ***a.*** Insufficient cooling water
- ***b.*** Scored piston or cylinder
- ***c.*** Broken valves or valve springs
- ***d.*** Excessive carbon deposits
- ***e.*** Packing too tight
- ***f.*** Insufficient lubrication
- ***g.*** Corroded or clogged cylinder water passages

5. Water in cylinders
- ***a.*** Leaking head gaskets
- ***b.*** Cracked cylinder or head
- ***c.*** Condensate caused by too much cooling water or inoperative trap

6. High intercooler pressure
- ***a.*** Broken or leaking high-pressure valves
- ***b.*** Defective gage
- ***c.*** Defective or leaking valve-seat gaskets

7. Low intercooler pressure
- ***a.*** Broken or leaking low-pressure valves
- ***b.*** Leak in intercooler
- ***c.*** Piston-rod-packing leaking

8. Knocks
- ***a.*** Excessive carbon deposits
- ***b.*** Scored piston or cylinder
- ***c.*** Defective lubricator
- ***d.*** Foreign material in cylinder
- ***e.*** Piston hitting cylinder head
- ***f.*** Loose piston or piston pin
- ***g.*** Burned-out or worn rod bearings
- ***h.*** Loose main bearings
- ***i.*** Scored crosshead or crosshead guides
- ***j.*** Loose valve set screws

(*Continued*)

TABLE 10.1 Service Check Chart, Mechanical Parts (*Continued*)

9. Scored cylinder, liner, or piston
- ***a.*** Foreign material
- ***b.*** Dirty or inefficient air filter
- ***c.*** Lack of lubrication
- ***d.*** Too much and too cold cooling water causing excess condensate and washing out lubrication
- ***e.*** Excessive heat
- ***f.*** Plugged water jackets

10. Broken valves and springs
- ***a.*** Too much condensation, causing rust
- ***b.*** Carbon deposits
- ***c.*** Foreign materials not removed by air filter
- ***d.*** Incorrect assembly
- ***e.*** Acid condition prevailing at location of suction air inlet

11. Control trouble
- ***a.*** Suction-valve unloader stuck open or closed
- ***b.*** Pressure switch defective
- ***c.*** Solenoid burned out
- ***d.*** Foreign material in three-way valves
- ***e.*** Excessive vibration of control
- ***f.*** Voltage drop or loss of power
- ***g.*** Plugged air line or strainer
- ***h.*** Incorrect voltage or cycle

12. Incorrect operation of suction-valve unloaders
- ***a.*** Leaks in unloader line
- ***b.*** Foreign material in guides or seats
- ***c.*** Worn plungers
- ***d.*** Leaking or ruptured diaphragms and O-rings
- ***e.*** Broken springs
- ***f.*** Manual shutoff partly closed
- ***g.*** Wrong pressure-switch settings

Note: Remember to read the instruction book carefully and to keep it and the parts list in an accessible place so that when information to make adjustments and repair is needed, shutdown time can be held to a minimum.

CHAPTER 11
VALVES

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This chapter discusses the most common designs of industrial valves. The various designs of valves are developed from one idea—to place a disk over a seat opening in such a way that the resulting closure is tight. This one idea branches out into several basic designs of valves, including globe, check, gate, ball, and butterfly valves. Cross sections of globe, check, and gate valves are shown in Fig. 11.1. Cross sections of ball and butterfly valves are shown later in Figs. 11.9 and 11.10, respectively. Each design places the disk over the seat in a different manner.

Valves are usually made of one of three different metals, for the following reasons:

Bronze, for temperatures up to 550°F. Bronze is corrosion-resistant to a large majority of fluids, and it is easy to cast and machine. Bronze valves are usually made in sizes 3 and smaller.

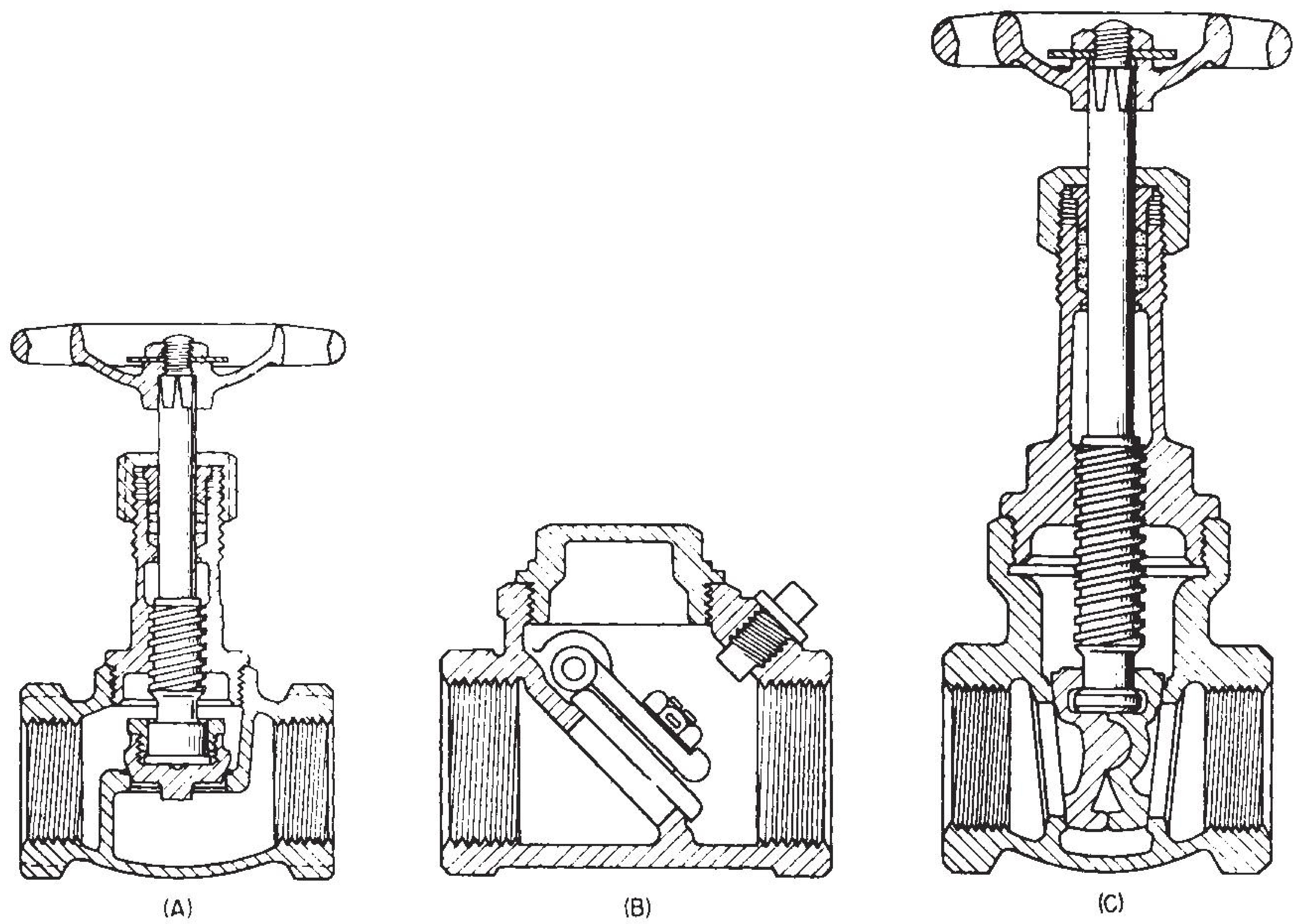


FIGURE 11.1 Three basic designs. *A.* Globe. *B.* Check *C.* Gate.

Cast iron, for temperatures up to 450°F. Cast iron is cheaper than bronze, hence the cost of cast-iron valves larger than size 2 is decidedly reduced. Cast-iron valves usually have either a bronze or an all-iron trim. Valves with a bronze trim are called "I.B.B.M." (iron-body, bronze-mounted) valves, while valves with an iron trim are called "A.I." (all-iron) valves. All-iron valves are used for solutions that attack bronze but not iron, such as caustic soda and concentrated sulfuric acid.

Cast steel, for temperatures up to 1100°F. Steel is stronger at high temperatures than bronze or cast iron.

In addition to the different basic designs of valves and the three basic metals, valves are available in different pressure temperature ratings and with different end connections. Pressure ratings will be explained in the next section, and the most common pressure ratings and end connections for each of the basic valve designs and materials will be given in the section on that particular valve.

WHAT IS A VALVE?

In order to understand the operation, application, and maintenance of valves, it might be well if the definition of a valve were considered. For the purposes of this discussion, a valve is a mechanical device usually used in connection with a pressure-containing vessel to completely stop or regulate flow.

As a *mechanical device,* a valve should be selected to do the job expected of it and should be properly installed. It will then give long service before it starts to leak or wear out. After installation, and periodically during service, a valve should be checked to ensure that it has the necessary seat tightness.

When wear or leakage shows up, it will require some maintenance to restore the valve to its original efficiency. General maintenance methods for typical valves are described in this chapter, but the valve manufacturer's literature should be consulted for specific procedures. Another helpful reference is MSS-SP-92, "MSS Valve Users Guide."[1] Wear occurs more frequently in globe or check valves, and features are built into these valves to facilitate maintenance or renewal of parts. The seat of all globe valves is directly opposite the top opening of the body, making it easy to get at the seat for inspection or repair. Gate valves are installed where they are not operated very often, and hence do not wear out quickly, and they do not as a rule have the maintenance features of globe and check valves.

Mechanical devices should be operated occasionally. Valves which are placed in lines and then forgotten may become hard to operate. This is especially so in hot-water lines, hard-water lines, or any other lines in which there is a tendency to deposit scale or solids. Valves actually have been known to scale up or coke up so badly over a period of years that they had to be disassembled and cleaned before they were usable.

The statement that a valve is used *in connection with a pressure-containing vessel* deserves consideration because the pressures and temperatures at which a valve of a given size may be used depend upon wall thickness and material of the pressure-containing parts. For the purposes of indicating the pressure temperature ratings of valves, the American National Standards Institute (ANSI) has established pressure class numbers (or classes for short). The pressure class number corresponds to the former steam pressure (SP) or primary pressure rating of the valve. Since the tensile and yield strength of valve materials is higher at room temperature than at the temperature of steam, the rating of a given class of value is higher at room temperature than at the temperature of steam. The pressure rating of the valve at ambient temperature (0 to 150°F for bronze and iron and 0 to 100°F for steel) is called the cold-working pressure (CWP) rating of the valve. The CWP rating of the valve corresponds to the former water-oil-gas (WOG) or secondary pressure rating of the valve. The CWP is about two times the pressure class number for Classes 300 and below, and about 2.4 times the pressure class number for Classes 350 and higher, depending upon material and size.

The statement that a valve is used *to completely stop or regulate flow* deserves consideration, as it indicates when a globe valve or when a gate valve is to be used. A globe valve is used to regulate

[1]Available from the Manufacturer's Standardization Society of the Valve and Fitting Industry, Inc., 5203 Leesburg Pike, Suite 502, Falls Church, VA 22041.

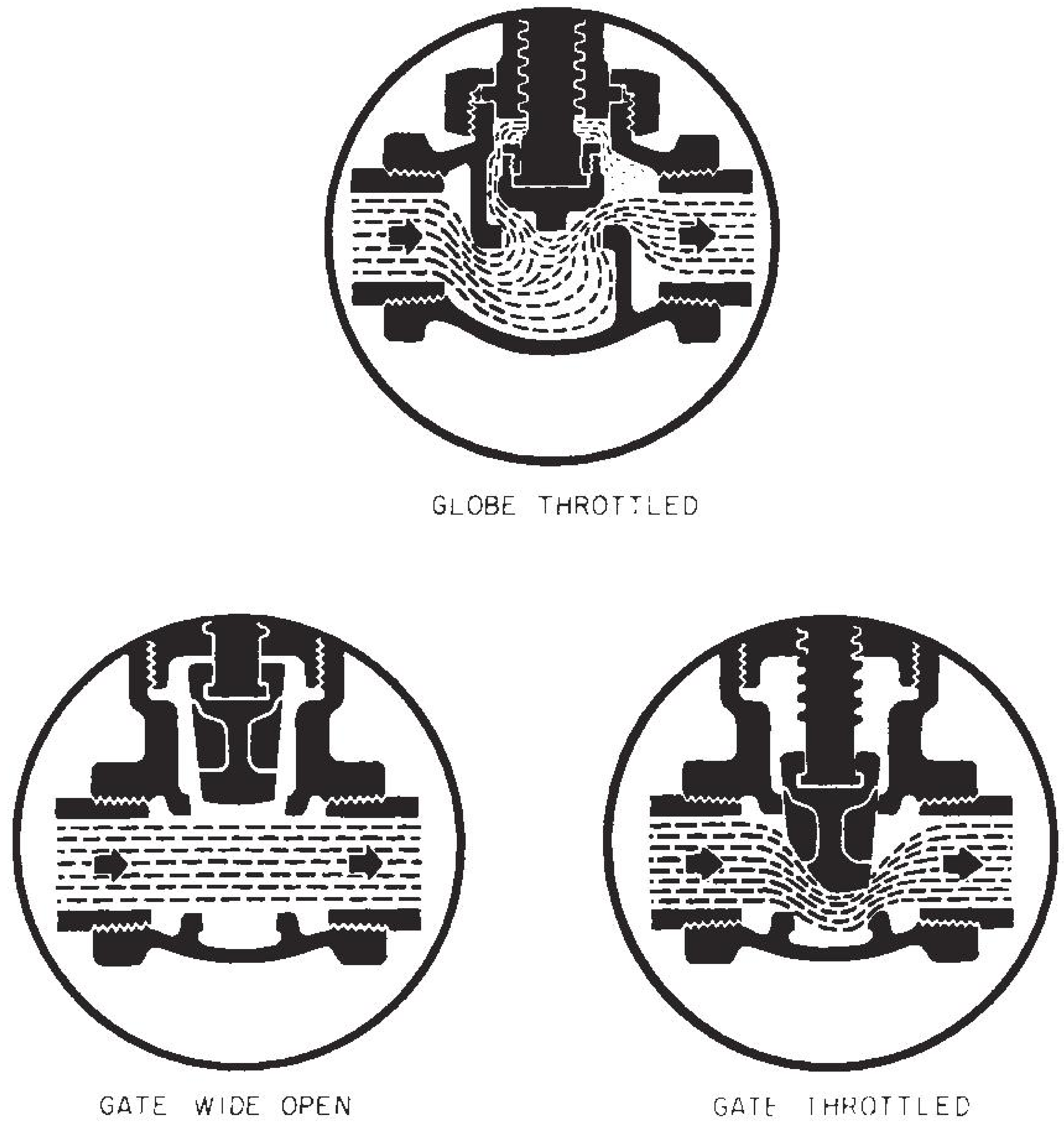


FIGURE 11.2 Diagrams of flow condition. These diagrams illustrate the answer to the question as to where you use globe valves and where gate valves. Globe valves are used to throttle flow. Note that the flow is around the entire periphery of the disk, giving even wear. Globe valves are easily repaired or reground. Gate valves are used when you want unobstructed flow and little line loss. The illustrations show the uneven wear when gate valves are misused for throttling.

flow, and a gate valve should be used where the service requires the valve to be in full open or closed position. The flow through a throttled globe valve is distributed uniformly around the entire periphery of the disk, giving even and less rapid wear. The flow through a throttled gate valve is concentrated at the bottom of the wedge, giving uneven and more rapid wear. This is illustrated in Fig. 11.2.

Also owing to the construction of the valve, a globe valve is recommended when the valve is to be operated frequently. The disk in a globe valve touches the seat only at the instant of closing. In a gate valve, the wedge travels over the full face of the seat and consequently sliding wear will develop.

When a globe valve in these services finally wears, the globe valve is easier to repair than a gate valve.

Frequently, engineering specifications will state: "Globe valves shall be used on throttling service or where the valve is to be opened and closed frequently. Gate valves shall be used for full-flow conditions or where the valve is normally in an open or closed condition."

When an ordinary globe valve is used in severe throttling service, rapid wear of the seat and the disk can result. For tight closing it is sometimes better to use two valves on the line, one for throttling and one that is either full open or closed.

BRONZE VALVES

Bronze globe, check, and gate valves are available in Classes 125, 150, 200, 250, 300, and 350. They come with integral bronze seats and bronze disks and stems, and with nickel alloy and stainless steel seat rings and disks and bronze stems. Bronze valves are normally made with threaded or solder ends in sizes ¼ to 2.

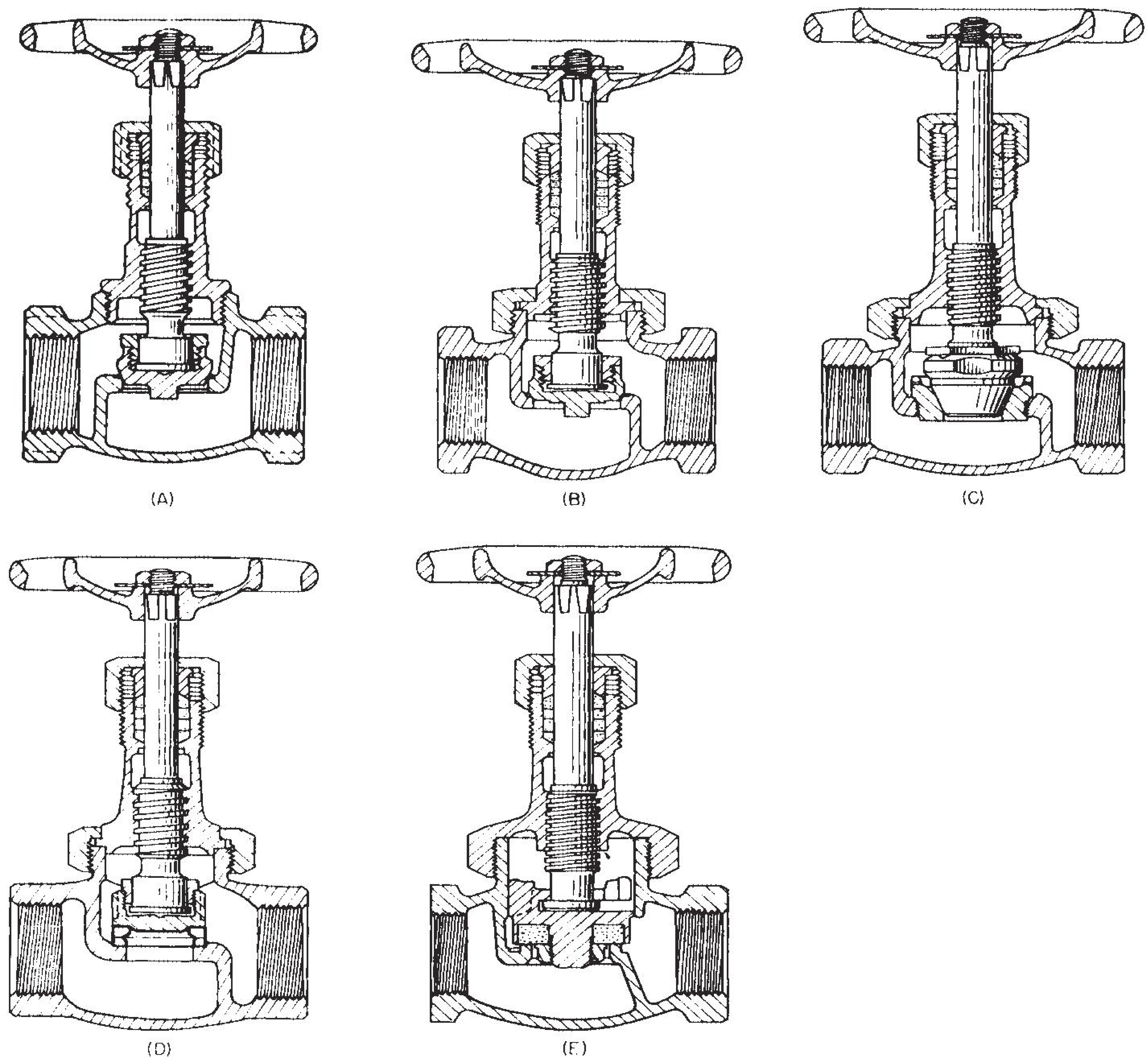


FIGURE 11.3 Bronze globe valves. *A*. Threaded-bonnet globe. *B*. Union-bonnet regrinding globe. *C*. Renewo globe, plug type. *D*. Flat-seat globe, 600 Brinell. *E*. Nonmetallic-disk globe.

Bronze Globe Valves

Globe valves are of various designs, some of which are illustrated in Fig. 11.3. A discussion of typical globe valves and their maintenance follows.

Threaded-Bonnet Globe Valve (Fig. 11.3*A*)**.** It is designed for use on inexpensive installations where the valve is not used frequently. Contractors often use it in low-pressure heating systems and on plumbing lines. *Maintenance* capability is very limited because the threaded body bonnet connection makes it difficult to regrind the valve. This valve can be repacked while under pressure by turning the valve stem full open. Caution should be taken if the valve is to be repacked under pressure as there is a possibility of dirt, scratches, or mars developing on the stem or bonnet back-seat in service, resulting in a poor seating condition. For this reason, repacking under pressure is not recommended unless absolutely necessary, and never for valves in hazardous services.

Union-Bonnet Regrinding Valve. It is made in two different pressure ratings, 200 and 350 lb SP. It was originally designed for easy *maintenance*, without removal from the line. It can be reground and repaired. A small metal plate clamped between the end of the stem and the disk is used to prevent the disk from swiveling on the stem during the regrinding operation. The handwheel is used

for the tool; the bonnet lip is used for a guide in the body neck. Valve-reseating tools, which can be obtained in sets from any industrial-supply house, can be used to dress up the seat if disk and seat are too worn for regrinding. The hardness of the bronze seat and disk is 85 Brinell, which makes the use of the reseating tool possible. The valve can be repacked under pressure. (See Fig. 11.3*B*.)

Plug-Type Renewo Globe Valve. It is designed for severe throttling, drain, drip, water-column blowdown, and other services demanding high resistance to destructive action on seat bearings. *Maintenance* consists of renewing the seat and disk. Because of their hardness, these valves present quite a regrinding problem by hand. They are cone-shaped and are always installed in pairs. Their hardness is 500 Brinell. The valve can be repacked under pressure. (See Fig. 11.3*C*.)

600-Brinell Flat-Seat Globe Valve. There is practically no *maintenance* on the valve except for repacking because the flat seats are extremely hard and erosion resistant. (See Fig. 11.3*D*.) This valve is useful on steam, air, water, oil, gas, or other media, and will be equally tight in all these services.

Nonmetallic-Disk Globe Valve (Fig. 11.3*E*)**.** It is also known as a composition disk globe valve. This valve is one of the most popular globe valves because of its easy maintenance. *Maintenance* consists principally of renewing the disk as it wears out. It can be easily removed from the disk holder, and a new disk inserted. Two kinds of disks are available: one for steam and hot water and one for cold water, air, gas, oxygen, solvents, acids, and alkalies. If the raised seat becomes worn or grooved, it can be resurfaced with a reseating tool. Brinell hardness is 85. The valve can be repacked under pressure.

Bronze Check Valves

Check valves are the guardians against backflow in a pipeline. They are entirely automatic in action and are of various designs, some of which are illustrated in Fig. 11.4. They fall into two general groups, commonly known as “swing-check” valves and “lift-check” valves. A swing-check valve is usually used where full flow is desired. A lift-check valve is usually used on air or gases or when the operation of the check valve is quite frequent. A discussion of typical check valves and their maintenance follows.

Nonmetallic-Disk Lift-Check Valve (Fig. 11.4*A*)**.** The seat is rounded to give line contact on the disk, in comparison with the flat seat in a similar globe valve. The line contact is necessary, as there is usually little pressure to hold the disk to its seat. Sometimes a spring is inserted to act on the disk to increase this pressure. *Maintenance* of this valve consists of renewing the disk when necessary, smoothing the upper and lower disk guides when necessary, and removing grooves or worn places from the seat with a reseating tool.

Swing-Check Valve (Fig. 11.4*B*)**.** Swing-check valves are probably the most popular and the most used of all check valves. They can be installed in horizontal or in vertical lines with flow up. *Maintenance* consists of regrinding the disk to its seat by applying a screwdriver to the slot in the top of the disk and using a grinding compound. If carrier pin, side plugs, or disk carrier become worn, they can be easily and inexpensively replaced with new parts. These maintenance suggestions apply also to I.B.B.M. swing-check valves.

Regrinding Lift-Check Valve. This is a good check valve in that it has both upper and lower guides to guide the disk to its seat. All parts are renewable except the seat, which is integral. *Maintenance* of this valve consists of regrinding the disk to its seat by means of the screwdriver slot in the disk stem. Brinell hardness of seat and disk is 85 Brinell, allowing for the use of a valve-reseating tool. (See Fig. 11.4*C*.)

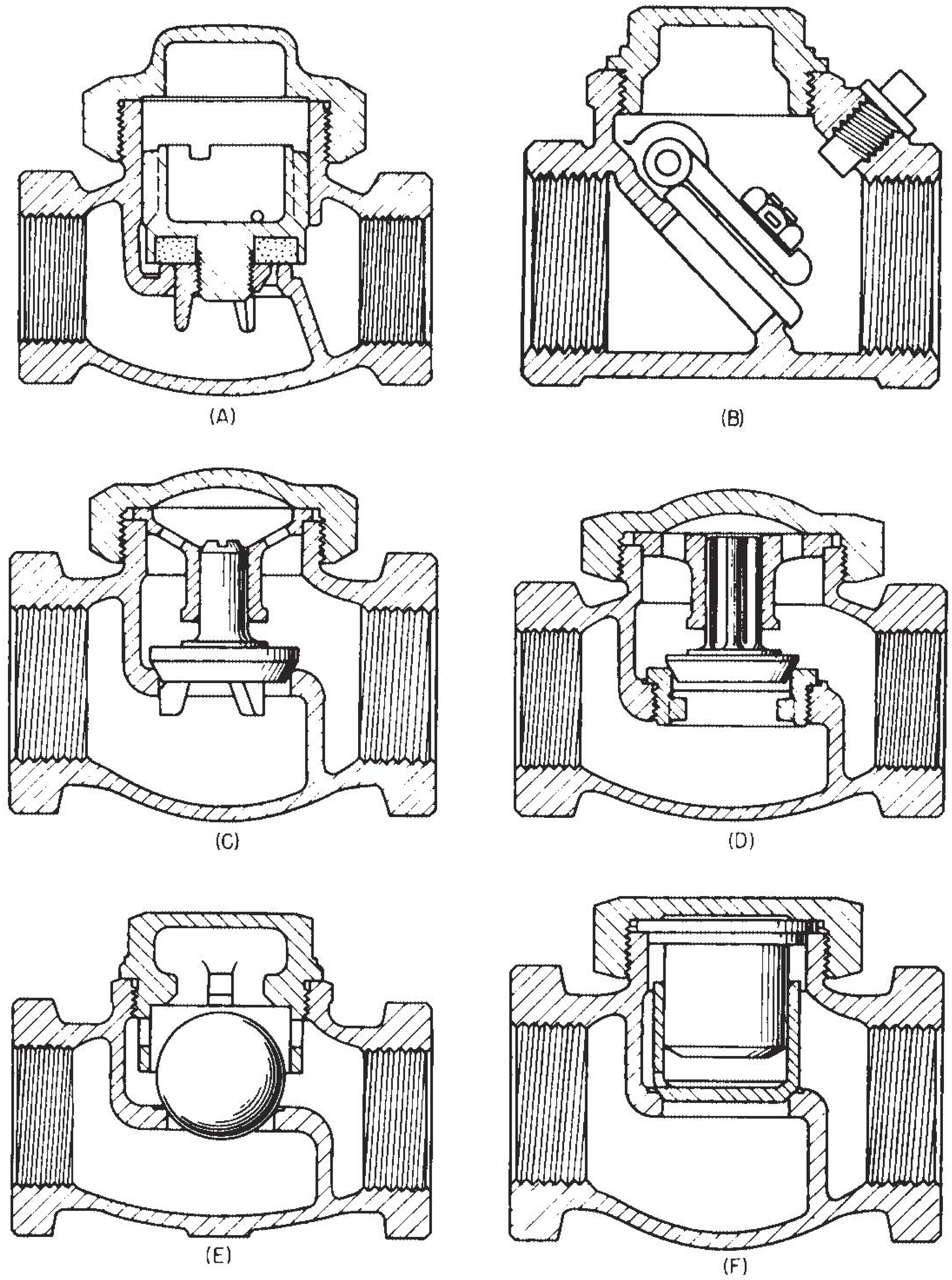


FIGURE 11.4 Bronze check valves. *A.* N-M-D lift-check nonmetallic disk. *B.* Swing-check regrinding seat. *C.* Lift-check regrinding seat. *D.* Lift-check renewable seat. *E.* Ball check. *F.* Air-compressor check.

Renewo Lift-Check Valve. This check valve has renewable seats and disks of nickel alloy, Brinell hardness 185. The disk has only an upper guide; so it is not as accurate in seating as the regrinding lift-check valve. *Maintenance* consists of regrinding the disk to its seat or replacing it if too badly worn. (See Fig. 11.4*D.*)

Ball-Check Valve. Some people consider the ball-check valve the ideal check valve. But in the opinion of valve men, it should be used only on viscous or heavy liquids, such as varnish, molasses, muddy water, or liquids containing solids. Any of these would clog the mechanism of the other check valves. There is little *maintenance* on a ball-check valve, as there is no means of holding the ball for regrinding the seats. This ball must be as perfect a sphere as possible, and the seat must be perfectly round. (See Fig. 11.4*E.*)

Air-Compressor Check Valve. This check valve (see Fig. 11.4*F*) is especially designed for this service, which is the hardest known for check valves. An ordinary check valve opens and closes once

at each revolution of the compressor. Swing-check valves have been known to disintegrate in 5 min with this frequency of operation.

The air-compressor check valve shown incorporates a stainless-steel disk operating over a bronze disk guide in such a manner as to provide an air cushion to reduce pounding. This air cushion damps the movement of the disk so the disk opens when the compressor starts, and stays open until the compressor shuts down. The disk then eases itself to its seat and is held tight by back pressure. Carry-over oil in the air line improves the efficiency of the air cushion. Maintenance consists of removing the cap and oiling the parts inside of the disk if there is insufficient carry-over oil, replacing renewable parts and reseating the seat. It is best to install the valve as far from the compressor as possible. This will reduce pulsations acting on the disk.

Bronze Gate Valves

Gate valves are by far the most popular and the most frequently used of the three types of valves—globe, check, and gate. As their correct installation calls for usage where they are opened and closed only infrequently, they last a long time and do not require much maintenance. If a gate valve is operated over ten times a day every day, it will quickly wear out and a globe valve should be substituted for it. The wear will be found on the downstream faces of the seat and the wedge because the line pressure forces all the wear on these surfaces. The upstream faces frequently will be found to be in good condition. Very often, worn gate valves can be reversed 180° and they will be as good as new. Gate valves should be installed with the stem vertical if at all possible. Installation with the stem in a horizontal position is permissible, but is not as good.

The three gate valves shown at the top of Fig. 11.5 are kindred valves, in that all are the same body. The first one shown is the most popular valve known—a standard double-wedge bronze gate valve with rising stem.

Double-Wedge Rising-Stem Gate Valve (Fig. 11.5*A*)**.** The double wedges are of ball-and-socket construction or of the uniball type where each single wedge has a half socket and half ball. They readily adjust themselves to the taper seats, ensuring a tight valve. It should be evident that line strains could distort the angle of the taper seats in the bronze body. Slight distortion will not affect the tightness of double-wedge valves. The rising stem indicates whether the valve is open or closed. There is little *maintenance* to be done on a gate valve. It should be taken apart occasionally over the years and cleaned, especially valves on hot-water lines. The valve can be repacked under pressure by opening the valve to the limit of the stem travel.

Solid-Wedge Rising-Stem Gate Valve (Fig. 11.5*B*)**.** This valve is used on heavy liquids such as molasses or varnish or any other liquids that would tend to make ball-and-socket wedges inoperative. The solid-wedge bronze gate valve is not as tight on thin liquids or gases as is the double-wedge valve. *Maintenance* is the same as for the double-wedge rising-stem gate valve.

Single-Wedge Nonrising-Stem Gate Valve (Fig. 11.5*C*)**.** This valve is popular with contractors and is used extensively on marine service. The nonrising-stem feature allows the valve to be used in cramped spaces where overhead construction interferes with the operating of a rising-stem gate valve.

Small Outside Screw-and-Yoke Rising-Stem Gate Valve (Fig. 11.5*D*)**.** Note that this valve has a union bonnet and a single wedge. It is required by code on the lines leading to the top and bottom connections on the water column of a boiler. It must be locked in open position by means of a chain and padlock so that there is no wear on it. It is an emergency valve; hence its *maintenance* consists of test operation and inspection to see that it is in working condition. It can become limed up rather quickly, especially the valve on the lower line, as this line contains hot water.

Renewable-Wedge-and-Seat Bronze Gate Valve (Fig. 11.5*E*)**.** The valve shown has a nonrising stem and renewable nickel-alloy seats and wedge. It is popular in the chemical industry, as the seats

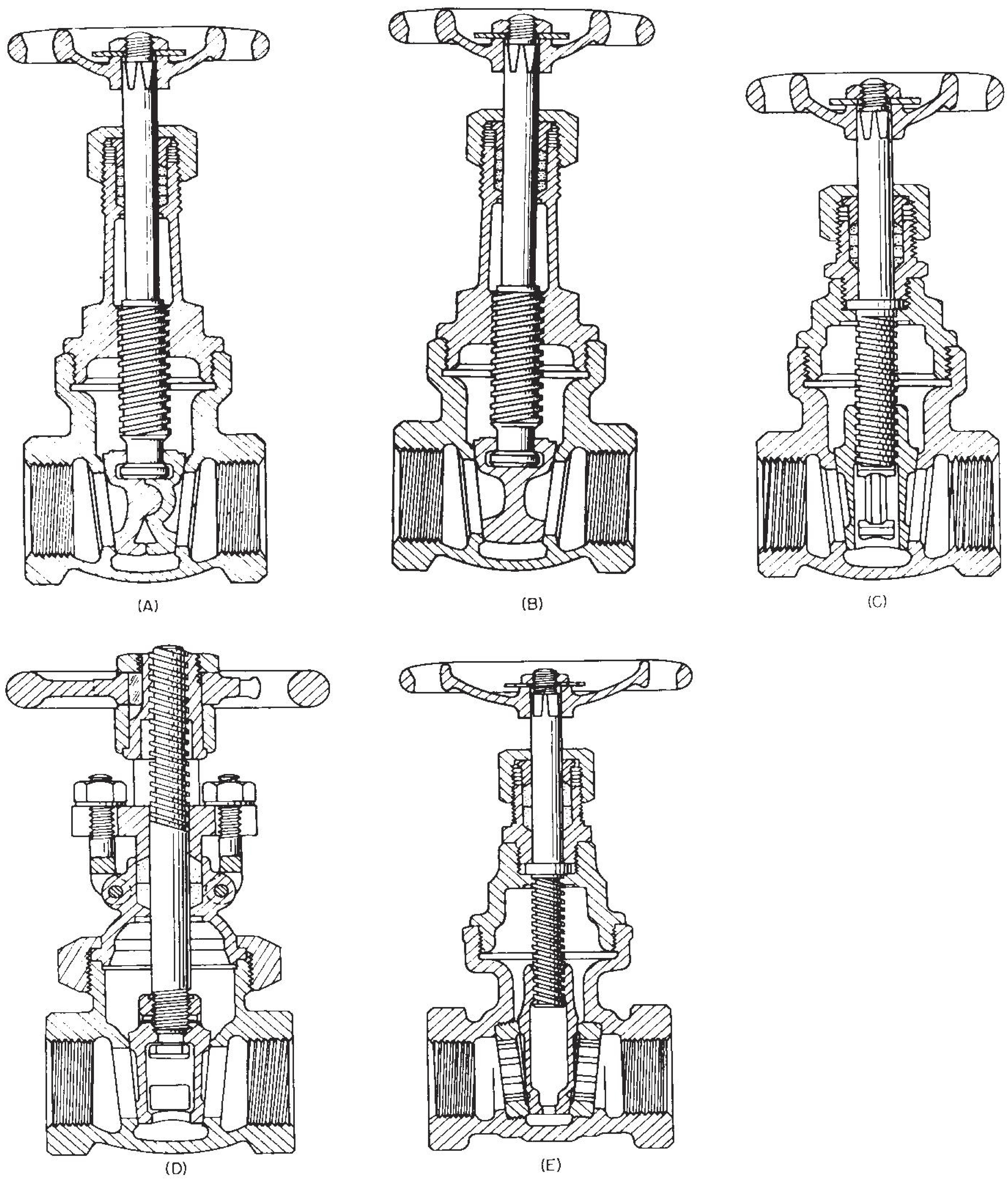


FIGURE 11.5 Bronze gate valves. *A.* Rising-stem, double-wedge disk. *B.* Solid-wedge disk, rising stem. *C.* Nonrising-stem, single-wedge disk. *D.* Outside screw and yoke union bonnet, single-wedge disk. *E.* Nonrising stem, single-wedge disk, renewable seat rings.

and disks can be renewed as they become unsatisfactory in operation. The valve should be removed from the line during this repair. Gate valves are not like globe valves, which can be repaired while on the line.

IRON VALVES

Cast-iron globe, check, and gate valves are available in Classes 125 and 250. They come with bronze seat rings, disk facings, and stem (iron-body bronze-mounted, or I.B.B.M.) and with integral iron

seats or steel seat rings, iron disk facings, and steel stems (all-iron, or A.I.). Iron valves are normally made with threaded and grooved ends in sizes 2 to 6 and with flanged ends for sizes 2 to 30.

Iron Globe and Check Valves

Figure 11.6 shows iron-body globe and angle valves, respectively. The application and *maintenance* of these is similar to bronze globe valves.

Iron-body swing and lift-check valves are also available.

Iron Gate Valves

The size 6 iron-body bronze-mounted (I.B.B.M.), outside screw-and-yoke (O.S. & Y.), flanged-end (F.E.) gate valve shown in Fig. 11.7 is the most popular in size and design of all the large valves.

The *maintenance* of the valve follows commonsense lines. The stem threads should be kept lubricated and free from dirt. When the valve is wide open for a long period of time, the exposed stem threads should be protected by a light sheet-iron tube placed over them.

To repack the valve, move the swing gland bolts out of the way. The gland is raised and rests on the ledges provided for that purpose. The stuffing box is then accessible for renewal of the packing. Each ring of new packing should be compressed by the gland before another ring is added. Splits in split ring packing should be staggered. The valve can be repacked under pressure.

Should the downstream seats become scored, the upstream seats will frequently be found to be in good condition. Reverse the valve 180°, and the valve will be as good as new.

Should it become necessary to replace the seat rings, remove the valve from the line and prepare a correct-size pipe with square notches to fit the lugs in the seat rings. As the pipe with lugs is twisted (by means of a bar), tap the body smartly with a hammer to help loosen the ring. Clean all threads

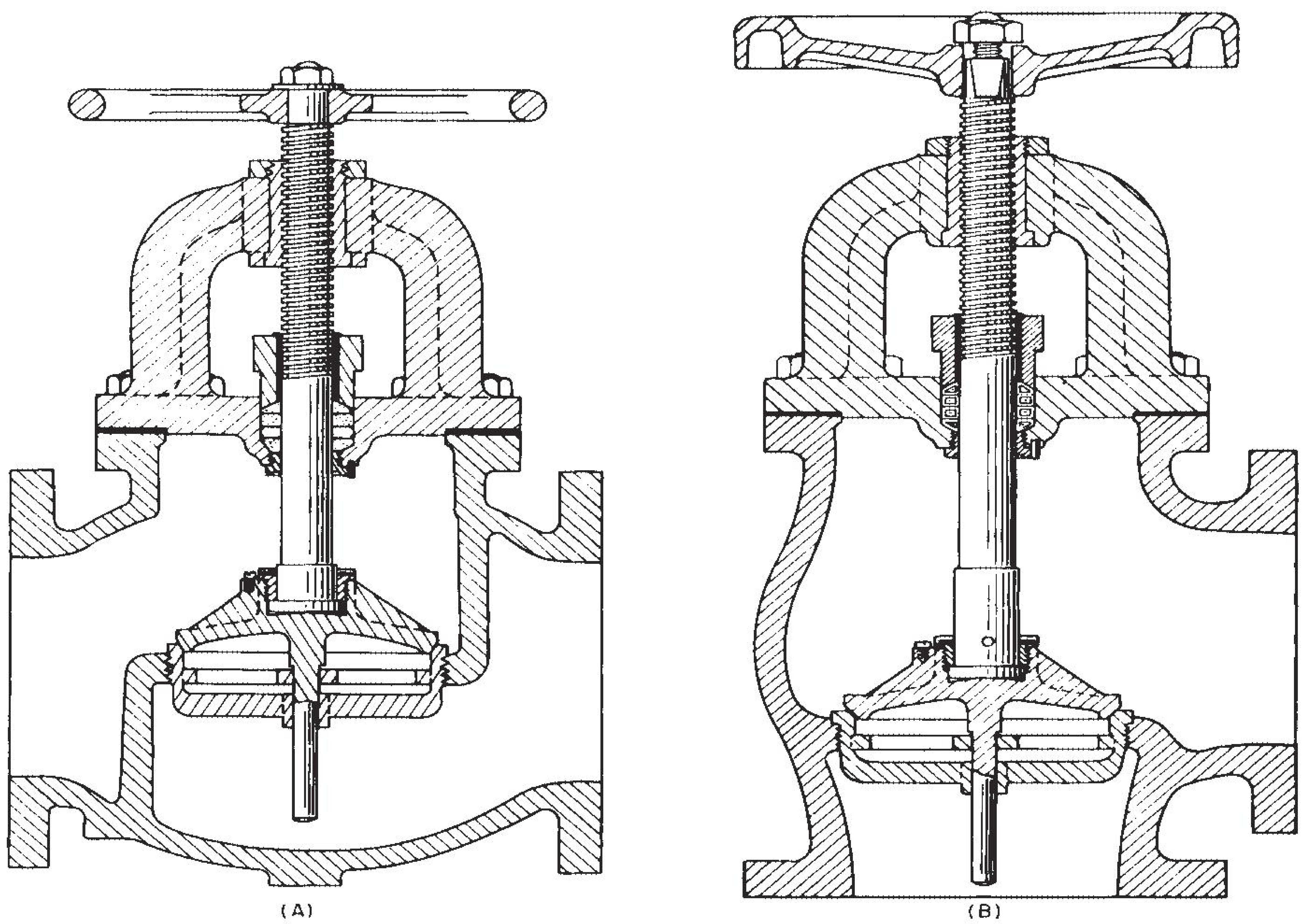


FIGURE 11.6 Variations of basic valves as to design. *A.* Globe. *B.* Angle.

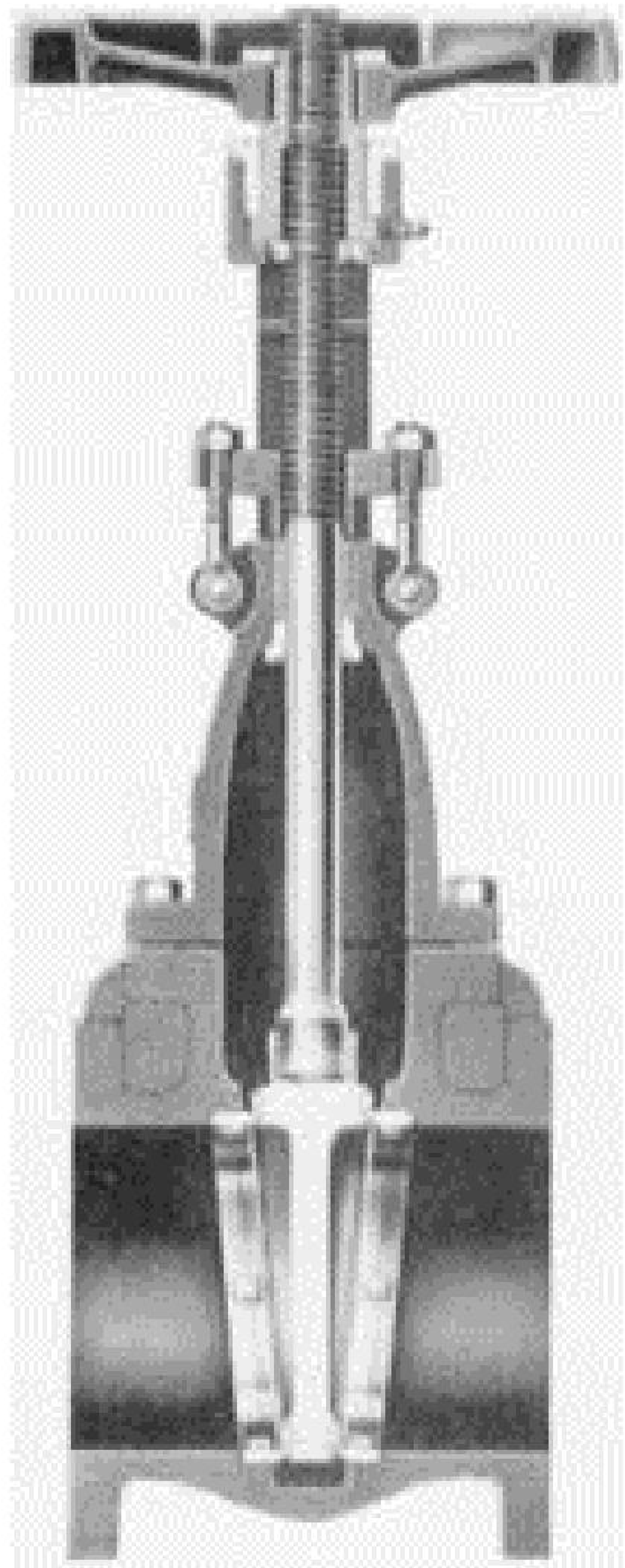


FIGURE 11.7 Iron-body gate valve.

and seating surfaces with a wire brush before installing new rings. Graphite or pipe dope can be used. A new disk should be installed with new rings. It may have to be lapped in. Retighten body bonnet bolts uniformly using a crisscross pattern and at least three passes.

STEEL VALVES

Forged-steel globe, check, and gate valves are available in Classes 150, 300, 600, 800, 1500, 2500, and 4500. They are normally made with flanged ends in sizes ½ to 2 for Classes 150, 300, and 600, and threaded and socket weld ends in sizes ¼ to 2 for Classes 800, 1500, 2500, and 4500.

Cast-steel bolted-bonnet globe, check, and gate valves are available in Classes 150, 300, and 600. They come with 410 stainless-steel seat and wedge facing and stems for normal service, and hard-faced seat and wedge facing and 410 stainless-steel stems for severe service. They are also available in bronze, monel, and 316 trims for special services. They may have screwed-in or welded-in seat rings. They are normally made with flanged or butt-weld ends in sizes 2 to 30.

Cast-steel breech-lock and pressure-seal globe, check and gate valves are made in Classes 600, 900, 1500, 2500, and 4500. They come with hardfaced welded-in seat rings, hardfaced wedges, and 410 stainless-steel stems. They are normally made with butt-weld ends in sizes 3 to 24. These valves are used in high-pressure, high-temperature steam and water service in power plants. Figure 11.8 shows breech-lock gate valves with three different means of actuation.

The *maintenance* of steel valves consists of periodic checks for shell and seat tightness, periodic lubrication and exercising, and repacking when needed. Repair of steel valves is more difficult than for other valves because of their harder materials, integral lay-on or welded-in seat rings, larger size

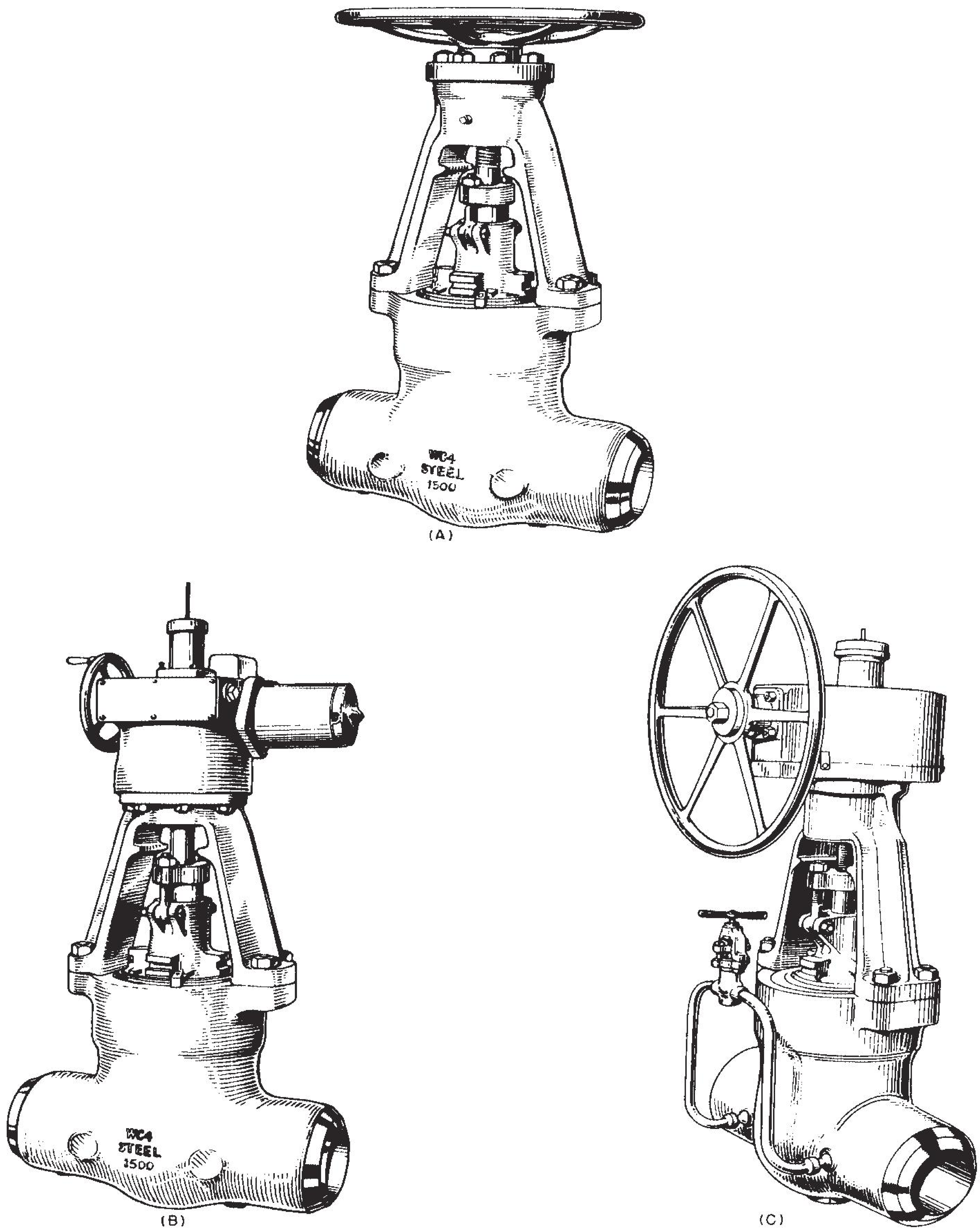


FIGURE 11.8 Variations of basic valves as to methods of operation. *A*. Breech lock, hand-operated. *B*. Breech lock, motor-operated. *C*. Breech lock with bypass, bevel-gear operated.

and weight, and use of welded pipe connections. Globe and gate valves must be removed from the line and machine tools used to renew the seats, or else special power equipment can be used to renew the seats in line. Valves with welded-in seat rings must be removed from the line for replacement of seat rings, which is best done by the manufacturer or a valve reconditioning shop. Gate valve wedges can be hand lapped on a surface plate with lapping compound if their condition is not too bad.

BALL VALVES

The ball valve has a spherical closure element (or ball) with a port through it (equivalent to the disk in a conventional valve) mounted on renewable seats of Teflon. To open the valve, the ball is rotated so that the through-port lines up with the seat openings. When the valve is closed, line pressure forces the ball against the downstream seat, in an action similar to that of a gate valve. The ball valve is more compact, tighter sealing, quicker operating, and more easily maintained than conventional gate or globe valves, and has better flow characteristics. It is used in many services where conventional valves would have been used formerly. (See Fig. 11.9.)

Ball valves come in sizes ¼ through 24 and larger. Pressure ratings are 250 to 3000 psig cold-working pressure, and the temperature is limited by the sealing elastomer. They are available in bronze, and carbon and alloy steel shell materials with Teflon and other sealing elastomers. They are made with threaded, solder, flanged, socket-welding, and butt-welding end connections, depending upon shell material and size. Design variations include top-entry (Fig. 11.9*B*), end-entry (Fig. 11.9*C*), and three-piece (Fig. 11.9*D*) designs. The top-entry valve can be repaired without removing it from the line. The end-entry design is the most economical and has a simpler or no body joint. The three-piece design offers a built-in union connection.

The maintenance of ball valves is simple. Should the valve show leakage, the seats, seals, and balls can be replaced easily, thus giving a new valve.

BUTTERFLY VALVES

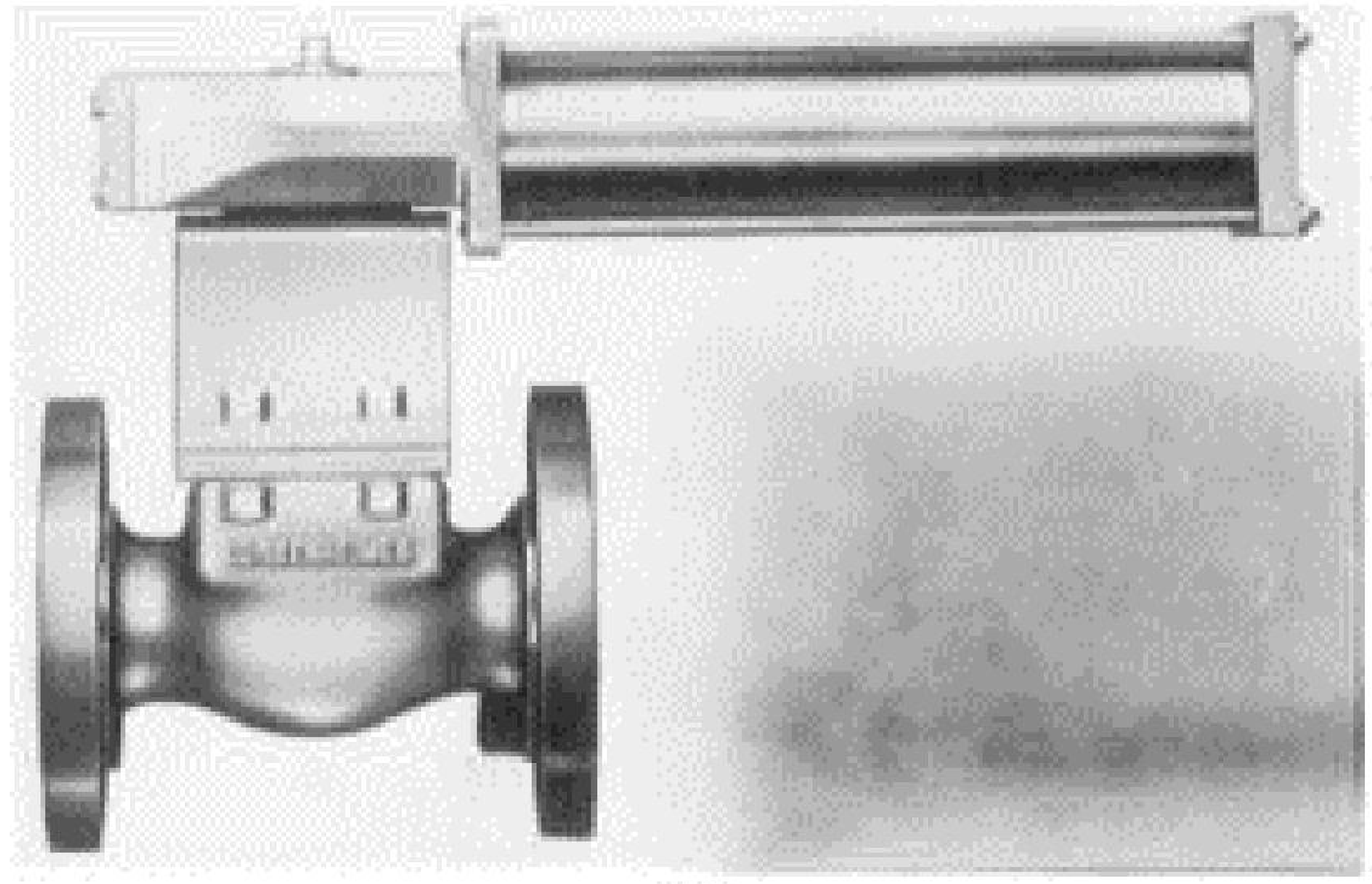
The butterfly valve has a disk-shaped closure element which rotates about a central shaft (or stem). (See Fig. 11.10.) To close the valve, the disk is rotated so that its face is across the pipe, blocking flow. Depending upon the type of valve, the valve seat may consist of a bonded resilient liner, a mechanically fastened resilient liner, an insert-type reinforced resilient liner (Fig. 11.10*C*), a mechanically fastened resilient seal (Fig. 11.10*D*), or an integral metal seat with an O-ring inserted around the edge of the disk. Butterfly valves are much more easily repaired than conventional valves. Butterfly valves find applications in most categories where valves are used. They are used extensively in gathering lines in oil-country installations. They are also used throughout industry for all fluids with which they are compatible.

Butterfly valves are commonly available in sizes 1½ through 24. Cold working pressure ratings are 150, 200, 275, and 720 psig, with temperature limited by seat and sealing elastomers. (Butterfly valves with the latter two ratings are often called high-performance butterfly valves.) Body materials are aluminum, cast and ductile irons, and carbon and stainless steels; seat and seal elastomers are Buna N, EPT, viton, and Teflon. There are threaded- and grooved-end body styles for sizes 1½ through 6, and wafer and lug styles for sizes 1½ through 24. The wafer and lug types have very narrow bodies and are installed by bolting between standard ANSI flanges. On-off detent-type levers; 10-position lift or squeeze-type levers; manual worm gear actuators; and electric, pneumatic, and hydraulic actuators are available. Actuators are recommended for size 8 and larger butterfly valves.

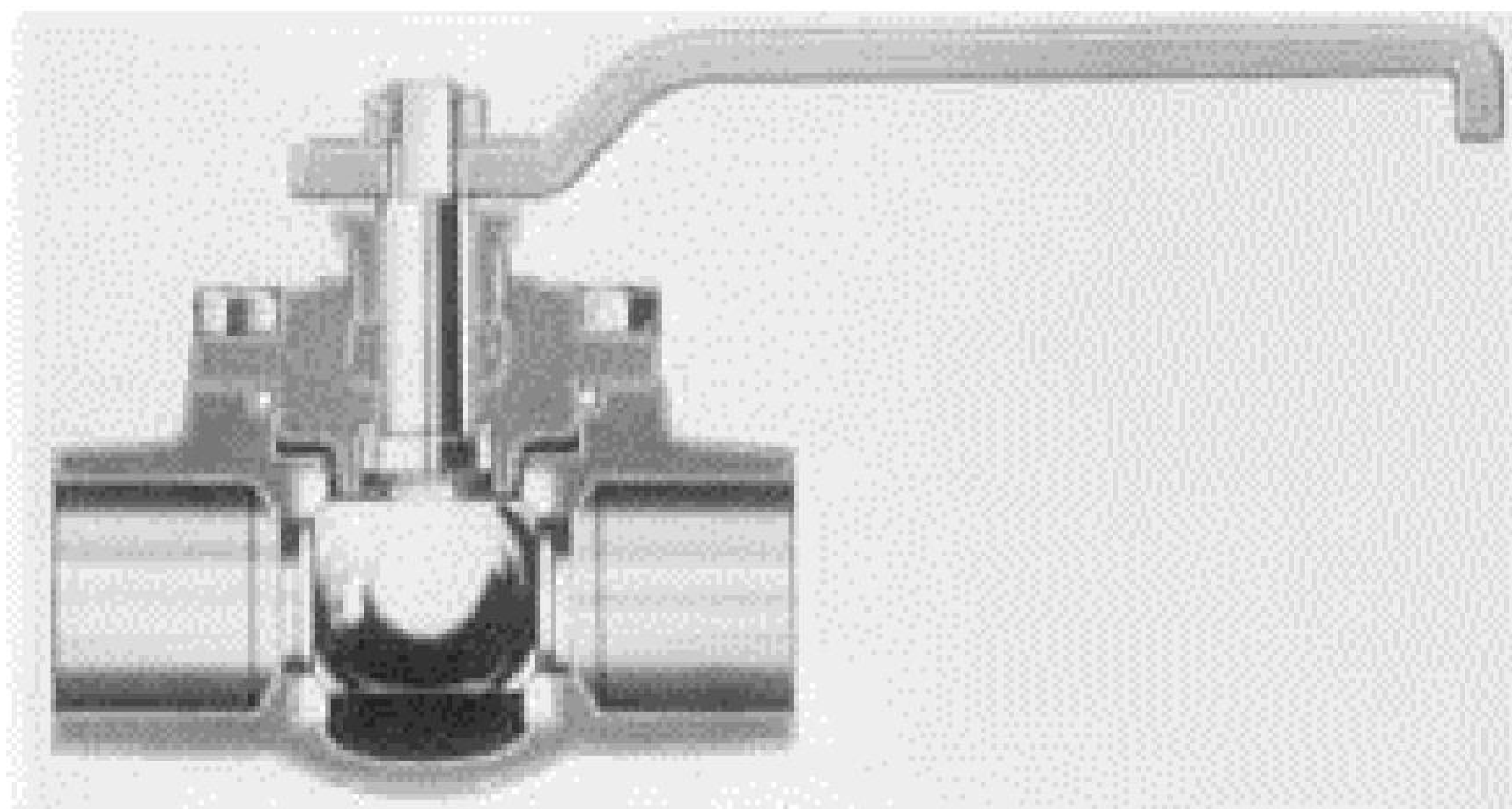
The maintenance of butterfly valves is quite simple. No lubrication is necessary until such time as stem O-rings need replacement. Disks and stems are readily replaceable, as are disk O-rings and replaceable resilient seats. Replacement of the bonded resilient liner is a factory job, but usually when that type of seat deteriorates it is more economical to purchase a complete new valve, since the cost of butterfly valves is low.

ORDERING SPARE PARTS

Whenever it is necessary or advisable to order spare or replacement parts, the first problem that arises is to identify the part, and the second problem is to identify the valve.



(*A*)



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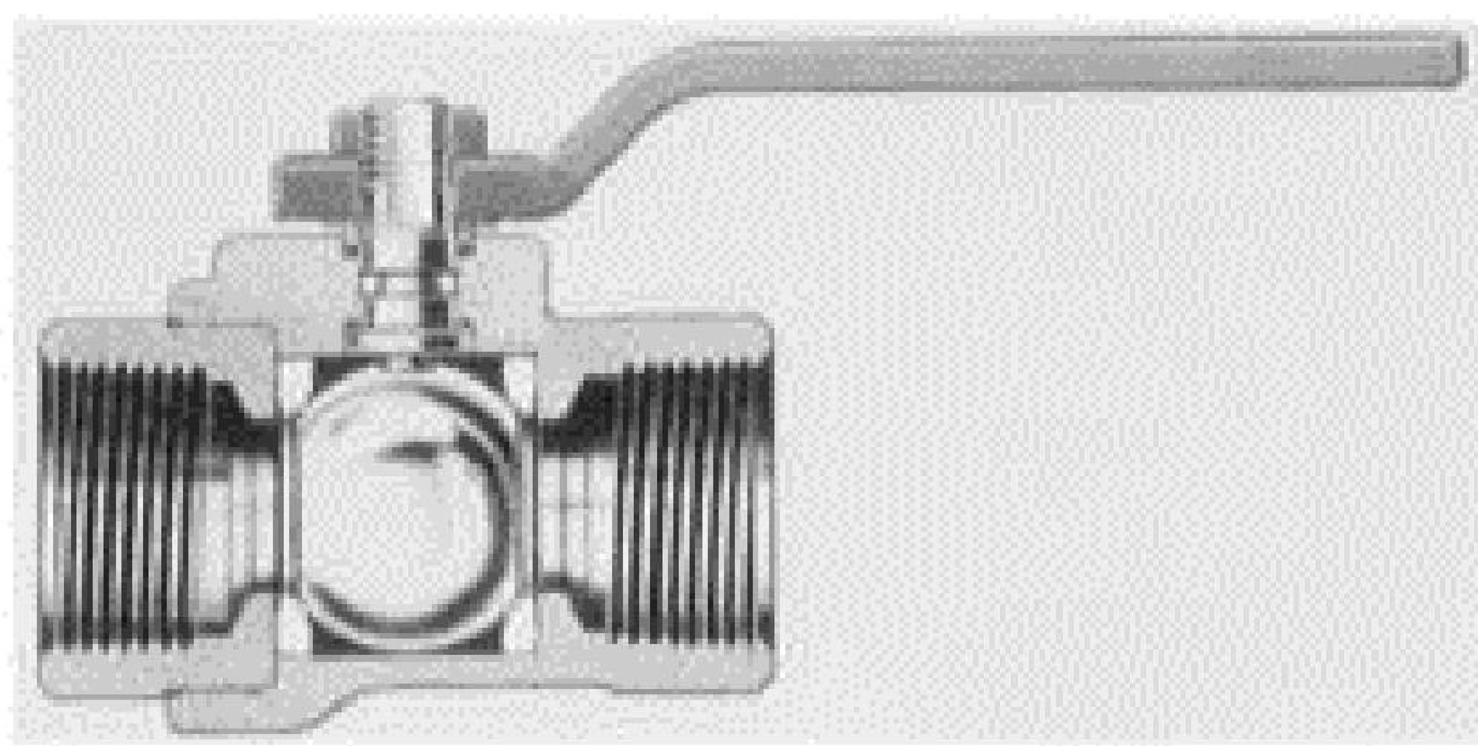

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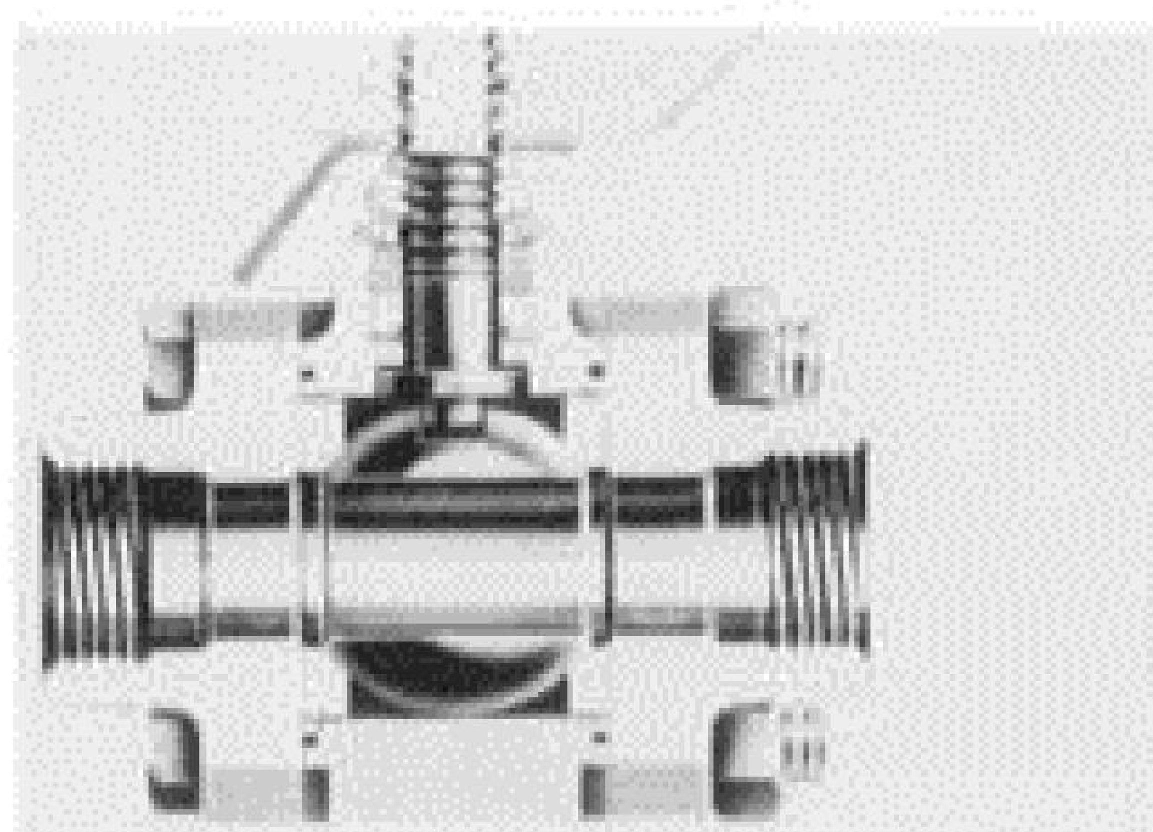

(*D*)

FIGURE 11.9 Variations of ball valves. *A*. Flange-end ball valve with pneumatic actuator. *B*. Top-entry ball valve. *C*. End-entry ball valve. *D*. Three-piece ball valve.

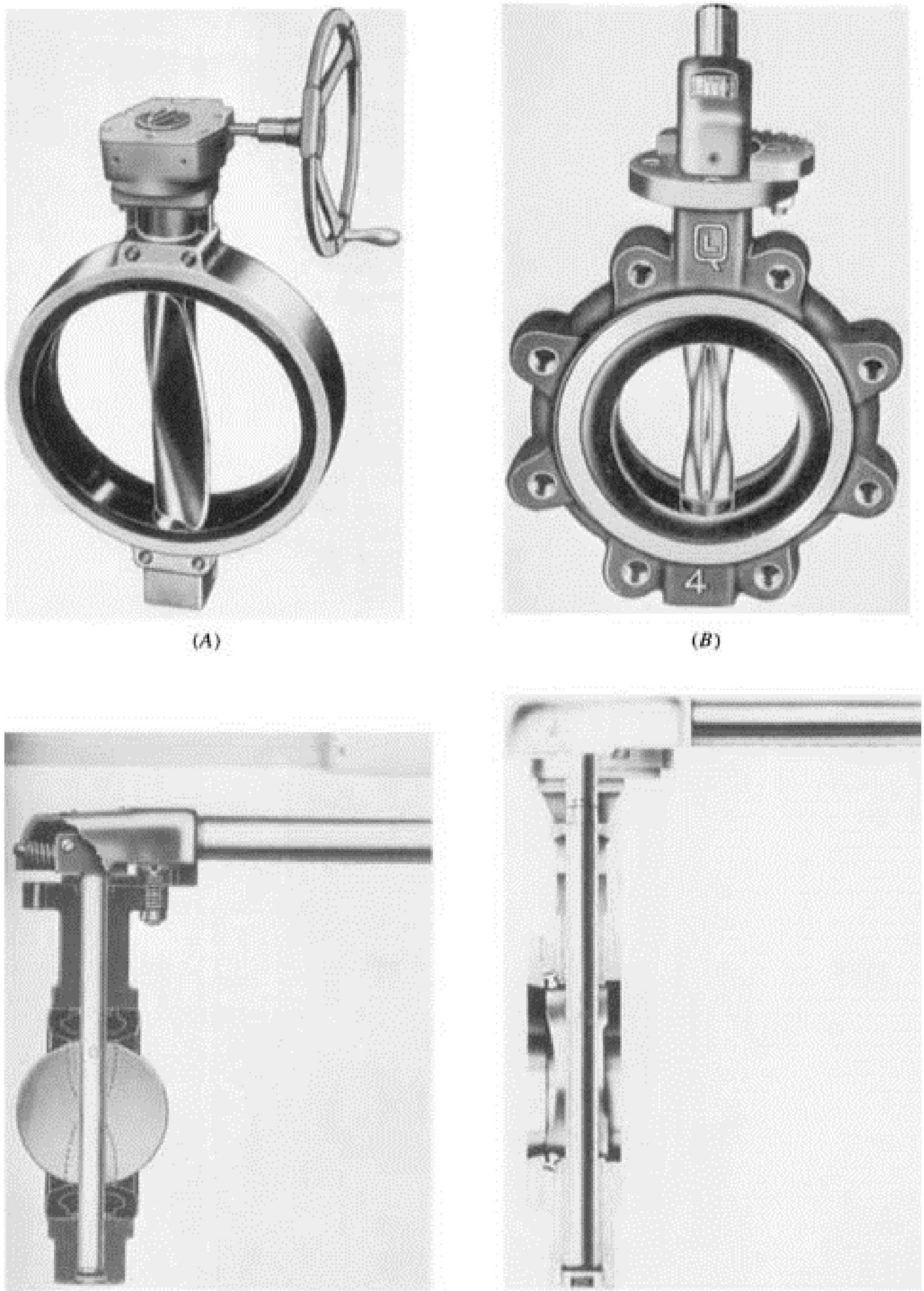

FIGURE 11.10 Variations of butterfly valves. *A*. Wafer type with gear operator. *B*. Lug type with replaceable seat. *C*. Wafer type with replaceable seat and lever operation. *D*. High-performance butterfly valve.

Identifying the Parts. It is well to specify the correct name of the part wanted; and for this reason, most valve catalogs carry illustrations naming the parts. This type of information is shown in Fig. 11.11 in the cross sections of the gate and globe valves, with the size and figure number being a necessary part of the information. If you have a valve catalog of one manufacturer and are ordering parts for another make of valve, specify the catalog and page number you are using to identify the parts. The names of each part of a valve are not identical with all the various valve manufacturers.

Most valve catalogs also show exploded views of their valves with each part illustrated. This makes it a little easier to identify the part desired. This type of information is shown in Fig. 11.12 for a swing-check valve.

Identifying the Valve. Most modern valves carry a nameplate, which makes it easy to identify the manufacturer and figure number. Nameplates originated with steel valves; but after the close of World War II, they were placed on bronze and iron valves, also. Unfortunately, many of the valves without nameplates have not worn out, and we are forced to identify the valve in some other way. If the valve is covered with insulation, it will be necessary to remove this to view the markings on the body.

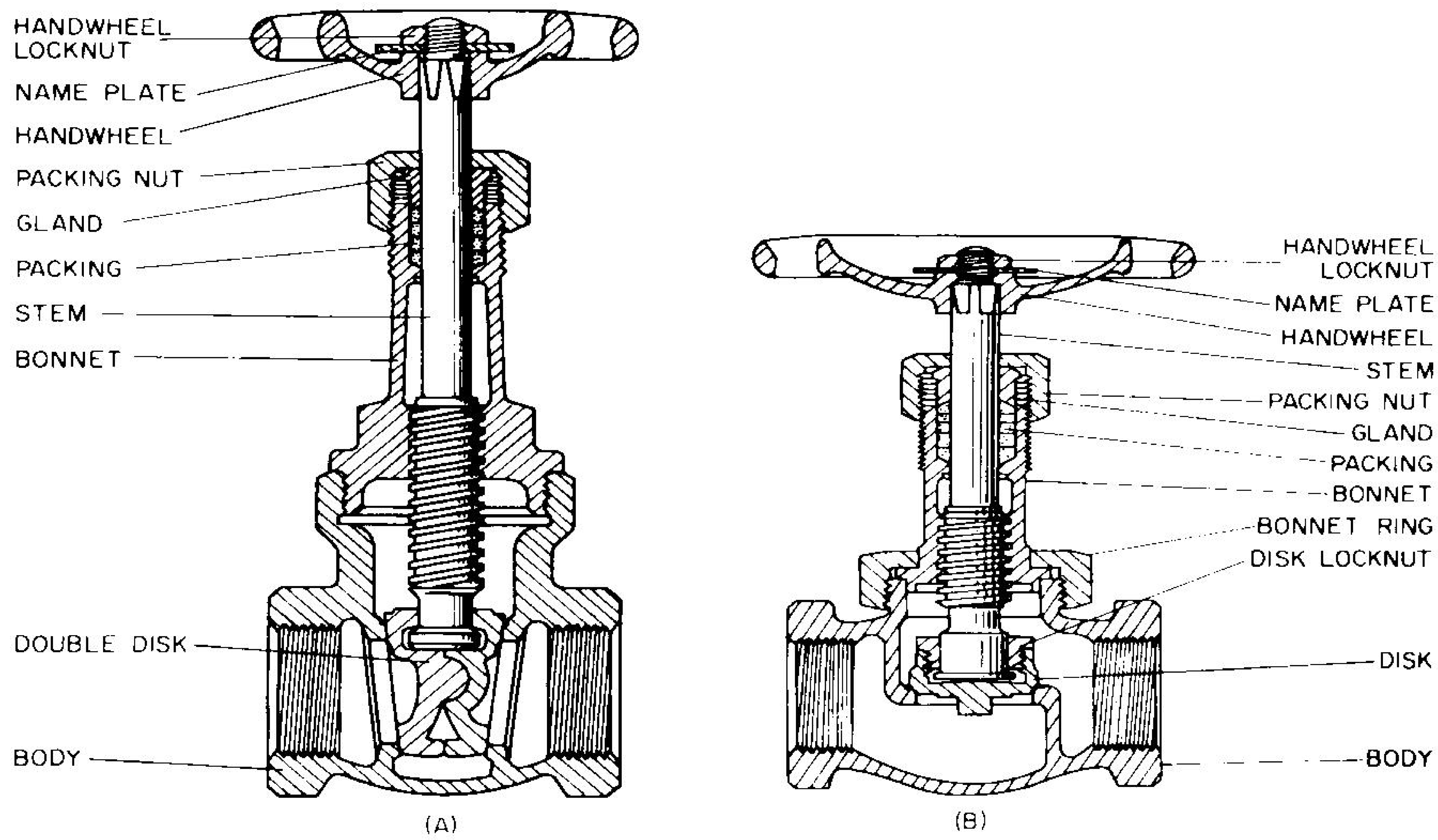

FIGURE 11.11 Parts identification. *A.* Rising-stem, double-wedge valve. *B.* Union-bonnet regrinding globe valve.

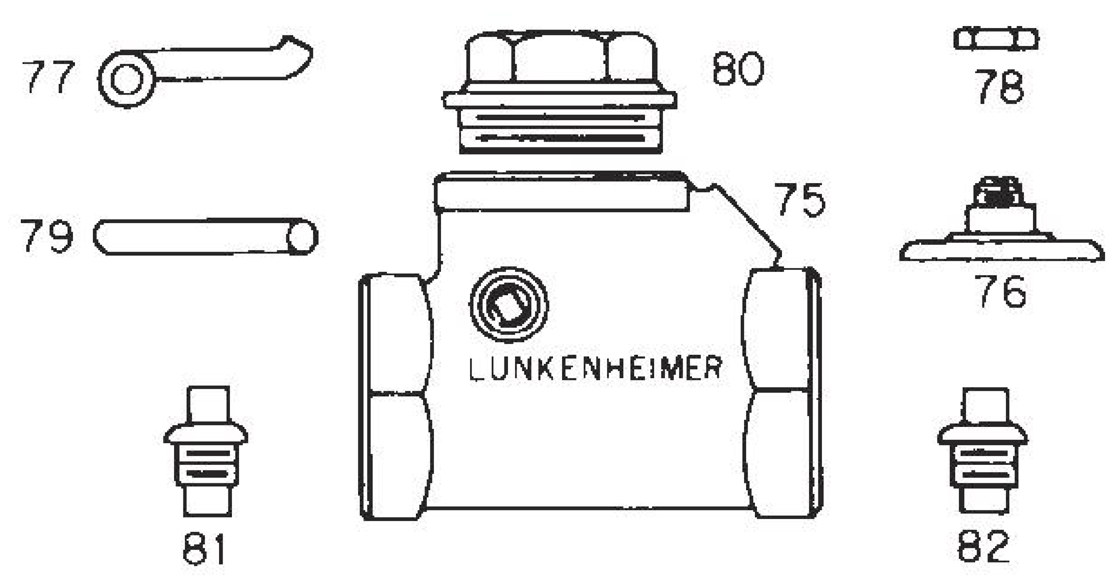

FIGURE 11.12 Ordering parts. Illustrated is bronze regrinding swing-check valve, Figs. 624 and 596 in maker's catalog. Orders should specify size and figure number of valve for which part is intended, quantity and name of repair part, and reference to part key number and catalog page or drawing number, if available.

There you will probably find the name of the manufacturer, the size of the valve, and the steam pressure, and other ratings. This information may be on both sides of the body. The figure number is rarely found on the body casting, as the same casting is often used to make valves carrying various figure numbers.

Copy all this information and include it with the order, which should specify the metal the valve is made of, a brief description together with the type of valve (globe, check, or gate), and the type of ends (screw or flange). In the case of flanged valves, the type of face (whether flat or raised face), the diameter of the flange, the diameter of the bolt circle, the size of the bolts or bolt holes, and finally the face-to-face dimension of the flanges should be given. If possible, give the approximate date of the installation of the valve. Valve designs and details and figure numbers change over the years, and all the above information suggested will be helpful, especially in the case of an old valve. Sometimes it may be necessary to call in a representative of the valve manufacturer to help identify an old valve.

Writing the Order. In writing the order, it is well to specify first the part wanted, and let the other information follow, thus:

> Disk only, Size 1½ Lunkenheimer
> Fig. 554Y, class 200 valve, screw ends.

Frequently repair-parts orders read, "One size 1½-in., Fig. 554Y, disk only." This will probably result in a complete valve being shipped, unless some well-informed order checker scans the order.

RECOMMENDED PIPING PRACTICE

Clean the inside of the pipe before installing or repairing a valve. This will remove rust, scale, welding beads, and dirt, which could be carried into a valve and cause trouble. Do not remove flange or thread protectors from the valve until ready for installation. When threading pipe, do not cut the threads too long. Long threads allow the pipe to enter the valve too deeply and distort the seat or will hit the diaphragm. Apply pipe dope or Teflon tape to male threads only when making up a threaded joint. When installing a screw-end valve, do not employ enough force to distort the valve body. Use a crescent wrench or monkey wrench on the valve end that is being made up. Employ a pipe wrench on pipe only. Allow a new valve to warm up gradually. Packing glands are assembled hand-tight at the factory, and on installation should be tightened only enough to prevent leakage.

When installing a globe or angle valve to a pipeline, the direction of flow should be so that the pressure is under the disk except in steam, open-end, and drain lines where the pressure should be on top of the disk.

When installing a check valve in liquid lines, it is suggested that the following guidelines be adhered to:

1. Check valves should be installed as far as possible from the pump discharge.
2. Do not use a swing-check valve on reciprocating liquid pumps; always use a vertical-lift type.
3. In cases of water hammer, noise, or shock during closing of a swing-check valve, it is recommended that the check valve be changed to vertical-lift type, but increase the size by one so that the pressure drop will remain the same for the same flow. If noise is still excessive, a gas-over-liquid damper can be added to the line by inserting a tee and a vertical pipe as high as possible. The pipe should be two or three sizes larger than the supply pipe. An accumulator-type damper can be installed where shock is critical.

Finally, it is extremely important that a valve be closed tightly by hand only. Do not use a wrench or persuader. Dirt under the disk can usually be flushed out by operating the valve a number of times. A valve that is cracked open is subject to the most severe wire-drawing or throttling conditions possible, decreasing valve life and increasing maintenance.

CHAPTER 12
PUMPS: CENTRIFUGAL AND POSITIVE DISPLACEMENT

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Pumps are designed to transfer a specific volume of liquid at a particular pressure from a fixed source to a final destination in a process system. A pump's operating envelope is defined either by a hydraulic curve for centrifugal pumps or a pressure–volume (PV) diagram for positive-displacement pumps.

CENTRIFUGAL PUMPS

This section provides the basic knowledge needed to evaluate a centrifugal-pump application to determine its operating dynamics and to identify any forcing function that may contribute to chronic reliability problems, premature failures, or loss of process performance.

Centrifugal pumps are highly susceptible to variations in process parameters, such as suction pressure, specific gravity of the pumped liquid, back-pressure induced by control valves, and changes in demand volume. Therefore, the dominant reasons for centrifugal-pump failures are usually process-related.

Several factors dominate pump performance and reliability:

1. Internal configuration
2. Suction conditions
3. Total system head (or pressure)
4. Total dynamic head (or pressure)
5. Hydraulic curve
6. Brake horsepower
7. Installation
8. Operating methods

These factors must be understood and taken into consideration when evaluating any centrifugal-pump-related problem or event.

Internal Configuration

Centrifugal pumps are not all alike. Variations in the internal configuration occur in the impeller type and orientation. These variations have a direct impact on a pump's stability, useful life, and performance characteristics.

Impeller Type. There are a variety of impeller types used in centrifugal pumps. They range from simple radial flow, open designs to complex variable-pitch, high-volume enclosed designs. Each of these types is designed to perform a specific function and should be selected with care. In relatively small, general-purpose pumps, the impellers are normally designed to provide radial flow and the choices are limited to either and *enclosed* or an *open* design.

Enclosed impellers are cast with the vanes fully encased between two disks. This type of impeller is generally used for clean, solid-free liquids. It has a much higher efficiency than the open design.

Open impellers have only one disk and the opposite side of the vanes is open to the liquid. Because of its lower efficiency, this design is limited to applications where slurries or solids are an integral part of the liquid.

Impeller Orientation. In *single-stage* centrifugal pumps, impeller orientation is fixed and is not a factor in pump performance. However, it must be carefully considered in *multistage* pumps, which are available in two configurations: in-line and opposed. These configurations are illustrated in Fig. 12.1.

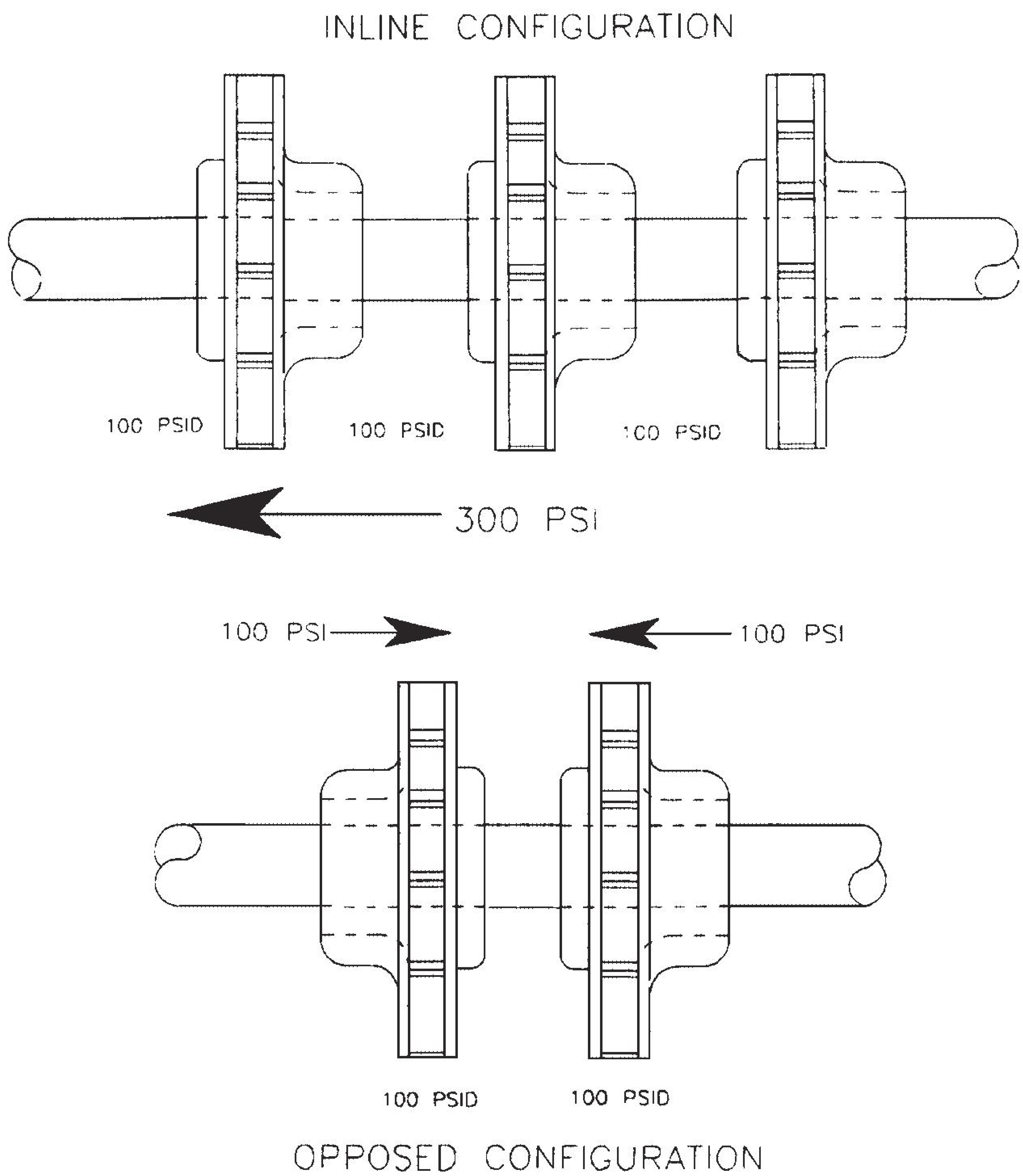

FIGURE 12.1 Impeller orientation of multistage centrifugal pumps.

In-line. In-line configurations have all impellers facing in the same direction. As a result, the total differential pressure between the discharge and inlet is axially applied to the rotating element toward the outboard bearing. Because of this configuration, in-line pumps are highly susceptible to changes in the operating envelope.

Because of the tremendous axial pressures that are created by the in-line design, these pumps must have a positive means of limiting end-play, or axial movement, of the rotating element. Normally, one of two methods is used to fix or limit axial movement:

1. A large thrust bearing is installed at the outboard end of the pump to restrict movement.

2. Discharge pressure is vented to a piston mounted on the outboard end of the shaft.

Method 1 relies on the holding strength of the thrust bearing to absorb energy generated by the pump's differential pressure. If the process is reasonably stable, this design approach is valid and should provide relatively trouble-free service life. However, this design cannot tolerate any radical or repeated variation in its operating envelope. Any change in the differential pressure or transient burst of energy generated by flow change will overload the thrust bearing, which may result in instantaneous failure.

Method 2 uses a bypass stream of pumped fluid at full discharge pressure to compensate for the axial load on the rotating element. While this design is more tolerant of process variations, it cannot compensate for repeated, instantaneous changes in demand, volume, or pressure.

Opposed. Multistage pumps that use opposed impellers are much more stable and can tolerate a broader range of process variables than those with an in-line configuration. In the opposed-impeller design, sets of impellers are mounted back to back on the shaft. As a result, the thrust or axial force generated by one of the pairs is canceled out by the other. This design approach virtually eliminates axial forces. As a result, the pump does not require a massive thrust bearing or balancing piston to fix the axial position of the shaft and rotating element.

Since the axial forces are balanced, this type of pump is much more tolerant of changes in flow and differential pressure than the in-line design. However, it is not immune to process instability or to the transient forces caused by frequent radical changes in the operating envelope.

Suction Conditions

Factors affecting suction conditions are the net positive suction head, suction volume, and entrained air or gas.

Net Positive Suction Head. Suction pressure, called net positive suction head (NPSH), is one of the major factors governing pump performance. The variables affecting suction head are shown in Fig. 12.2.

A centrifugal pump must have a minimum amount of consistent and constant positive pressure at the eye of its impeller. If this suction pressure is not available, the pump will be unable to transfer liquid. The suction supply can be open and below the pump's centerline, but the atmospheric pressure must be greater than the pressure required to lift the liquid to the impeller eye and to provide the minimum NPSH that is required for proper pump operation.

At sea level, atmospheric pressure exerts a pressure of 14.7 pounds per square inch (psi) on the surface of the supply liquid. This pressure minus vapor pressure, friction loss, velocity head, and static lift must be enough to provide the minimum NPSH requirements of the pump. These requirements vary with the volume of liquid transferred by the pump.

Most pump curves provide the minimum NPSH required for various flow conditions. This information, which is labeled $NPSH_R$, is generally presented as a rising curve located near the bottom of the hydraulic curve. The data are usually expressed in "feet of head" rather than psi.

To convert from psi to feet of head for water, multiply by 2.31. For example, 14.7 psi is 14.7 times 2.31, or 33.957 feet of head. To convert feet of head to psi, multiply the total feet of head by 0.4331.

Suction Volume. The pump's supply system must provide a consistent volume of single-phase liquid equal to or greater than the volume delivered by the pump. To accomplish this, the suction

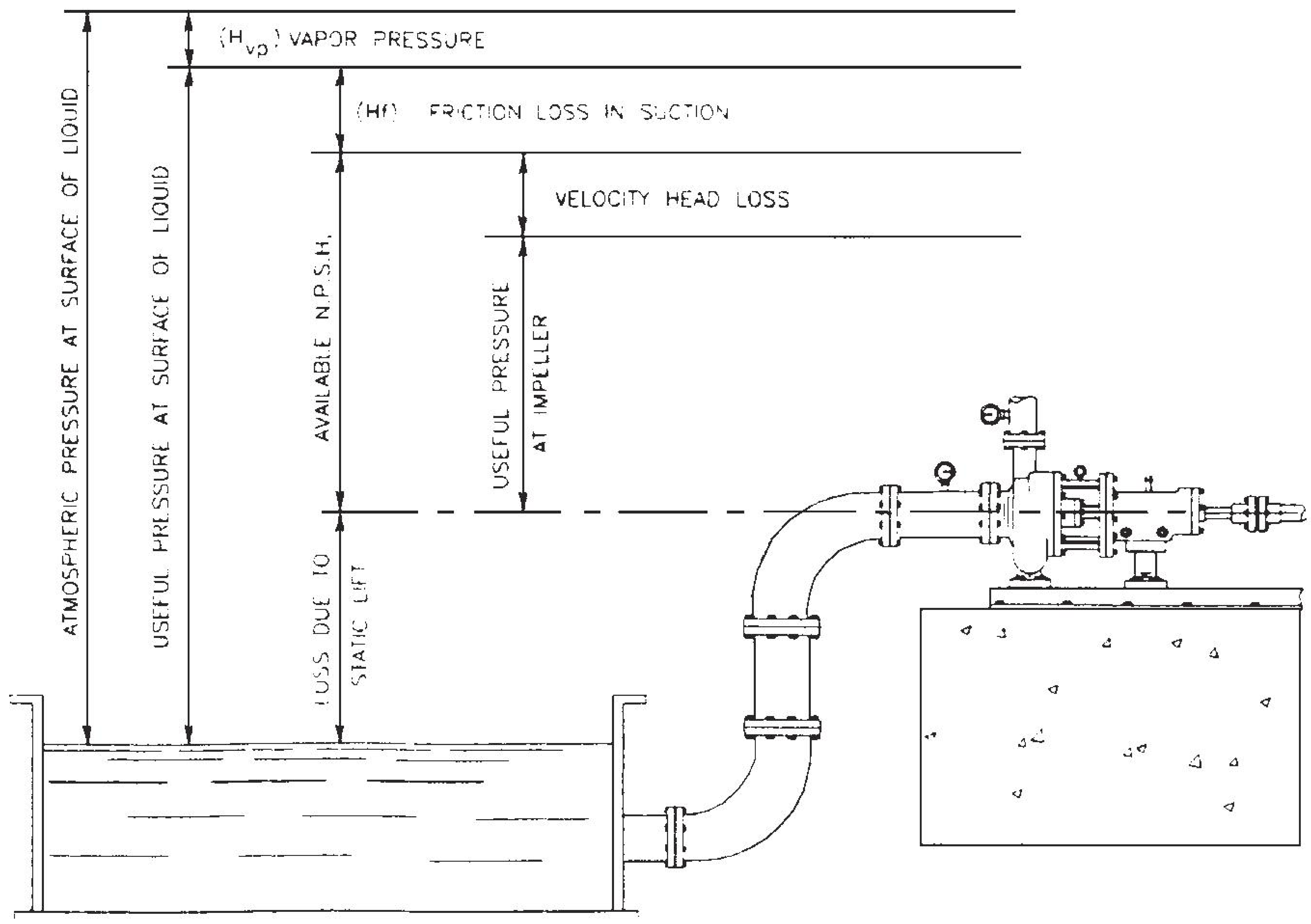

FIGURE 12.2 Net positive suction head in suction lift application.

supply should have relatively constant volume and properties (e.g., pressure, temperature, specific gravity). Special attention must be paid in applications where the liquid has variable physical properties (e.g., specific gravity, density, viscosity). As the suction supply's properties vary, effective pump performance and reliability will be adversely affected.

In applications where two or more pumps operate within the same system, special attention must be given to the suction flow requirements. Generally, these applications can be divided into two classifications: pumps in series and pumps in parallel.

Pumps in Series. The suction conditions of two or more pumps in series are extremely critical (see Fig. 12.3). For each pump, both the flow and pressure must match the required suction conditions of the next pump in the series, since each pump depends on the flow and pressure of the preceding pump.

For example, the first pump in the series may deliver 1000 gallons per minute (gpm) and 100 feet of total dynamic head. The next pump in the series will then have an inlet volume of 1000 gpm, but the inlet pressure will be 100 feet of head minus the pressure losses created by the total vertical lift between the two pumps' centerlines and all friction losses caused by the piping, valves, etc.

This pressure at the suction of the second pump must be at least equal to its minimum NPSH operating requirements. If too low, the pump will cavitate and will not generate sufficient volume and pressure for the process to operate properly.

Pumps in Parallel. Pumps that operate in parallel normally share a common suction supply or discharge (or both). This is illustrated in Fig. 12.4. Typically, a common manifold (i.e., pipe) or vessel is used to supply suction volume and pressure. The manifold's configuration must be such that all pumps receive adequate volume and net positive suction head. Special consideration must be given to flow patterns, friction losses, and restrictions.

One of the most common problems with pumps in parallel is suction starvation. This is caused by improper inlet piping that permits more flow and pressure to reach one or more pumps but supplies insufficient quantities to the remaining pumps. In most cases, this is the result of poor piping or manifold design and may be expensive to correct.

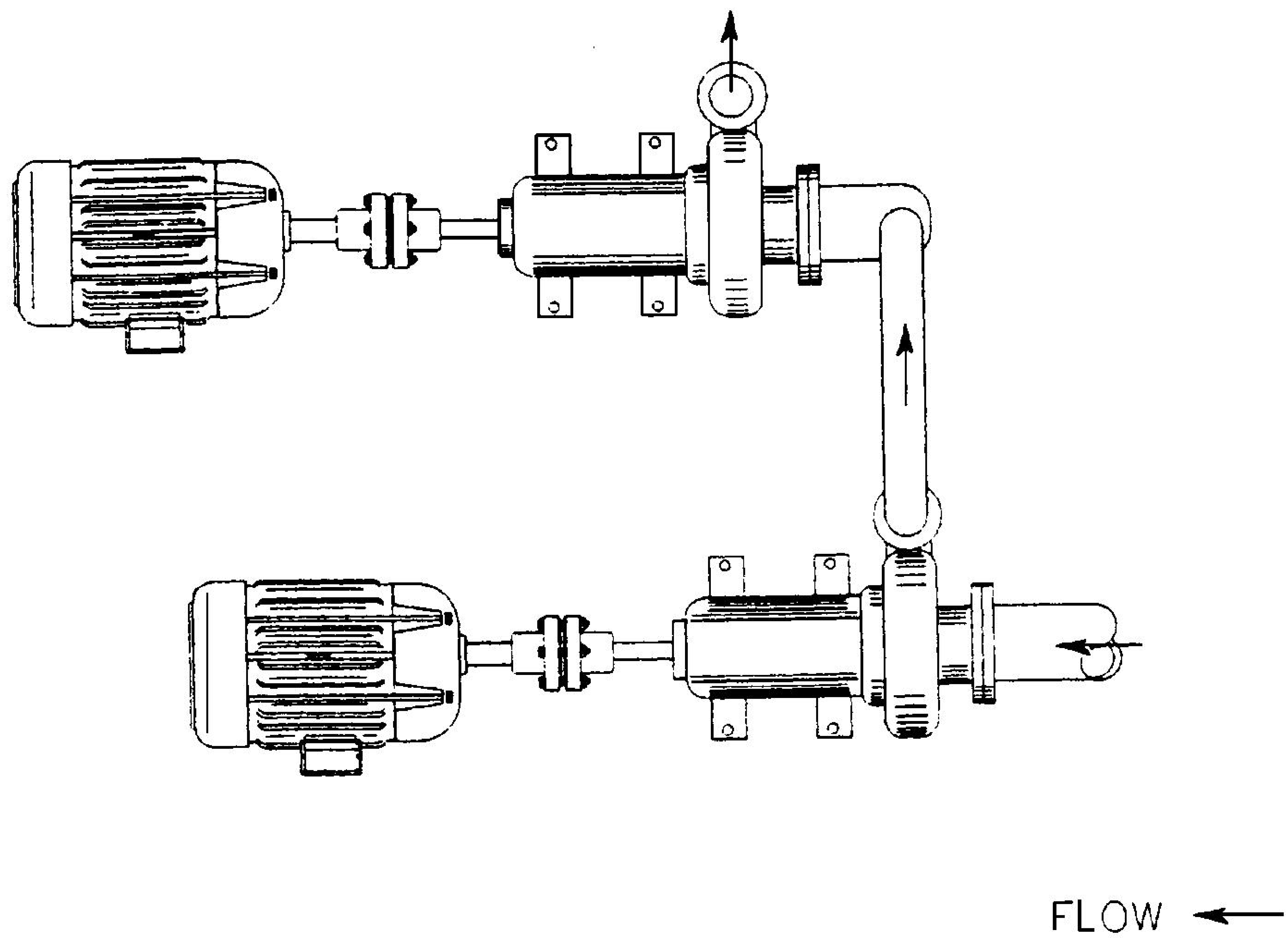

FIGURE 12.3 Pumps in series must be properly matched.

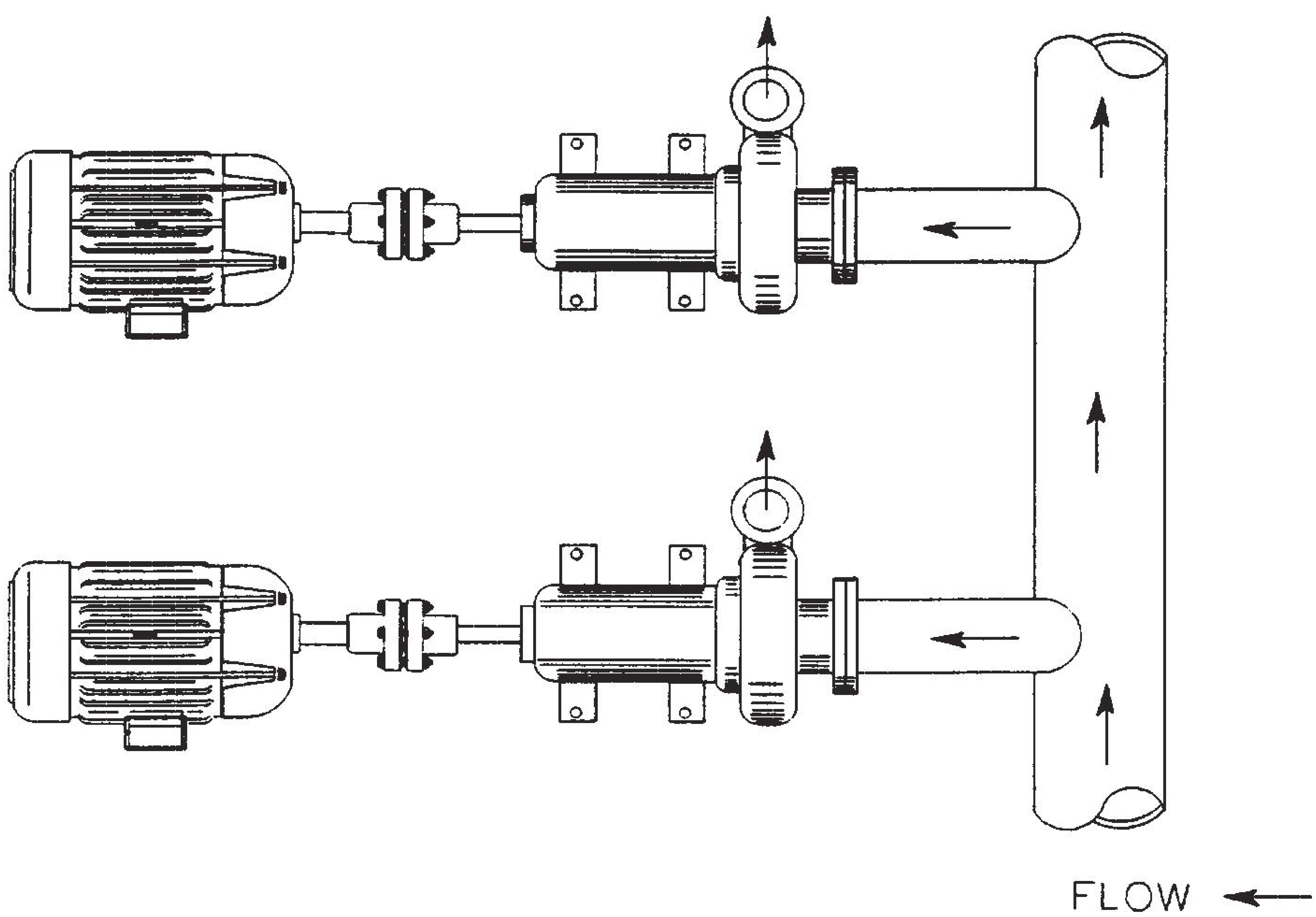

FIGURE 12.4 Pumps in parallel may share suction supply.

Always remember that, when evaluating flow and pressure in pumping systems, the flow will always take the path of least resistance. For example, given a choice of flowing through a 6-in. pipe or a 2-in. pipe, most of the flow will go through the 6-in. pipe. Why? Simply because there is less resistance.

In parallel pump applications, there are two ways to balance flow and pressure to the suction inlet of each pump. The *first* way is to design the piping so that the friction loss and flow path to each of the pumps are equal. While this is theoretically possible, it is extremely difficult to accomplish. The *second* method is to install a balancing valve in each of the suction lines. Throttling or partially closing these valves will allow you to tune the system to ensure proper flow and pressure to each pump.

Entrained Air or Gas. Most pumps are designed to handle single-phase liquids within a limited range of specific gravities or viscosities. Entrainment of gases, such as air or steam, has an adverse effect on both the pump's efficiency and its useful operating life. This is one form of cavitation, which is a common failure mode of centrifugal pumps. The typical causes of cavitation are leaks in suction piping and valves, or a change of phase induced by liquid temperature or suction pressure deviations. As an example, a 1-lb suction pressure change in a boiler-feed application may permit the deaerator-supplied water to flash into steam. The introduction of a two-phase mixture of hot water and steam into the pump causes accelerated wears, instability, loss of pump performance, and chronic failure problems.

Total System Head

Centrifugal pump performance is controlled by the total system head (TSH) requirement, unlike positive-displacement pumps. TSH is defined as the total pressure required to overcome all resistance at a given flow. This value includes all vertical lift, friction loss, and back-pressure generated by the entire system. It determines the efficiency, discharge volume, and stability of the pump.

Total Dynamic Head

Total dynamic head (TDH) is the difference between the discharge and suction pressure of a centrifugal pump. Pump manufacturers use this value to generate hydraulic curves, such as those shown in Figs. 12.5 to 12.7. These curves represent the performance that can be expected for a particular pump under specific operating conditions. For example, a pump having a discharge pressure of 100 psig and a suction positive pressure of 10 psig will have a TDH of 90 psig.

Hydraulic Curve

Most pump hydraulic curves define pressure to be the TDH rather than the actual discharge pressure. This is an important consideration when evaluating pump problems. For example, a variation in suction pressure has a measurable impact on both discharge pressure and volume. Figure 12.5 is a simplified hydraulic curve for a single-stage, centrifugal pump. The vertical axis is TDH and the horizontal axis is discharge volume or flow.

The best operating point for any centrifugal pump is called the *best efficiency point* (BEP). This is the point on the curve where the pump delivers the best combination of pressure and flow. In addition, the BEP defines the point where there will be the most stable pump operation with the lowest power consumption and longest maintenance-free service life.

In any installation, the pump will always operate at the point where its TDH equals the TSH. When selecting a pump, it is hoped that the BEP is near the required flow where the TDH equals TSH on the curve. If it is not, there will be some operating-cost penalty because of the pump's inefficiency. This is often unavoidable because the pumps that are available commercially do not provide this ideal performance.

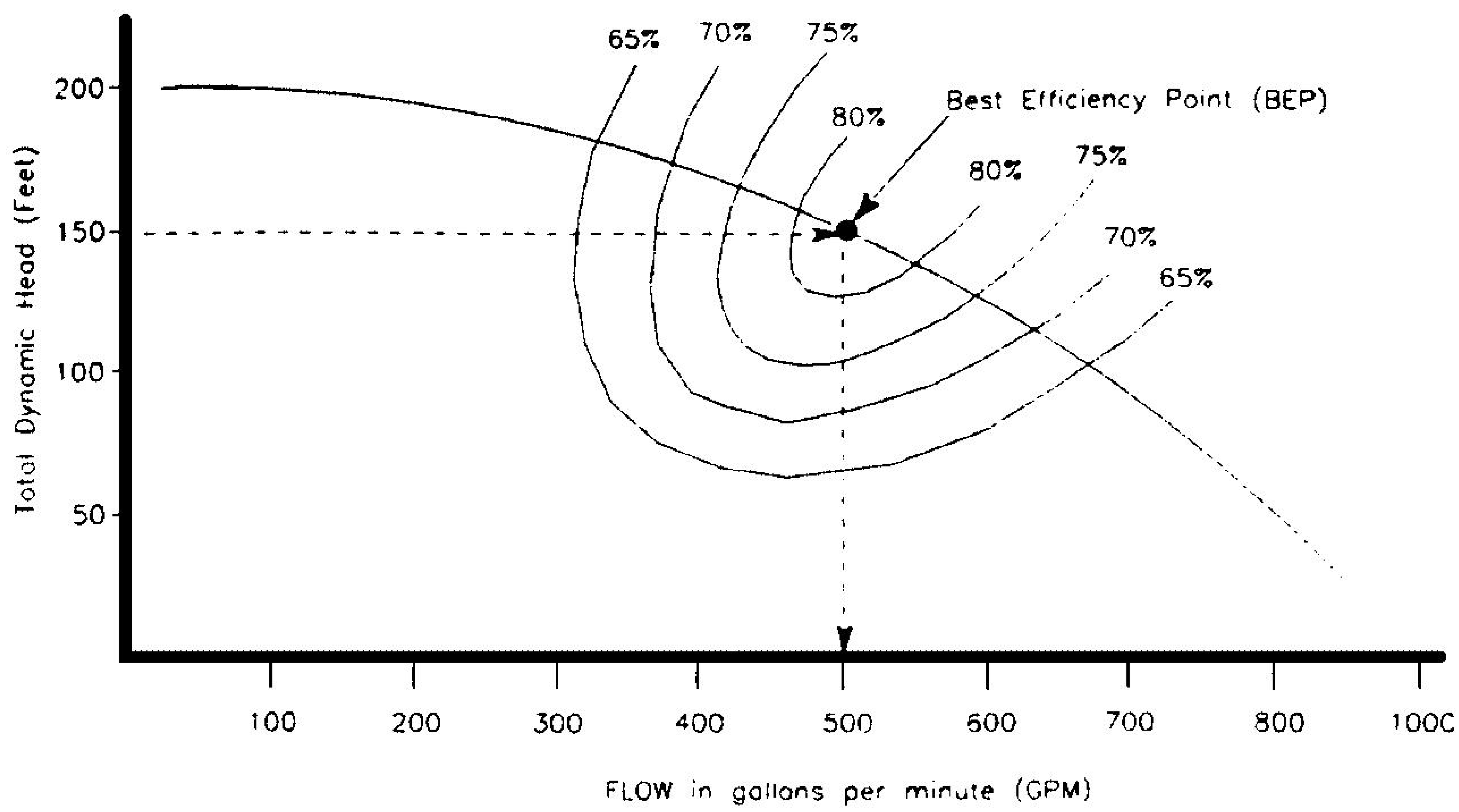

FIGURE 12.5 Simple hydraulic curve for centrifugal pump.

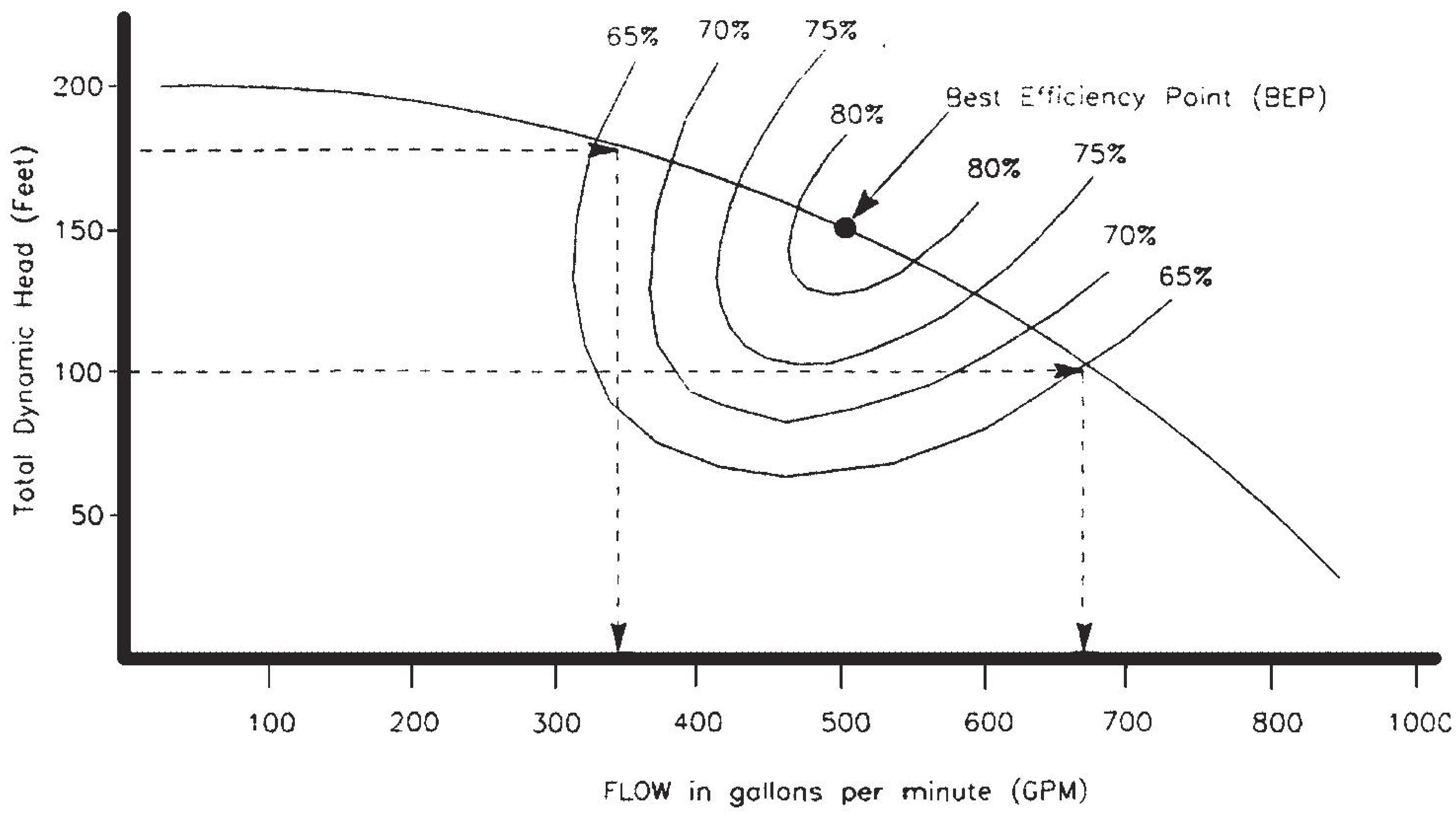

FIGURE 12.6 Actual centrifugal pump performance depends on total system head.

For the centrifugal pump illustrated in Fig. 12.6, the BEP occurs at a flow of 500 gpm with 150 feet TDH. If the TSH were increased to 175 feet, however, the pump's output would decrease to 350 gpm. Conversely, a decrease in TSH would increase the pump's output. For example, a TSH of 100 feet would result in a discharge flow of almost 670 gpm.

From an operating-dynamic standpoint, a centrifugal pump becomes more and more unstable as the hydraulic point moves away from the BEP. As a result, the normal service life decreases and the potential for premature failure of the pump or its components increases. A centrifugal pump should not be operated outside the efficiency range shown by the bands on its hydraulic curve, or 65 percent for the example shown in Fig. 12.5.

If the pump is operated to the left of the minimum recommended efficiency point, it may not discharge enough liquid to dissipate the heat generated by the pumping operation. This can result in a heat buildup within the pump that can cause catastrophic failure. This operating condition, which is called *shutoff,* is a leading cause of premature pump failure.

When the pump operates to the right of the last recommended efficiency point, it tends to overspeed and become extremely unstable. This operating condition, which is called *run-out,* also can result in accelerated wear and premature failure.

Brake Horsepower

Brake horsepower (BHP) refers to the amount of motor horsepower required for proper pump operation. The hydraulic curve for each type of centrifugal pump reflects its performance (i.e., flow and head) at various BHPs. Figure 12.7 is an example of a simplified hydraulic curve that includes the BHP parameter.

Note the diagonal lines that indicate the BHP required for various process conditions. For example, the pump illustrated in Fig. 12.7 requires 22.3 horsepower at its BEP. If the TSH required by the application increases from 150 feet to 175 feet, the horsepower required by the pump increases to 24.6. Conversely, when the TSH decreases, the required horsepower also decreases. The brake horsepower required by a centrifugal pump can be easily calculated by:

$$\text{Brake horsepower} = \frac{\text{flow (gpm)} \times \text{specific gravity} \times \text{total dynamic head (feet)}}{3960 \times \text{efficiency}}$$

With two exceptions, the certified hydraulic curve for any centrifugal pump provides the data required to calculate the actual brake horsepower. Those exceptions are specific gravity and TDH.

Specific gravity must be determined for the specific liquid being pumped. For example, water has a specific gravity of 1.0. Most other clear liquids have a specific gravity of less than 1.0. Slurries and other liquids that contain solids or are highly viscous generally have a higher specific gravity.

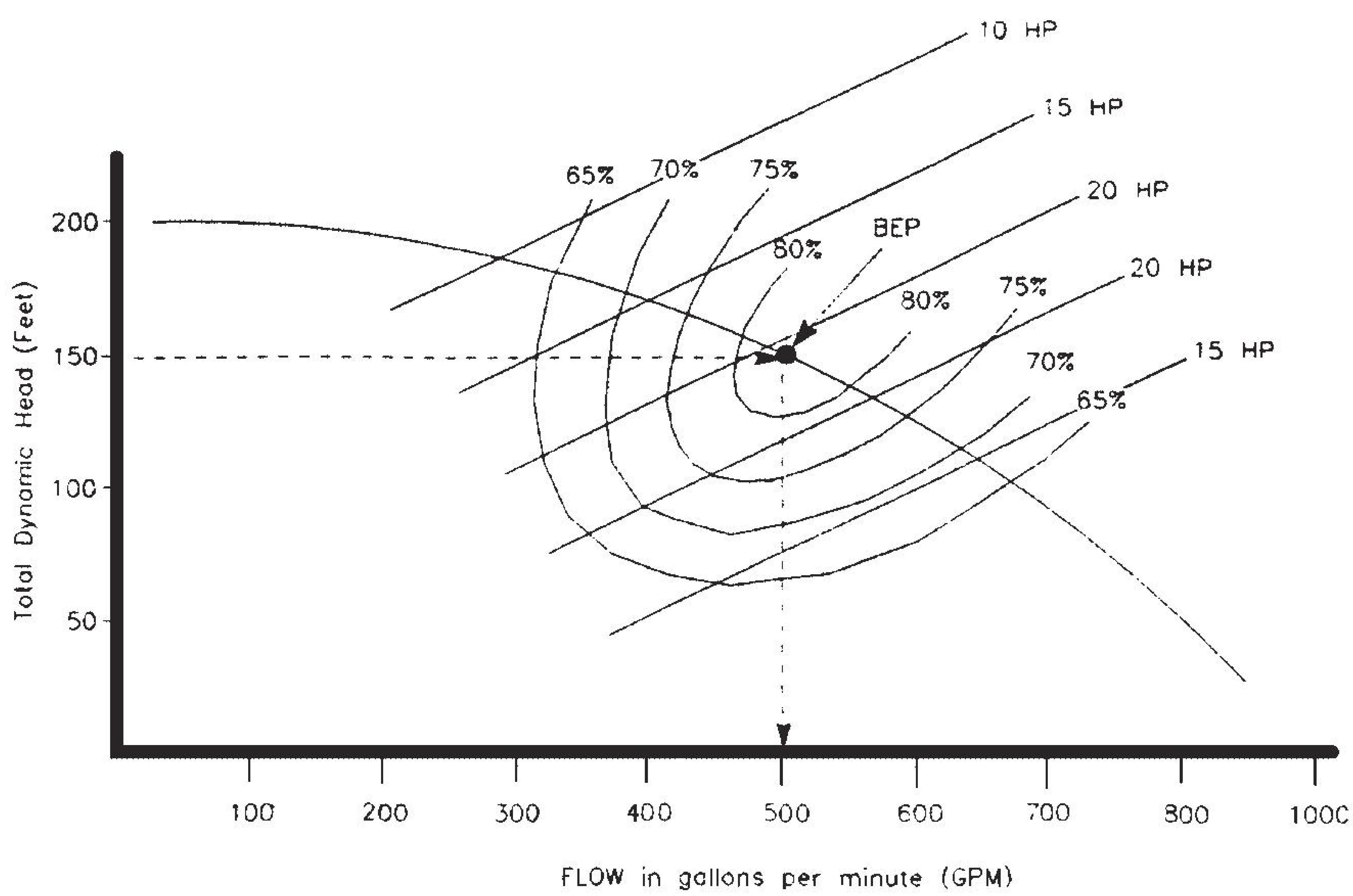

FIGURE 12.7 Brake horsepower needs change with process parameters.

Reference books, like Ingersoll Rand's *Cameron's Hydraulic Databook,* provide the specific-gravity values for many liquids.

The TDH can be directly measured for any application using two calibrated pressure gages. Install one gage in the suction inlet of the pump and another on the discharge. The difference between these two readings is the TDH.

With the actual TDH, flow can be determined directly from the hydraulic curve. Simply locate the measured pressure on the hydraulic curve by drawing a horizontal line from the vertical axis (i.e., TDH) to a point where it intersects the curve. From the intersection point, draw a vertical line downward to the horizontal axis (i.e., flow). This provides an accurate flow rate for the pump.

The intersection point also provides the pump's efficiency for that specific point. Since the intersection may not fall exactly on one of the efficiency curves, some approximation may be required.

Installation

Centrifugal pump installation should follow *Hydraulic Institute Standards,* which provide specific guidelines to prevent distortion of the pump and its baseplate. Distortions can result in premature wear, loss of performance, or catastrophic failure. The following should be evaluated as part of a root cause failure analysis: foundation, piping support, and inlet and discharge piping configurations.

Foundation. A centrifugal pump requires a rigid foundation that prevents torsional or linear movement of the pump and its baseplate. In most cases, this type of pump is mounted on a concrete pad having enough mass to securely support the baseplate, which has a series of mounting holes. Depending on size, there may be three to six mounting points on each side.

The baseplate must be securely bolted to the concrete foundation at all of these points. One common installation error is to leave out the center baseplate lag bolts. This permits the baseplate to flex with the torsional load generated by the pump.

Piping Support. Pipe strain causes the pump casing to deform and results in premature wear and/or failure. Therefore, both suction and discharge piping must be adequately supported to prevent strain. In addition, flexible isolator connectors should be used on both suction and discharge pipes to ensure proper operation.

Inlet-Piping Configuration. Centrifugal pumps are highly susceptible to turbulent flow. The *Hydraulic Institute Standards* provides guidelines for piping configurations that are specifically designed to ensure laminar flow of the liquid as it enters the pump. As a rule, the suction pipe should provide a straight, unrestricted run that is six times the inlet diameter of the pump.

Installations that have sharp turns, shutoff or flow-control valves, or undersized pipe on the suction side of the pump are prone to chronic performance problems. Such deviations from good engineering practices result in turbulent suction flow and cause hydraulic instability that severely restricts pump performance.

Discharge Piping Configuration. The restrictions on discharge piping are not as critical as for suction piping, but using good engineering practices ensures longer life and trouble-free operation of the pump. The primary considerations that govern discharge-piping design are friction losses and total vertical lift or elevation change. The combination of these two factors is called TSH, which represents the total force that the pump must overcome to perform properly. If the system is designed properly, the TDH of the pump will equal the TSH at the desired flow rate.

In most applications, it is relatively straightforward to confirm the total elevation change of the pumped liquid. Measure all vertical rises and drops in the discharge piping, and then calculate the total difference between the pump's centerline and the final delivery point.

Determining the total friction loss, however, is not as simple. Friction loss is caused by a number of factors and all depend on the flow velocity generated by the pump. The major sources of friction loss include:

- Friction between the pumped liquid and the sidewalls of the pipe
- Valves, elbows, and other mechanical flow restrictions
- Other flow restrictions, such as back-pressure created by the weight of liquid in the delivery storage tank or resistance within the system component that uses the pumped liquid

There are a number of reference books, like Ingersoll-Rand's *Cameron Hydraulics Databook,* that provide the pipe-friction losses for common pipes under various flow conditions. Generally, data tables define the approximate losses in terms of specific pipe lengths or runs. Friction loss can be approximated by measuring the total run length of each pipe size used in the discharge system, dividing the total by the equivalent length used in the table, and multiplying the result by the friction loss given in the table.

Each time the flow is interrupted by a change of direction, a restriction caused by valving, or a change in pipe diameter, there is a substantial increase in the flow resistance of the piping. The actual amount of this increase depends on the nature of the restriction. For example, a short-radius elbow creates much more resistance than does a long-radius elbow; a ball valve's resistance is much greater than that of a gate valve; and the resistance from a pipe-size reduction of 4 in. will be greater than for a 1-in. reduction. Reference tables are available in hydraulics handbooks that provide the relative values for each of the major sources of friction loss. As in the friction tables mentioned above, these tables often provide the friction loss as equivalent runs of straight pipe.

In some cases, friction losses are difficult to quantify. If the pumped liquid is delivered to an intermediate storage tank, the configuration of the tank's inlet determines if it adds to the system pressure. If the inlet is on or near the top, the tank will add no back-pressure. However, if the inlet is below the normal liquid level, the total height of liquid above the inlet must be added to the total system head.

In applications where the liquid is used directly by one or more system components, the contribution of these components to the total system head may be difficult to calculate. In some cases, the vendor's manual or the original design documentation will provide this information. If these data are not available, then the friction losses and back-pressure need to be measured or an overcapacity pump needs to be selected for service based on a conservative estimate.

Operating Methods

Normally, little consideration is given to operating practices for centrifugal pumps. However, some critical practices must be followed, such as using proper start-up procedures, using proper bypass operations, and operating the pumps under stable conditions.

Start-up Procedures. Centrifugal pumps should always be started with the discharge valve closed. As soon as the pump is activated, the valve should be slowly opened to its full-open position.

The only exception to this rule is when there is positive back-pressure on the pump at start-up. Without adequate back-pressure, the pump will absorb a substantial torsional load during the initial start-up sequence. The normal tendency is to overspeed because there is no resistance on the impeller.

Bypass Operation. Many pump applications include a bypass loop intended to prevent deadheading (i.e., pumping against a closed discharge). Most bypass loops consist of a metered orifice inserted in the bypass piping to permit a minimal flow of liquid. In many cases, the flow permitted by these metered orifices is not sufficient to dissipate the heat generated by the pump or to permit stable pump operation.

If a bypass loop is used, it must provide sufficient flow to assure reliable pump operation. The bypass should provide sufficient volume to permit the pump to operate within its designed operating envelope. This envelope is bound by the efficiency curves that are included on the pump's hydraulic curve, which provides the minimum flow needed to meet this requirement.

Stable Operating Conditions. Centrifugal pumps cannot absorb constant, rapid changes in an operating environment. For example, frequent cycling between full-flow and no-flow assures premature failure of any centrifugal pump. The radical surge of back-pressure generated by rapidly closing a discharge valve, referred to as *hydraulic hammer,* generates an instantaneous shock load that can literally tear the pump from its piping and foundation.

In applications where frequent changes in flow demand are required, the pump system must be protected from such transients. Two methods can be used to protect the system.

- *Slow down the transient.* Instead of instant valve closing, throttle the system over a longer time interval. This will reduce the potential for hydraulic hammer and prolong pump life.
- *Install proportioning valves.* For applications where frequent radical flow swings are necessary, the best protection is to install a pair of proportioning valves that have inverse logic. The primary valve controls flow to the process. The second controls flow to a full-flow bypass. Because of their inverse logic, the second valve will open in direct proportion as the primary valve closes, keeping the flow from the pump nearly constant.

POSITIVE-DISPLACEMENT PUMPS

Centrifugal and positive-displacement pumps share some basic design requirements. Both require an adequate, constant suction volume to deliver designed fluid volumes and liquid pressures to their installed systems. In addition, both are affected by variations in the liquid's physical properties (e.g., specific gravity and viscosity) and flow characteristics through the pump.

Unlike centrifugal pumps, positive-displacement pumps are designed to displace a specific volume of liquid each time they complete one cycle of operation. As a result, they are not as prone to variations in performance as a direct result of changes in the downstream system. However, there are exceptions to this. Some types of positive-displacement pumps, such are screw-types, are extremely sensitive to variations in system back-pressure.

When positive-displacement pumps are used, the system must be protected from excessive pressures. This type of pump will deliver whatever discharge pressure is required to overcome the system's total head. The only restrictions to its maximum pressure are the burst pressure of the system's components and the maximum driver horsepower.

Because of their ability to generate almost unlimited pressure, all positive-displacement pump systems must be fitted with relief valves on the downstream side of the discharge valve. This is required to protect the pump and its discharge piping from overpressurization. Some designs include a relief valve that is integral to the pump's housing. Others use a separate valve installed in the discharge piping.

Positive-displacement pumps deliver a definite volume of liquid for each cycle of pump operation. Therefore, the only factor, other than pipe blockage, that affects flow rate in an ideal positive-displacement pump application is the speed at which it operates. The flow resistance of the system in which the pump is operating does not affect the flow rate through the pump. Figure 12.8 shows the characteristics curve (i.e., flow rate versus head) for a positive-displacement pump.

The dashed line in Fig. 12.8 shows actual positive-displacement pump performance. This line reflects the fact that, as the discharge pressure of the pump increases, liquid leaks from the discharge back to the suction-inlet side of the pump casing. This reduces the pump's effective flow rate. The rate at which liquid leaks from the pump's discharge to its suction side is called *slip.* Slip is the result of two primary factors: (1) design clearance required preventing metal-to-metal contact of moving parts and (2) internal part wear.

Minimum design clearance is necessary for proper operation, but it should be enough to minimize wear. Proper operation and maintenance of positive-displacement pumps limits the amount of slip caused by wear.

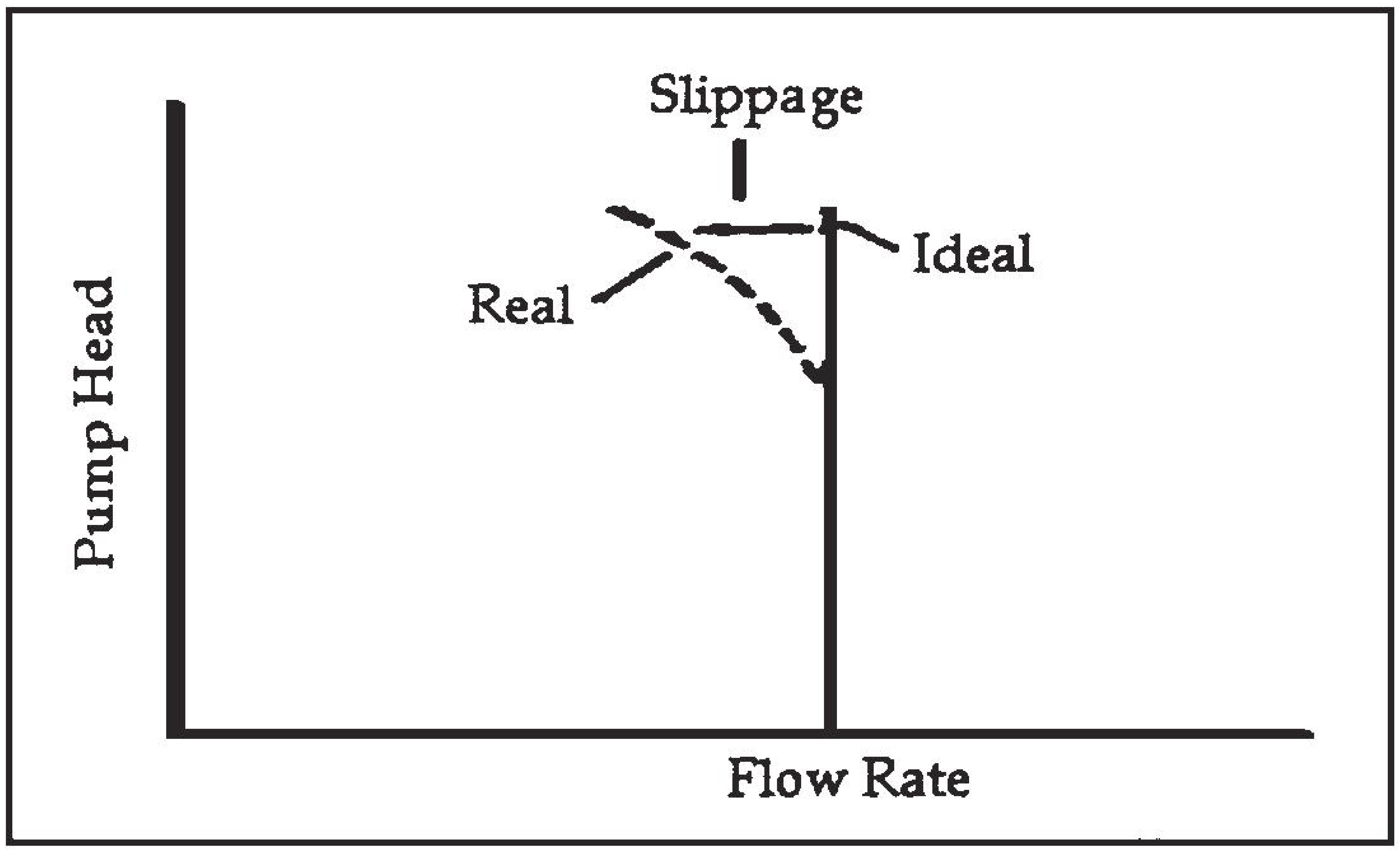

FIGURE 12.8 Positive-displacement pump characteristics curve.

Configuration

Positive-displacement pumps come in a variety of configurations. Each has a specific function and should be selected based on its effectiveness and reliability in a specific application. The major types of positive-displacement pumps are gear, screw, vane, and lobe.

Gear. The most common type of positive-displacement pump uses a combination of gears and configurations to provide the liquid pressure and volume required by the application. Variations of gear pumps are spur, helical, and herringbone.

Spur. The simple spur-gear pump shown in Fig. 12.9 consists of two spur gears meshing and revolving in opposite directions within a casing. A clearance of only a few thousandths of an inch exists between the case, gear faces, and teeth extremities. This design forces any liquid filling the space bounded by two successive gear teeth and the case to move with the teeth as they revolve. When the gear teeth mesh with the teeth of the other gear, the space between them is reduced. This forces the entrapped liquid out through the pump's discharge pipe.

As the gears revolve and the teeth disengage, the space again opens on the suction side of the pump, trapping new quantities of liquid and carrying them around the pump case to the discharge.

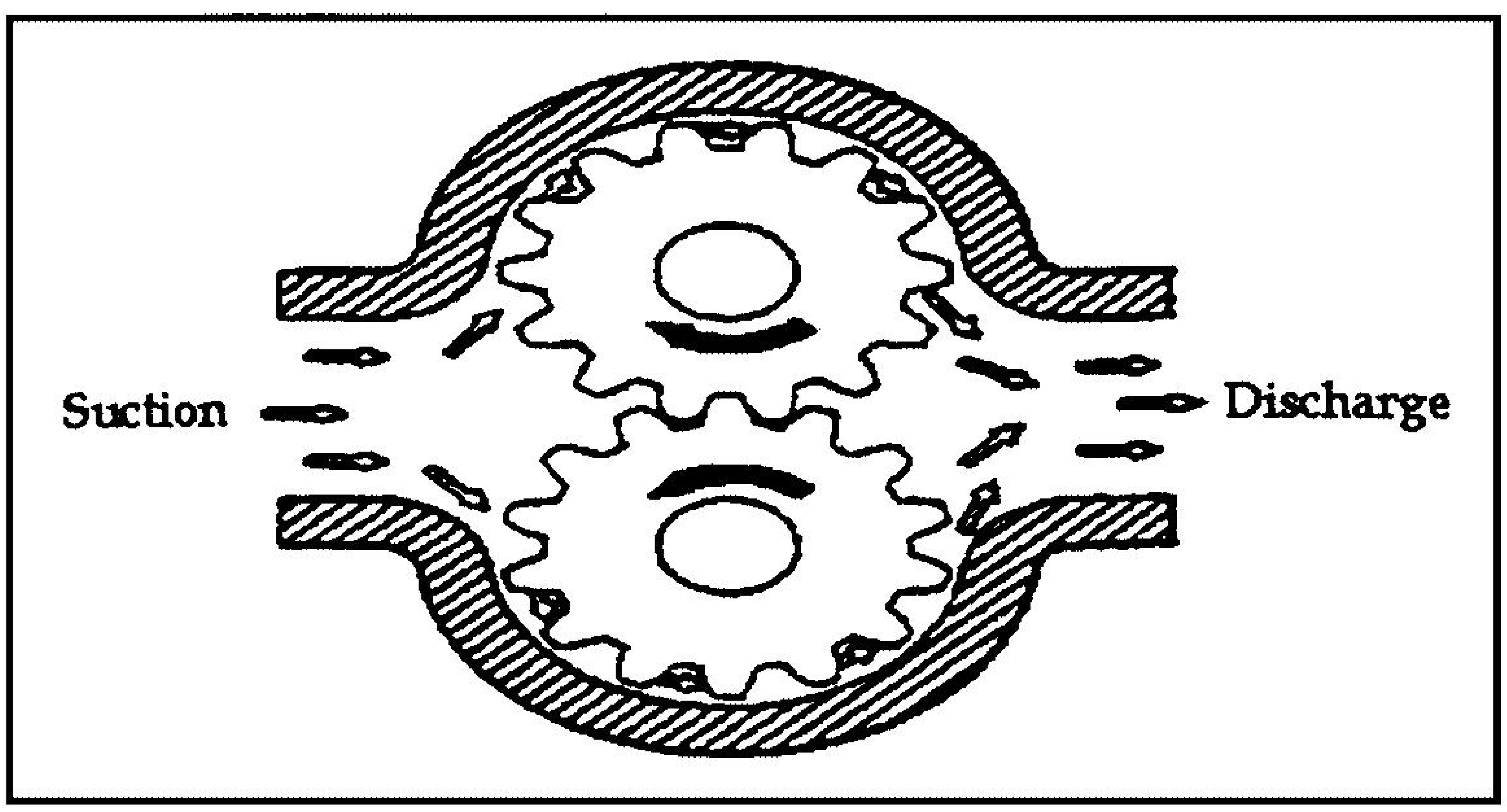

FIGURE 12.9 Simple spur-gear pump.

Lower pressure results as the liquid moves away from the suction side, which draws liquid in through the suction line.

For gears having a large number of teeth, the discharge is relatively smooth and continuous, with small quantities of liquid delivered to the discharge line in rapid succession. For gears having fewer teeth, the space between them is greater and the capacity increases for a given speed. However, this increases the tendency to have a pulsating discharge.

In a simple-gear pump, power is applied to one of the gear shafts, which transmits power to the driven gear through its meshing teeth. There are no valves in the gear pump to cause friction losses as in the reciprocating pump. The high impeller velocities required in centrifugal pumps, which result in friction losses, are not needed in gear pumps. This makes gear pumps well suited for viscous fluids, such as fuel and lubricating oils.

Helical. The helical-gear pump is a modification of the spur-gear pump and has certain advantages. With a spur gear, the entire length of the tooth engages at the same time. With a helical gear, the point of engagement moves along the length of the tooth as the gear rotates. This results in a discharge with a steadier pressure but with less pulsation than that of a spur-gear pump.

Herringbone. The herringbone-gear pump is also a modification of the simple-gear pump. The principal difference in operation from the simple-gear pump is that the pointed center section of the space between two teeth begins discharging fluid before the divergent outer ends of the preceding space complete discharging. This overlapping tends to provide a steadier discharge pressure. The power transmission from the driving gear to the driven gear is also smoother and quieter.

Screw. There are many design variations for screw-type, positive-displacement rotary pumps. The primary variations are the number of intermeshing screws, the screw pitch, and fluid-flow direction.

The most common type of screw pump consists of two screws that are mounted on two parallel shafts and mesh with close clearances. One screw has a right-handed thread, while the other has a left-handed thread. One shaft drives the other through a set of timing gears, which serve to synchronize the screws and maintain clearance between them.

The screws rotate in closely fitting duplex cylinders that have overlapping bores. While all clearances are small, no contact occurs between the two screws or between the screws and the cylinder walls. The complete assembly and the usual flow path for such a pump are shown in Fig. 12.10.

In this type of pump, liquid is trapped at the outer end of each pair of screws. As the first space between the screw threads rotates away from the opposite screw, a spiral-shaped quantity of liquid is enclosed when the end of the screw again meshes with the opposite screw. As the screw continues to rotate, the entrapped spiral of liquid slides along the cylinder toward the center discharge space while the next slug is entrapped. Each screw functions similarly and each pair of screws discharges

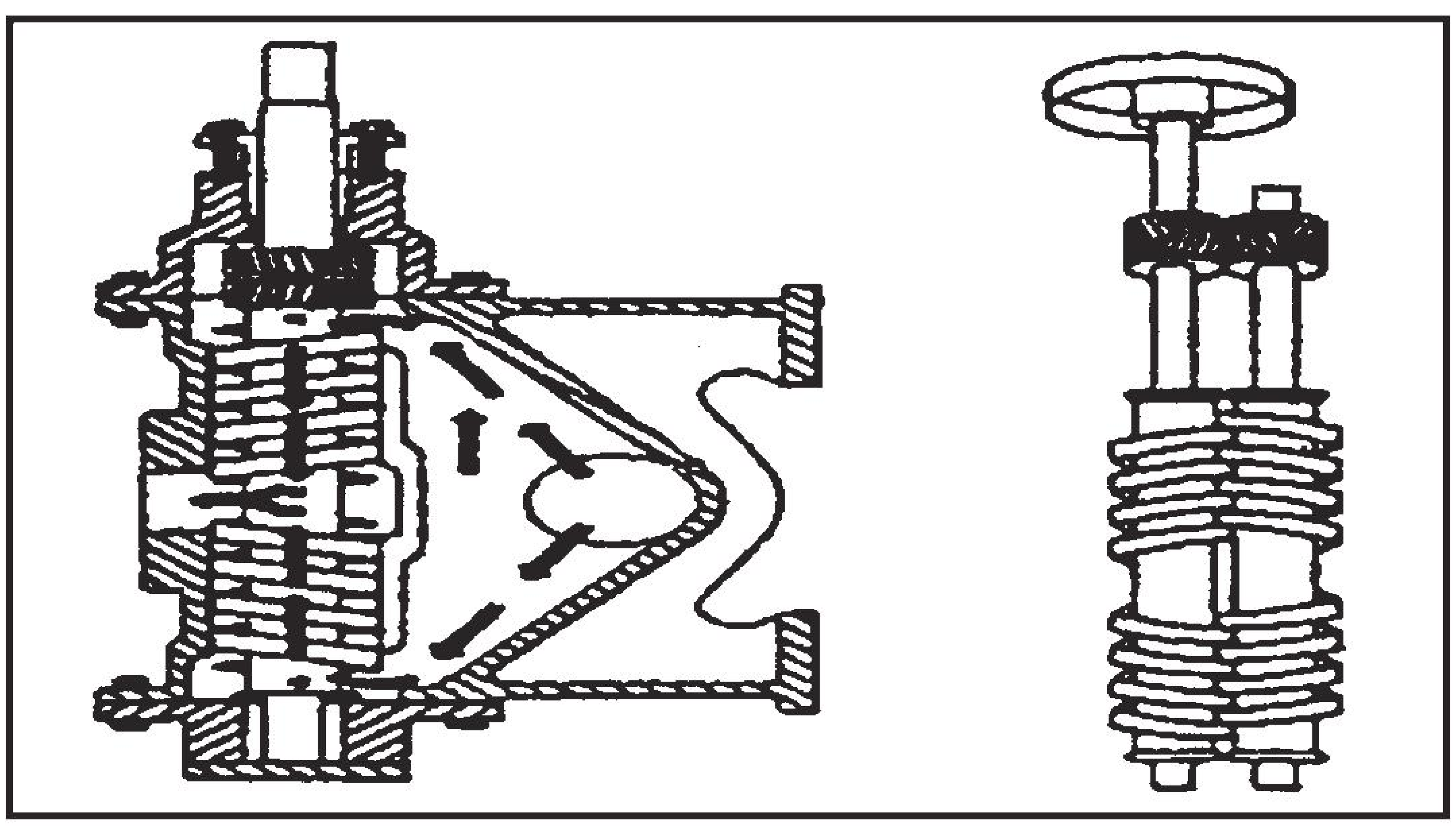

FIGURE 12.10 Two-screw, low-pitch screw pump.

an equal quantity of liquid in opposed streams toward the center, thus eliminating hydraulic thrust. The removal of liquid from the suction end by the screws produces a reduction in pressure, which draws liquid through the suction line.

Vane. The sliding-vane pump shown in Fig. 12.11, another type of positive-displacement pump, is used with viscous fluids. It consists of a cylindrical bored housing with a suction inlet on one side and a discharge outlet on the other. A cylindrical-shaped rotor having a diameter smaller than the cylinder is driven about an axis position above the cylinder's centerline. The clearance between the rotor and the top of the cylinder is small, but it increases toward the bottom.

The rotor has vanes that move in and out as it rotates, maintaining sealed space between the rotor and the cylinder wall. The vanes trap liquid on the suction side and carry it to the discharge side where contraction of the space expels it through the discharge line. The vanes may swing on pivots or they may slide in slots in the rotor.

Lobe. The lobe-type pump shown in Fig. 12.12 is another variation of the simple-gear pump. It can be considered to be a simple-gear pump having only two or three lobes per rotor. Other than this dif-

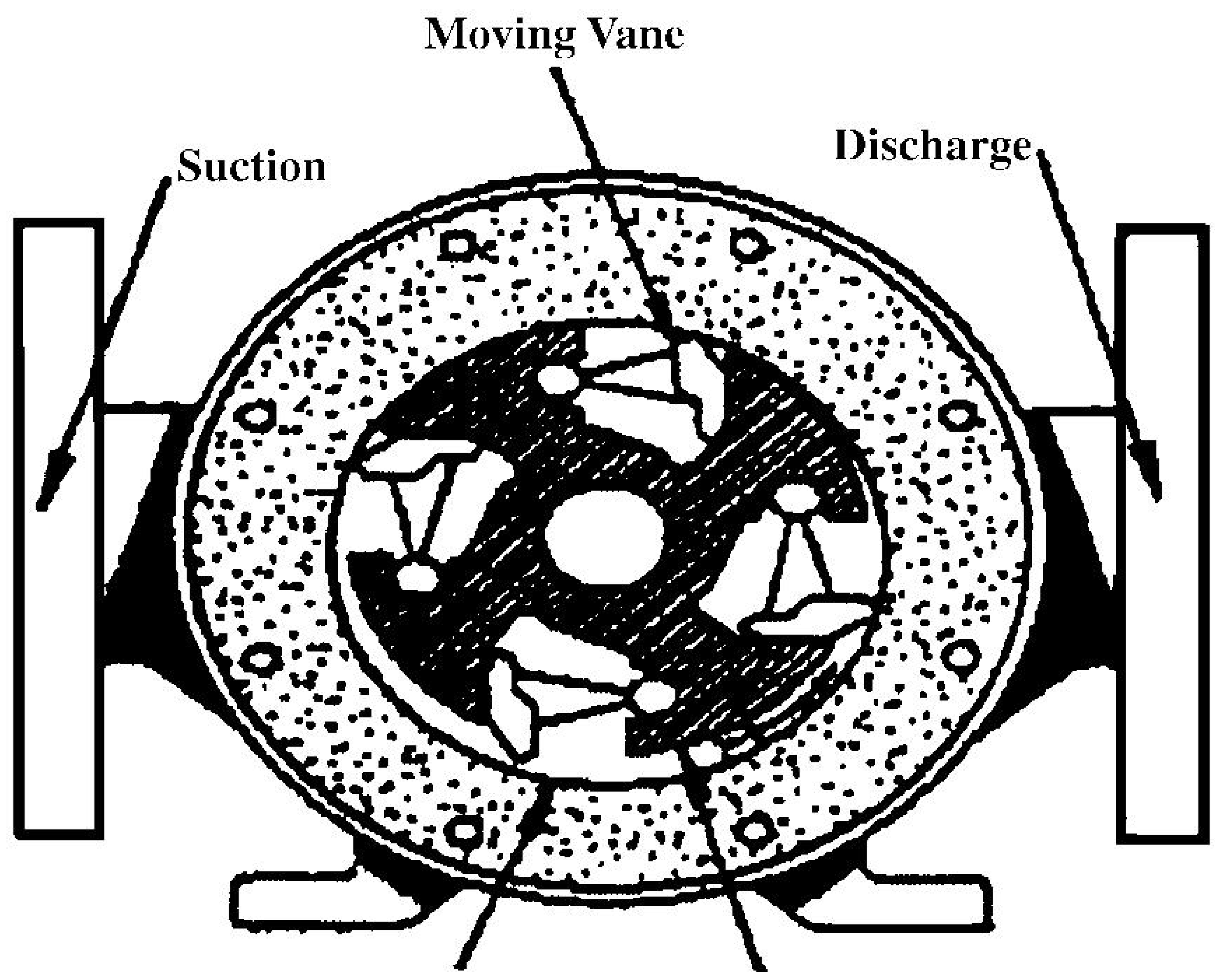

FIGURE 12.11 Rotary sliding-vane pump.

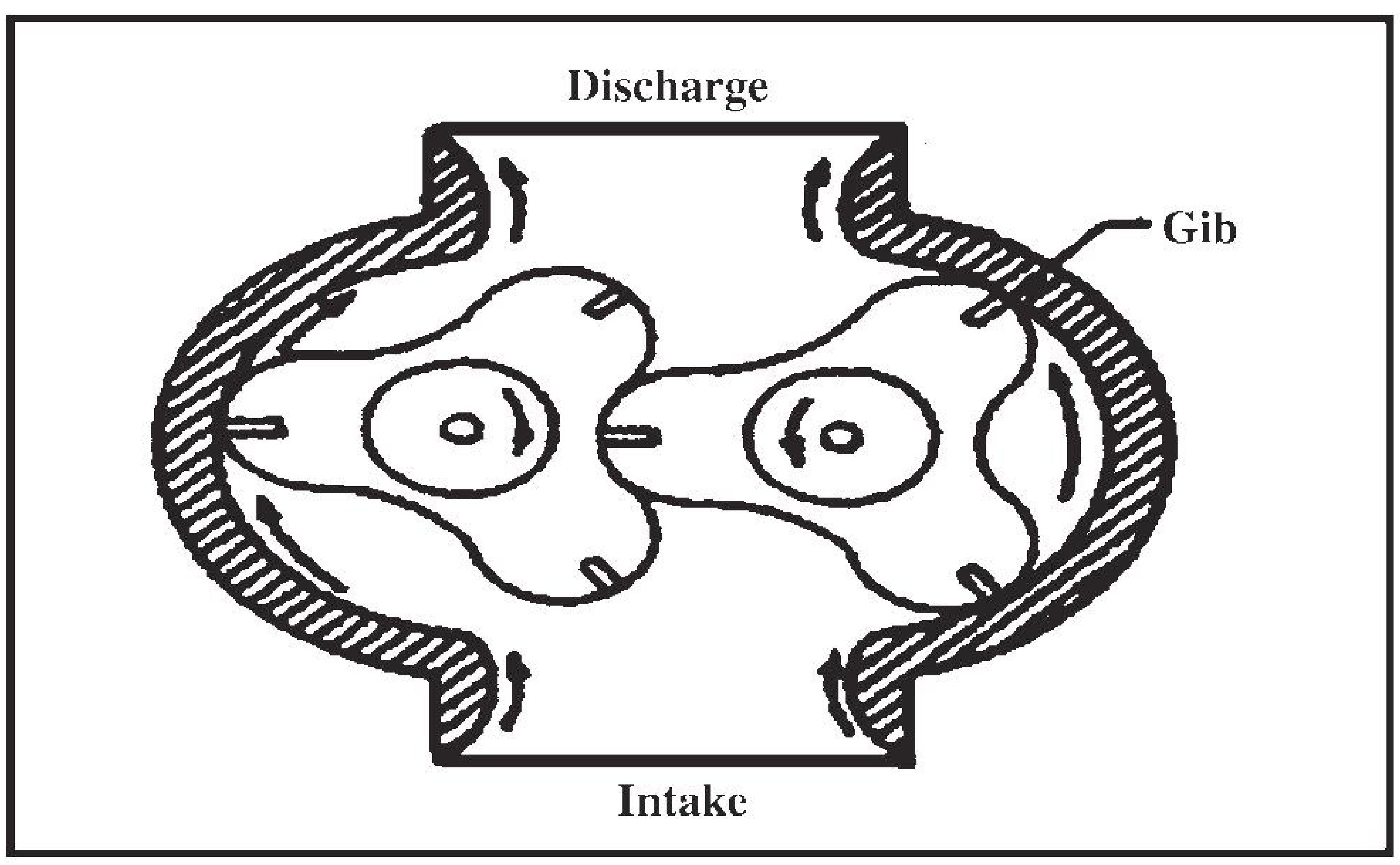

FIGURE 12.12 Lobe-type pump.

ference, its operation and the function of its parts are no different. Some designs of lobe pumps are fitted with replaceable gibes, or thin plates, carried in grooves at the extremity of each lobe where they make contact with the casing. Gibes promote tightness and absorb radial wear.

Performance

Positive-displacement pump performance is determined by three primary factors: liquid viscosity, rotating speed, and suction supply.

Viscosity. Positive-displacement pumps are designed to handle viscous liquids such as oil, grease, and polymers. However, a change in viscosity has a direct effect on the pump's performance. As the viscosity increases, the pump must work harder to deliver a constant volume of fluid to the discharge. As a result, the brake horsepower needed to drive the pump increases to keep the rotating speed constant and prevent a marked reduction in the volume of liquid delivered to the discharge. If the viscosity change is great enough, the brake horsepower requirements may exceed the capabilities of the motor.

Temperature variation is the major contributor to viscosity change. The design specifications should define an acceptable range of both viscosity and temperature for each application. These two variables are closely linked and should be clearly understood.

Rotating Speed. With positive-displacement pumps, output is directly proportional to the rotating speed. If the speed changes from its normal design point, the volume of liquid delivered also will change.

Suction Supply. To a degree, positive-displacement pumps are self-priming. In other words, they have the ability to draw liquid into their suction ports. However, they must have a constant volume of liquid available. Therefore, the suction-supply system should be designed to ensure that a constant volume of nonturbulent liquid is available to each pump in the system.

Pump performance and its useful operating life is enhanced if the suction-supply system provides a consistent positive pressure. When the pumps are required to overcome suction lift, they must work harder to deliver product to the discharge.

Installation

Installation requirements for positive-displacement pumps are the same as those for centrifugal pumps. Special attention should be given to the suction-piping configuration. Poor piping practices in hydraulic-system applications are primary sources of positive-displacement pump problems, particularly in parallel pump applications. Often the suction piping does not provide adequate volume to each pump in parallel configurations.

Operating Methods

If a positive-displacement pump is properly installed, there are few restrictions on operating methods. The primary operating concerns are bypass operation and speed-change rates.

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CHAPTER 9
PIPING[1]

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The first step in any piping maintenance program involves eliminating, as far as possible, basic conditions that make excessive maintenance necessary. These may include severe corrosion, water hammer, or poor piping layout. In the case of both corrosion and water hammer, little can be done until one understands the cause.

Corrosion. Probably the biggest single piping-maintenance problem is corrosion. Books have been written on its theory, but the important point is that internal corrosion of piping is generally caused by atmospheric oxygen dissolved in water, and it stops when oxygen is removed or used up by its attack on the metal. Water coming into a system from the outside is always saturated with oxygen, and will continue to corrode the piping until the oxygen is consumed in the process. That is why service-water lines (always supplied with new water and new oxygen) rust faster than hot-water heating lines, which constantly recirculate the same water.

In the steam-water circuit of power plants, dissolved air (oxygen) enters through the makeup water, and through leaks, into parts of the system under vacuum. The accepted cure is to minimize all such leaks by joint and packing maintenance, and then deaerate the feedwater in a suitably designed heater. Sometimes sodium sulfite is used to remove the last traces of oxygen. Corrosion of condensate lines of heating systems is usually caused by air getting in (through vents, reliefs, and joints) at points where the system is under vacuum.

External corrosion may be rapid where a pipe is frequently wet from "sweating" or other moisture-and particularly if the wet surface is repeatedly exposed to air containing sulfurous or acid fumes. To cure it, remove the cause of the sweating or waterproof the pipe. Pipe buried in cinders or soil will often corrode, particularly if the soil is damp and acid. A practical protection is a watertight covering, generally of asphaltic or similar material, applied directly to the pipe or a spiral wrapping of strong fabric. Normally pipe with perforations or cracks from corrosion or other causes is replaced at once. Where this is not possible because of operating conditions, emergency patches, like those shown in Fig. 9.1, may save a shutdown. These are used on iron and steel pipes.

United States Navy practice for a substantial brazed repair of leaking copper or brass pipe is as follows: Shape the copper patch to fit. Clean the mating surfaces with a file, emery cloth, and hydrochloric acid. Wire patch securely in place. Brick in an enclosure to confine the heat. Heat with an acetylene torch, but do not burn patch or piping. Run in spelter solder (with borax flux) between surfaces. Keep turning pipe back and forth so spelter will run between all parts of braze. When patch is cool, test with water pressure.

A small hole in brass or copper pipe may be closed with a rivet or screw plug. Small weak areas may be temporarily reinforced by tightly wrapping with wire thoroughly soldered together in layers to make a solid band.

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[1]For much of the information in this chapter, the author is indebted to the magazine *Power,* New York, and Crane Company, Chicago: also to other manufacturers who participated from time to time in the preparation of the *Power* material.

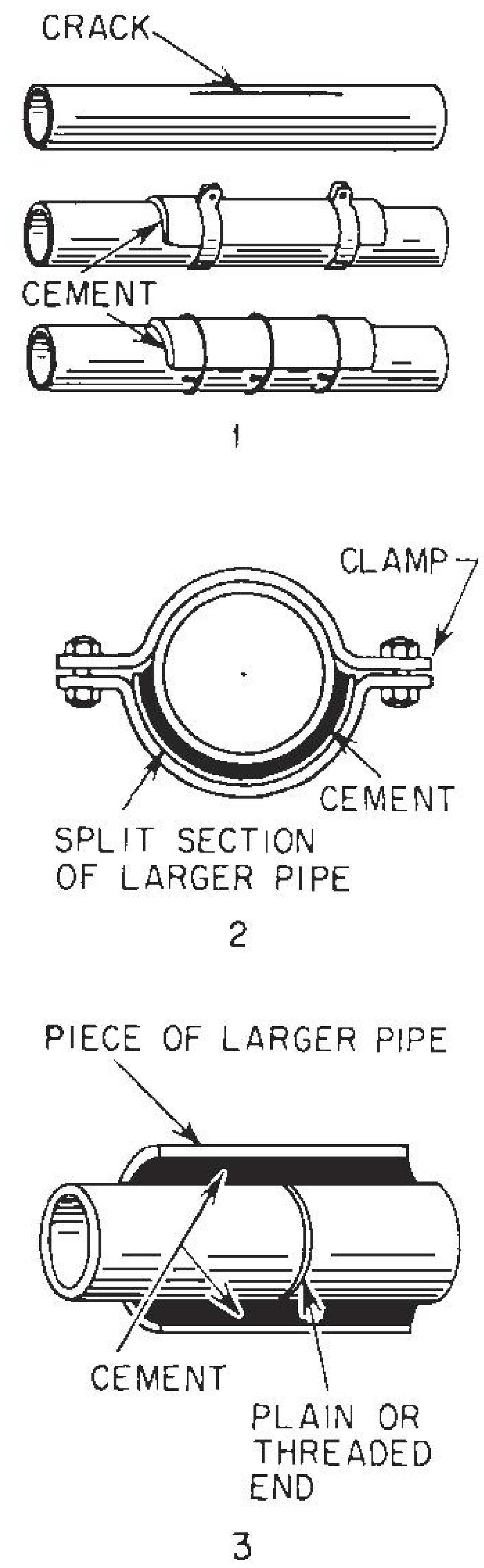


FIGURE 9.1 Emergency repairs for piping leaks. (1) To seal pipe crack, apply iron cement and bind it tight with a metal sheet. (2) Clamp on a half-shell cut from next larger pipe size and seal with cement or soft gasket. (3) For an emergency pipe joint, slide ends into larger pipe and caulk with iron cement.

Water Hammer. This occurs when a moving column of water in a pipe is suddenly stopped or retarded. If the cause is too sudden closing of a valve, the cure is either a mechanical speed limit on the valve or a tag urging cautious handling. Where a pipe is being constantly hammered by connected reciprocating equipment, anchor the pipe firmly and try relieving the shock by air chambers, surge tanks, or similar devices.

Drainage. Failure to remove condensate from steam lines is a major cause of water hammer. Drain all condensate pockets. Make sure that the traps are operating and that no pipe sags so far as to create a pocket. Watch out for condensate caught above closed valves in vertical lines or in back of globe valves in horizontal lines. If water hammer occurs only when steam is admitted to a cold system, it indicates that the system is not adequately pitched or trapped to take care of the large initial condensation. Gradual preheating may ease the situation. Where water hammer has continued for some time, inspect the pipe guides, anchors, and adjacent walls for serious cracks.

When the more obvious ills of a piping system have been remedied, it is time to set up an orderly maintenance procedure to forestall future trouble. In piping, as with other equipment, proper maintenance means preventive maintenance—fixing things before they break. That, in turn, implies organization, records, and definite inspection schedules. The starting point should be a complete set of drawings of the piping system, on which changes and repairs can be noted and dated as made.

Organization. It is not enough to have good piping mechanics, important as this is. Failure to organize maintenance can cause much trouble and unnecessary expense. Complete drawings of the system, with all changes and corresponding dates indicated, will repeatedly save ripping out this or that piece to rediscover facts that should be in the office files. Moreover, recorded installation dates on piping elements will warn the experienced maintenance engineer when trouble may be expected from the everyday causes that lead to piping failure.

When a leak occurs, the adjacent piping should be studied to locate anything else that needs repair, so that the whole job may be done at one time. If the hookup is deficient in unions, or otherwise difficult to maintain, the situation should be remedied while making necessary repairs. General routine inspection crews should check leaks, look for signs of corrosion or weakness, make sure anchors are holding and expansion joints working freely, and check hangers for alignment and distribution of load.

Temperatures. Inspection of an old system should consider any increased pressure or temperature since installation, because these may exceed the safe limits of the materials installed. The ANSI code "Pressure Piping," B31, sets the upper temperature limits for all common piping materials used in industrial plants. Consult the latest edition of the code to determine the proper operational temperature for various materials. Note that compliance with code requirements is purely voluntary.

Pipe Threads. Any detailed discussion of piping maintenance must start with the practical art of pipe threading for two reasons: (1) Badly made threaded joints increase maintenance and endanger plant operation; and (2) the pipe maintenance man is always a piping erector to some degree.

Good threaded joints are mainly a matter of making good threads. These, in turn, require a clean understanding of the proper shape and dimensions of the desired thread and of the die or cutting head that forms it. A brief review of the general scheme of standard American pipe and thread dimensions may bring out certain points often overlooked. Pipe 14 in. or larger is named by its actual outside diameter. Usual OD sizes are 14, 16, 18, 20, 24, 30 in., and larger.

Pipe Data. For 12-in. and smaller, the nominal pipe size is very roughly the inside diameter of so-called "Standard" pipe. The two heavier series called "Extra Strong" and "Double Extra Strong" have the same external diameters but smaller internal diameters. These traditional names will eventually be displaced by the more logical nomenclature of the American National Standards Institute, which sets up a series of pipe schedule numbers of progressively increasing thickness to cover the great variety of modern conditions. For pipe sizes up to 8 or 10 in., Schedule 40 is identical with "Standard," "Schedule 80" with "Extra Strong," "Schedule 160" with "Double Extra Strong." The complete size schedules and other specifications for piping are contained in the ANSI code for "Pressure Piping." Regardless of "weight" or "schedule," pipe of a given nominal size follows the outside and thread dimensions given by the American Standard pipe threads.

Most threads are cut with a set of four or six chasers mounted in a hand die or a machine-operated head. The lip angles are important, and Fig. 9.2 shows lip rake and cutting angles. Note that the cutting angle equals the lip angle only when the face of the chaser lies along a diameter of the pipe. If the chaser is not in this position, allow for it when grinding the lip angle. Since the chaser acts much like a lathe tool, there are similar variations in cutting angle. The cutting angle should be very small for brass, not over 16° for wrought iron, from 15 to 20° for bessemer pipe, and at least 25° for open-hearth pipe.

Heel clearance (Fig. 9.2) is formed in the chaser threads at the factory and cannot be altered or restored with maintenance tools. Clearance reduces as the front part of the chaser wears away with long use. The only cure is a new set. To get the chasers well started on the pipe, they are beveled at the entrance end with a heavy "lead" angle (Fig. 9.2) to a diameter larger than the end of the pipe. Figure 9.2 also shows the lead ends of four successive chasers from the same head. The lead angle is the same in all four, but the cutting edge of each is advanced a little over the next one so that the cutting load is distributed evenly. Maintaining this equal cutting by proper grinding is a matter of first importance.

Figure 9.3 shows how to grind the lip angle on a narrow chaser such as is used in a hand die. This method (using the side of the wheel) is not suitable for the wide, flat chasers used in machine-threading heads. These can be ground on a surface grinder or equivalent rig. Proper grinding of the lead requires the fixture shown (or equivalent), in which the table can be turned about either horizontal or vertical axes

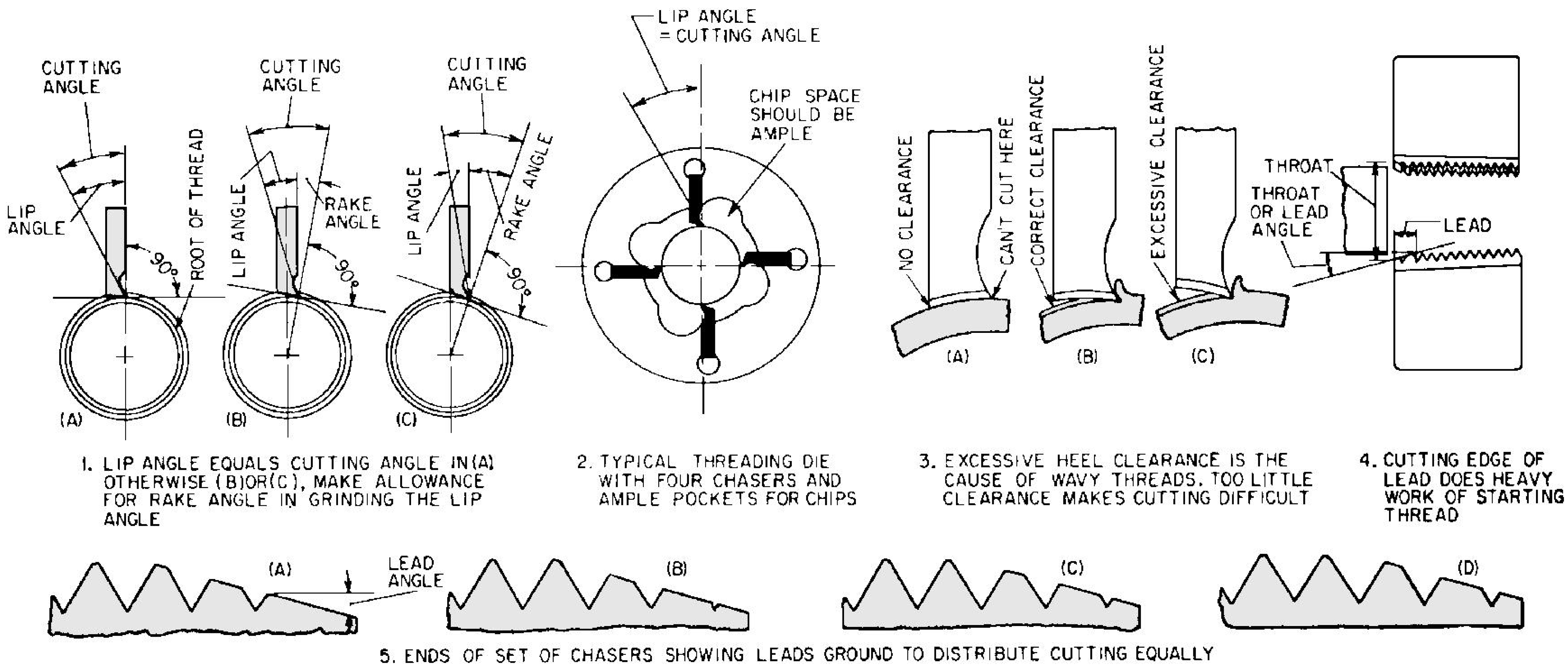


FIGURE 9.2 Lip, rake, and cutting angles for thread chasers.

to give universal adjustments. Before grinding chaser leads, arrange them in serial order 1, 2, 3, 4. Test the angles of the fixture to make sure the wheel bears squarely on the existing face of metal to be ground. Then take chasers in rotation and remove the same amount of metal from each. If any in the set is then found to be insufficiently ground, start back with number one and give each an equal additional grinding. This even treatment should be watched and tested to make sure each chaser carries its share of the cutting load.

Figure 9.4 gives a number of useful pointers in the care of broken and damaged threaded joints of many different types. Figure 9.5 shows ways to salvage the flanges of pipes in which the threads rust, preventing easy removal of the flange. The normal engagement to make tight joints is given in Fig. 9.6.

Flanges. These are designed to ensure a tight joint that can conveniently be broken for piping changes and repairs. Yet poor selection of flanges, bolts, and gaskets, plus careless joint makeup, can cause endless trouble. Once the desired face has been selected (see Fig. 9.7*A*), flange dimensions and materials are established by the ANSI code for "Pressure Piping." The customary flange bolt for low-pressure piping is square-headed with a hex nut and American Standard Coarse Thread Series threads. Above 160 psi and 450°F, alloy-steel studs with hex nuts on both ends are often used. These nuts are generally of carbon steel or at best a less strong alloy than the studs. Extreme strength is not needed and the difference in metals in contact reduces chances of thread "freezing."

For high-pressure-flange studs the 8-pitch thread series, which provides eight threads per inch for all bolts 1 in. or larger, is often used. Standard flange-drilling templates come in multiples of four holes—four, eight, twelve, sixteen, etc. The flange holes in valves and fittings are set to straddle the centerline. Of the flange faces shown in Fig. 9.7*A* some engineers prefer the raised face or the pipe-lap, with a ring gasket, because these joints (unlike the male-female and tongue-and-groove) do not have to be sprung apart to break a joint or remove a gasket. Grooved joints with thick, soft gaskets are often used for low-temperature, high-pressure hydraulic joints. With either male-female or tongue-and-groove joints, the gasket should be thinner than the depression to avoid "mushrooming."

Gaskets. Widely used gasket materials range from soft rubber for cold water to narrow, solid iron rings for high-pressure steam joints. Table 9.1 shows typical uses and limiting temperatures. Hand-cut gaskets for raised-face flanges should fit nearly inside the bolts and extend to, but not beyond, the edge of the pipe opening. If the joint has to be broken frequently, coat one side of the gasket with graphite to prevent sticking. When a joint has been newly made with soft packing, take up on the bolts again after the line has been hot for some time.

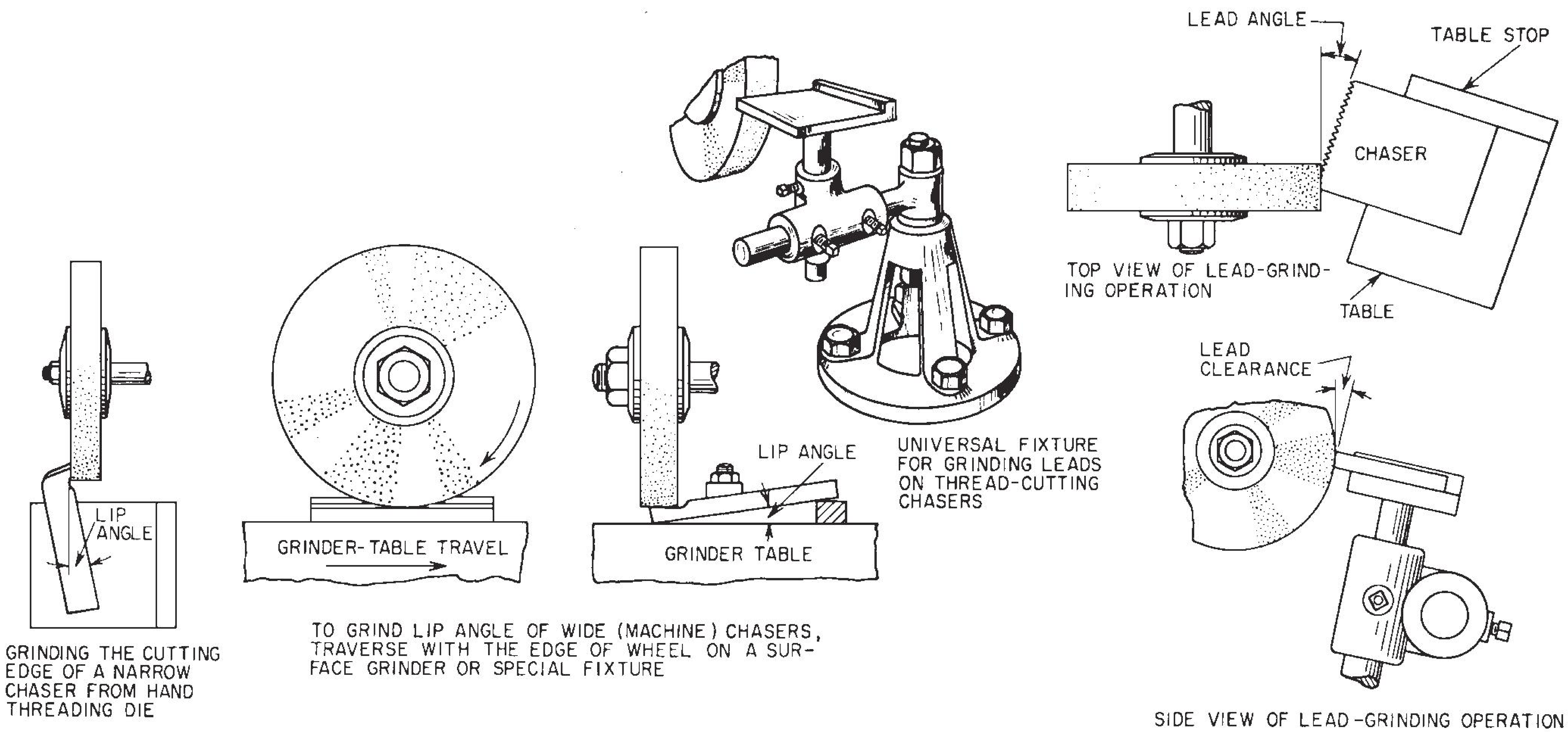


FIGURE 9.3 Steps in grinding chasers.

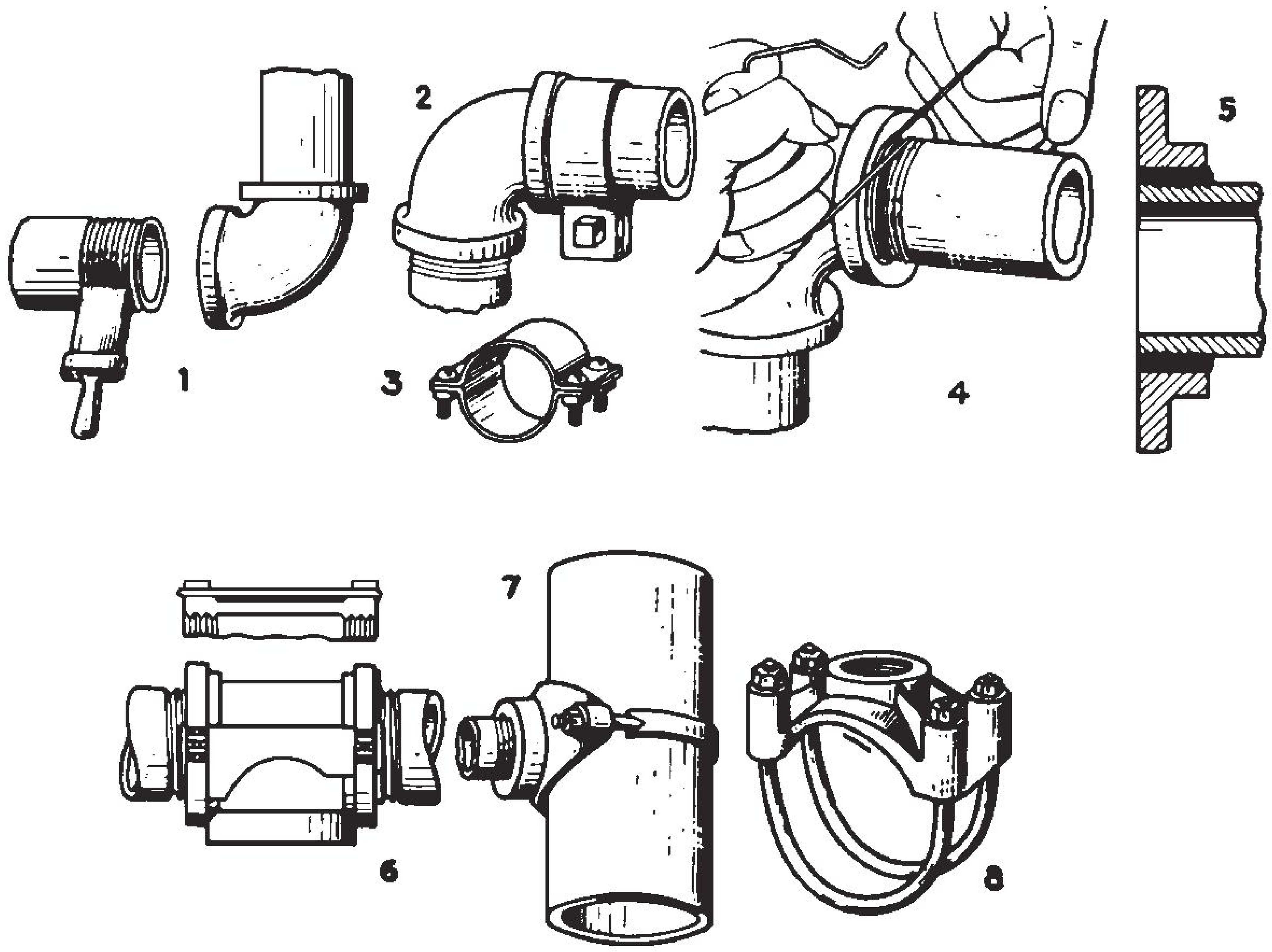


FIGURE 9.4 Repairs for threaded joints. Iron cements specially compounded for the service may be applied as paste or putty to seal pipe joints and cracks. Cement used thin makes a good new-thread dope (1). For best results, leading joints should be remade with cement; otherwise cement can be caulked under pipe clamp (3) alongside leaky thread (2). Another thread-leak repair involves winding soft wire (4) around cement putty at leak. Allow cement to harden before using. In emergency, a threaded pipe may be cemented into a flange of larger pipe size (5). Tamp cement solidly in place. Crack in fitting (6) sealed with cement backed by metal plate. Outlet saddles for water (7) and steam (8) are clamped to pipe over drilled holes to make emergency branch connections. Saddles may be packed with soft gaskets or cement.

Thin gaskets are less likely to blow than thick. If flanges fail to meet, it may be risky to fill the gap with a thick, soft gasket. It is better to use a metal filler, gasketed on both sides. Raised faces of flanges are often serrated. These serrations, like corrugations in gaskets and the use of narrow gaskets, are means for increasing the gasket pressure per square inch by reducing contact area. Most leaky joints are the result of insufficient bolt tension and gasket pressure. According to Crocker,[2] the initial gasket pressure should be at least 4000 psi for rubber, 12,000 psi for laminated asbestos in serrated joints, and 30,000 to 60,000 psi for solid-metal gaskets.

Ordinarily a tensile stress of about 7000 psi in the bolt would balance the actual steam pressure, yet the alloy-steel bolts of high-pressure joints are often stressed to 30,000 psi and sometimes to 60,000 psi to flow the gasket into the uneven surface of the flange and thereby ensure tightness. Tests have been made and tabulated showing how much wrench pull will create a given bolt tension in well-lubricated threads of a given pitch. The practical value of such tables is limited because of the great variation of pull with thread lubrication and also because it is rarely convenient to measure wrench pull, particularly where the wrench must be sledged. From the practical angle, in the case of all medium- and low-pressure work, the fitter may as well continue to tighten by "feel" and experience. When this will not do, as with 1400-psi steam lines, the only reliable determination of bolt tension is by micrometer measurement of bolt elongation, using studs with machined micrometer pads and taking care that before-and-after measurements are taken cold and at the same temperature. The measured elongation must be referred to the grip distance between the nuts and not to the full length of the stud.

For steel of any composition a stress of 30,000 psi corresponds closely to a stretch of 0.001 in. per in., with other stresses and stretches in proportion up to the elastic limit. Thus, for a bolt with 3-in. grip, the total stretch should be 0.002 in. to create a unit tension of 20,000 psi in the bolt.

[2]"Piping Handbook," 5th ed., McGraw-Hill Book Company, New York, 1967.

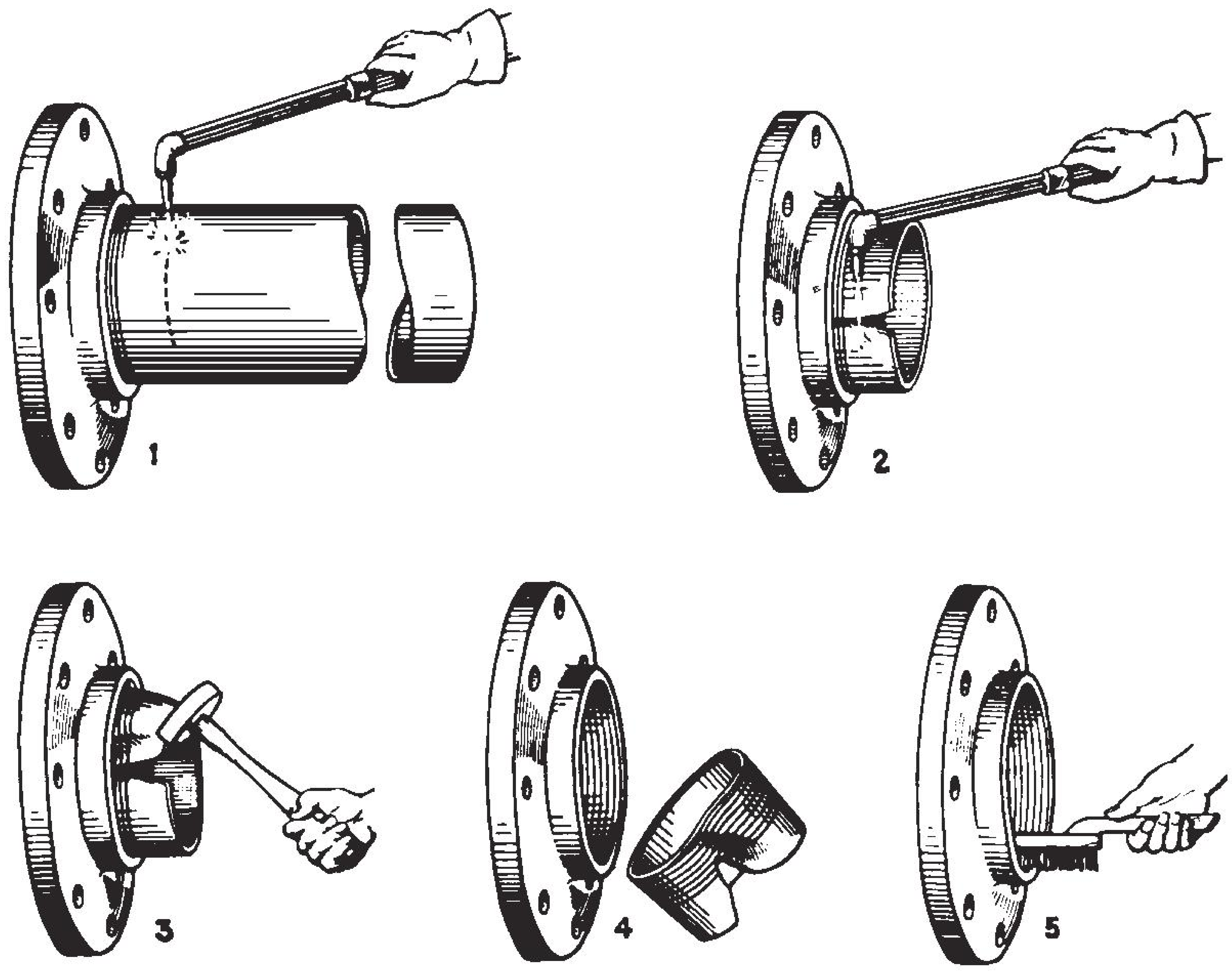
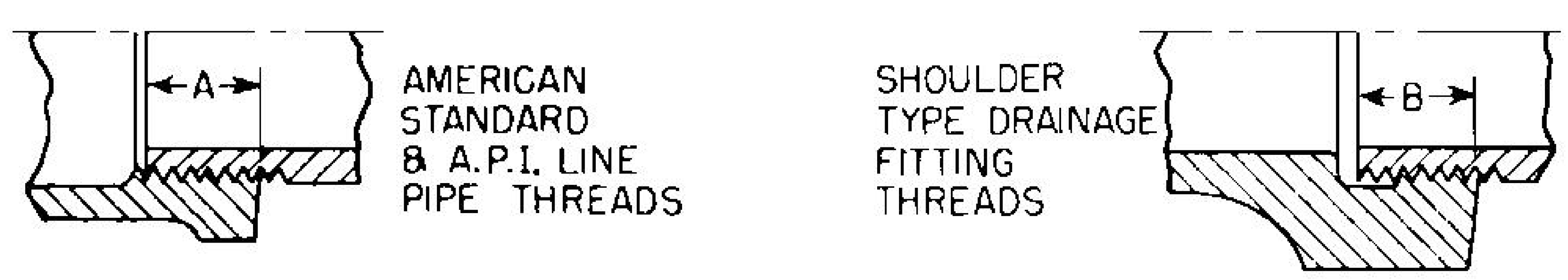


FIGURE 9.5 Salvaging flanges when the threads are frozen. (1) Server pipe near flange. (2) Cut V notch in pipe stub. (3) Collapse pipe with hammer. (4) Pipe falls out, leaving flange threads unharmed. (5) Clean threads thoroughly.



DIMENSIONS, IN INCHES

Dimensions given do not allow for variations in tapping or threading.

Size	1/8	1/4	3/8	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12	14
A	1/4	3/8	3/8	1/2	9/16	1 1/16	1 1/16	1 1/16	3/4	1 5/16	1	1 1/16	1 1/8	1 1/4	1 5/16	1 7/16	1 5/8	1 3/4	
*B**						9/16	5/8	5/8	5/8	7/8	1 5/16	1	1 1/16	1 3/16	1 1/4	1 3/8	1 9/16	1 11/16	1 7/8

FIGURE 9.6 Normal engagement between male and female threads to make tight joints.

Leaks. Like other piping leaks, those in flange joints grow rapidly. Fix them at the start. A frequent cause is poor alignment of piping. It should not be necessary to spring flanges into line (Fig. 9.7*B*). All steamfitters agree that nut tightening should follow a definite sequence. Two are shown in Fig. 9.8*A*—one "round and round," the other "crisscross." Either should be satisfactory. An improvement on the second might be to tighten a single bolt on one side, then on the opposite—then move 90° and repeat. Figure 9.8*B* gives a number of pointers on the maintenance of flanged joints.

Valves. Damage in storage or handling and poor installation handicap a valve from the start. Complete wrapping, wooden crating, or thin metal caps over the ends protect valves as shipped from the factory. Keep this protection in place and store the valve under cover until it is installed. Penetration of sand into working parts often follows storage on the ground, exposed to the weather. Rough handling easily damages valves; place them where they cannot fall and where other material cannot fall on them.

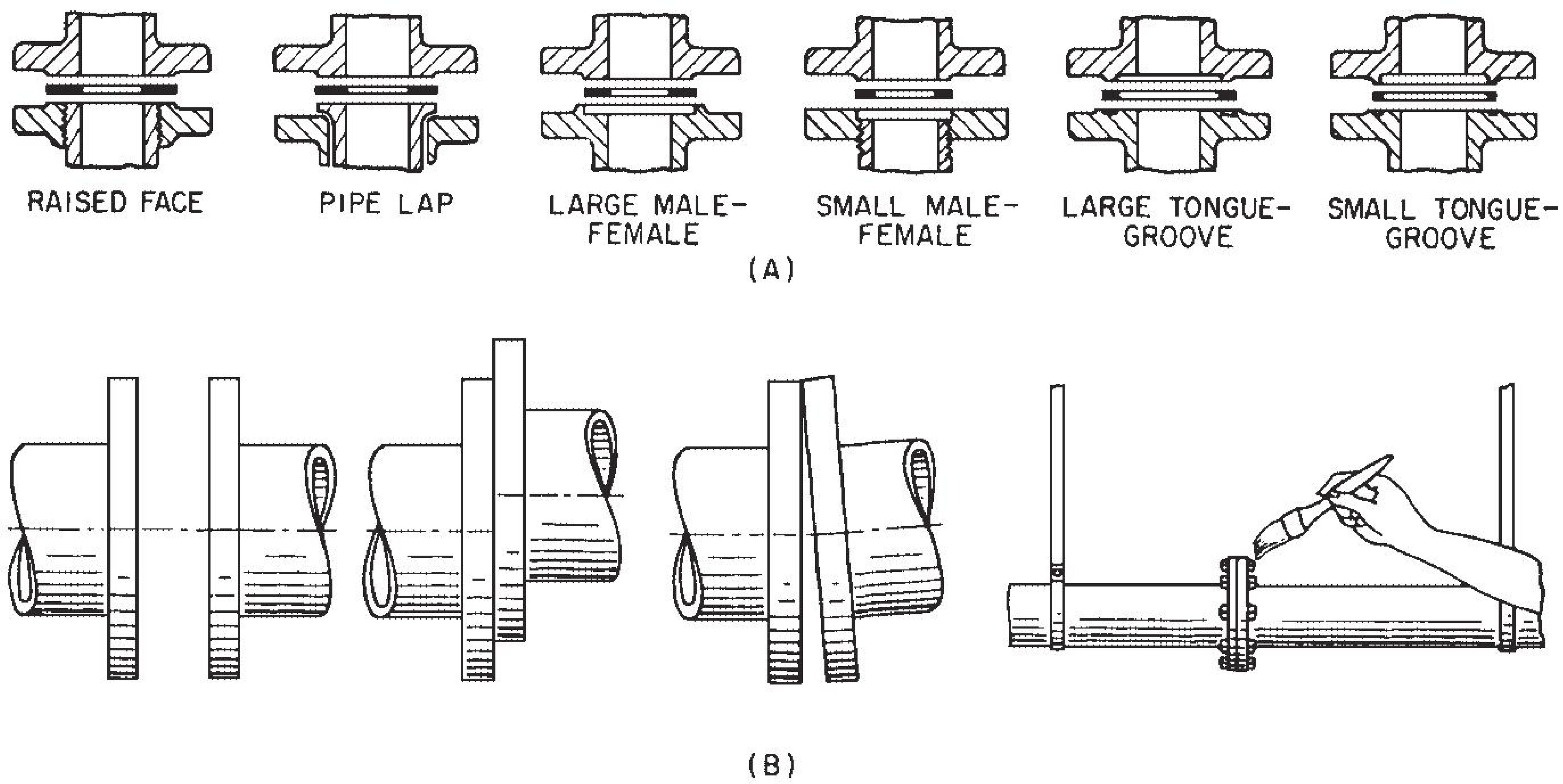


FIGURE 9.7 (*a*) Standard cast-steel flanges. (*b*) Align flanges to avoid trouble; careful aligning before bolting prevents excessive bolting stresses in valves, fittings, and pipe flanges. Use of thread lubricant cuts friction, protects threads, and makes joints easier to break for necessary repairs.

TABLE 9.1 Gasket-Material Selection

Gasket material	Fluid	Usual maximum temp, °F
Red rubber	Steam, air, water	250
Asbestos composition	Steam, water, oil	750
Fiber and paper	Oil	200
Synthetic rubber	Oil	200
Copper, corrugated or plain	Steam or water	600
Steel, corrugated or plain	Steam or water	1000
Stainless steel, 12–14% chromium, corrugated	Steam or water	1000
Hydrogen-annealed furniture iron	Steam or water	1000
Monel, corrugated or plain	Steam or water	1000
Ingot iron, special gasket for ring-type joint	Steam, water, oil	1000

From Crocker, "Piping Handbook," 5th ed., McGraw-Hill Book Company, New York, 1967.

Installation starts with removal of the valve's protective covering; then clean all grit and dirt from the inside. Pipe cleaning is just as necessary if damage to seats and disks is to be avoided. Blow out the valve with compressed air or flush it with water; clean the pipe in the same manner, or pull a swab through, to remove dirt and metal chips left from threading operations or storage.

Future troubles can be minimized by mounting valves properly, protecting them against outside damage, and locating them at the most suitable point in the line. Except for split-wedge and double-disk gates, most valves can be mounted at any angle, although it is always better, from the valve standpoint, to mount it with stem pointing upward. Any position with the stem pointing downward brings the bonnet under the line of flow, forming a pocket to catch pipe scale and other foreign matter. This soon cuts and destroys inside stem threads. On lines exposed to freezing temperatures, moisture trapped at this point may cause frozen and burst bonnets. Even when the valve is mounted with its stem upright, take the precaution of installing a drain plug in the bottom of the body. Figures 9.9 and 9.10 show a number of installation pointers.

Flow Direction. Direction of flow through globe valves depends on the nature of service and can usually be determined by asking the question, "Should the valve open or shut if the disk and stem part company?" The ASME Boiler Code requires pressure under the disk for globe valves in boiler-feed

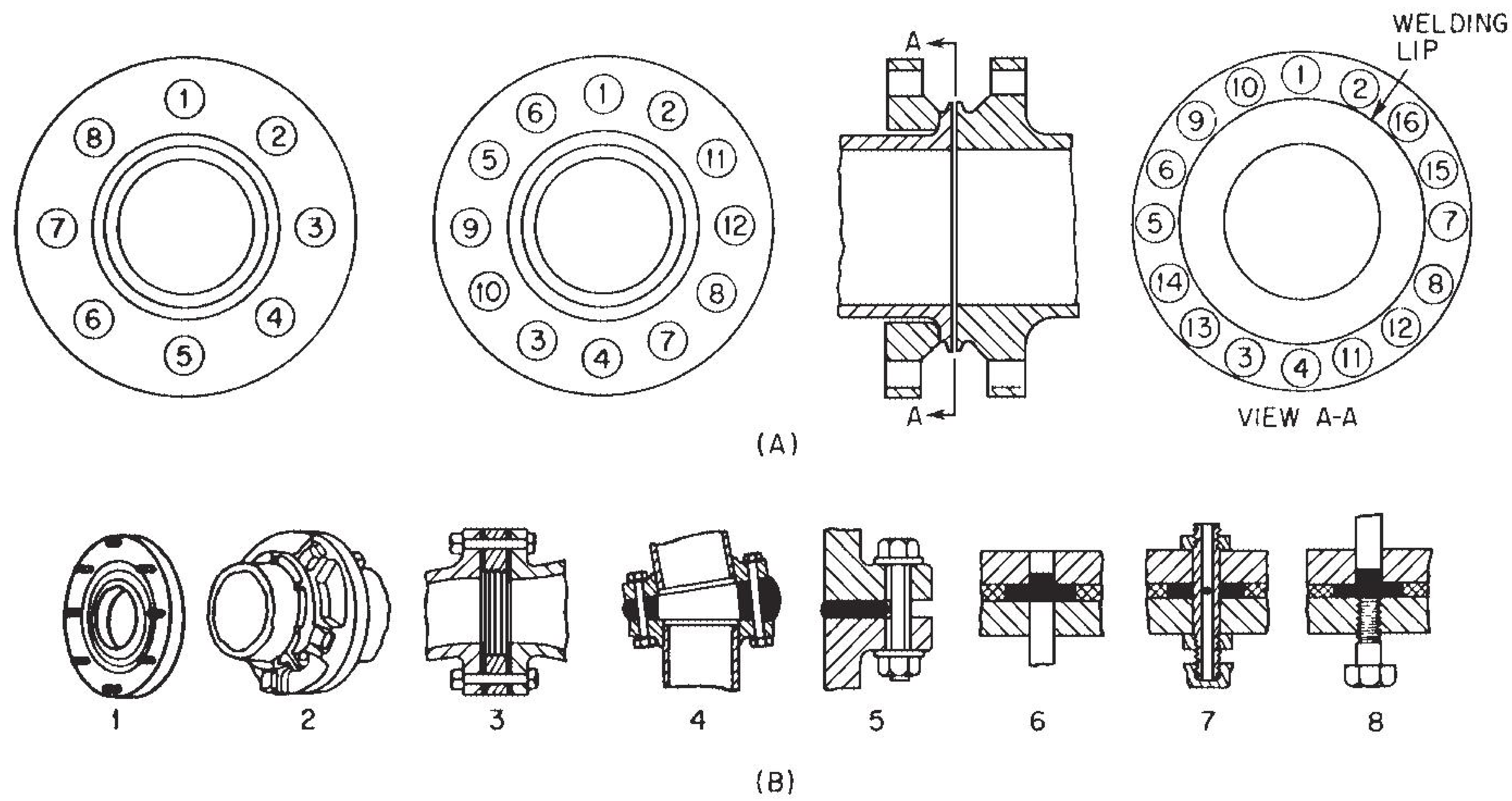


FIGURE 9.8 (*A*) Two sequences of bolt tightening. In either case, first set nuts finger-tight. For method at left, tighten nuts moderately in order 1, 2, 3, etc. Then make another round to set bolts a little tighter—and so on until bolts are equally and sufficiently tight and feelers show equal flange separation all around. Many prefer crossover method. Bolt tightening for seal-welded joint: First put in temporary bolts all around, hammer-tight. Remove 1 and 2, seal weld there, and replace temporary with alloy bolts set up hammer-tight. Repeat this operation at 3 and 4, 5 and 6, etc., crisscrossing as shown. Follow some general procedure for a different number of bolts, crisscrossing as before. (*B* Iron cement aids in making and repairing flanged joints. Corrugated-iron gasket coated both sides with iron cement will make tight joints (1) despite irregular flange faces. To plug leaky thread at flange, set band clamp with flat edge against flange (2) and tamp groove full of cement. If flanges cannot be aligned, turn a metal "Dutchman" (3,4) and seal it on both sides with iron cement or cement-plastered gaskets. To make a tight joint with rough-cast flanges (5), separate flanges by rope of soft spacers; then fill joint with cement. Localized leaks in flanges force cement through other side by hammering on a close-fitting rod. Or (7) inject cement with grease gun through pipe nipple with locknuts and a central side-outlet hole. If stud screws into flange, reverse stud (8) making a backstop for cement.

lines so that the loose disk will not act as a check and stop water flow. If the valve controls equipment which might overspeed, applying pressure over the disk forms a check valve which will shut down the unit if the disk comes loose. Drain valves with pressure under the disk will vibrate open if not tightly closed. If a valve is persistently left cracked open, reversing it to put pressure over the disk will ensure tighter closing.

To summarize: Unless the service clearly requires pressure under the disk, install the valve to put pressure on top of the disk. Pressure over the disk aids in keeping the valve closed, tending to compensate for any stem contraction caused by temperature changes. Valves of large size, 12 in. and over, present generous disk areas to line pressure. In unbalanced service, such as discharging from high to low pressure, this pressure makes valve operation increasingly difficult as exposed disk area increases. This is less pronounced in gates than in globes, because disk movement in the former crosses the line of flow (Fig. 9.11), whereas in the latter it opposes the flow. To minimize pressure difficulties, equip all gate valves 12 in. and over and all globe valves over 6 in. with throttling-globe bypass valves.

In pipelines handling sludge or other suspended matter, keep valves out of vertical lines whenever possible; stoppage of flow allows suspended matter to settle and choke a closed valve. This is especially troublesome when it interferes with check-valve operation. Never install a valve without thought as to access. To be sure valves are opened and closed correctly, make it convenient and safe to do so. Do not expect a man standing on a ladder reaching for an overhead valve to exert any great amount of force. Install the valve horizontally, and fit it with a chain operator which can be reached from floor level. Overhead valves on large lines, requiring two-man operation, should be vertical, to permit operators to stand on the pipe. If possible install a working platform, extend the valve stem through the floor above, and provide an impactor handwheel or a power operator.

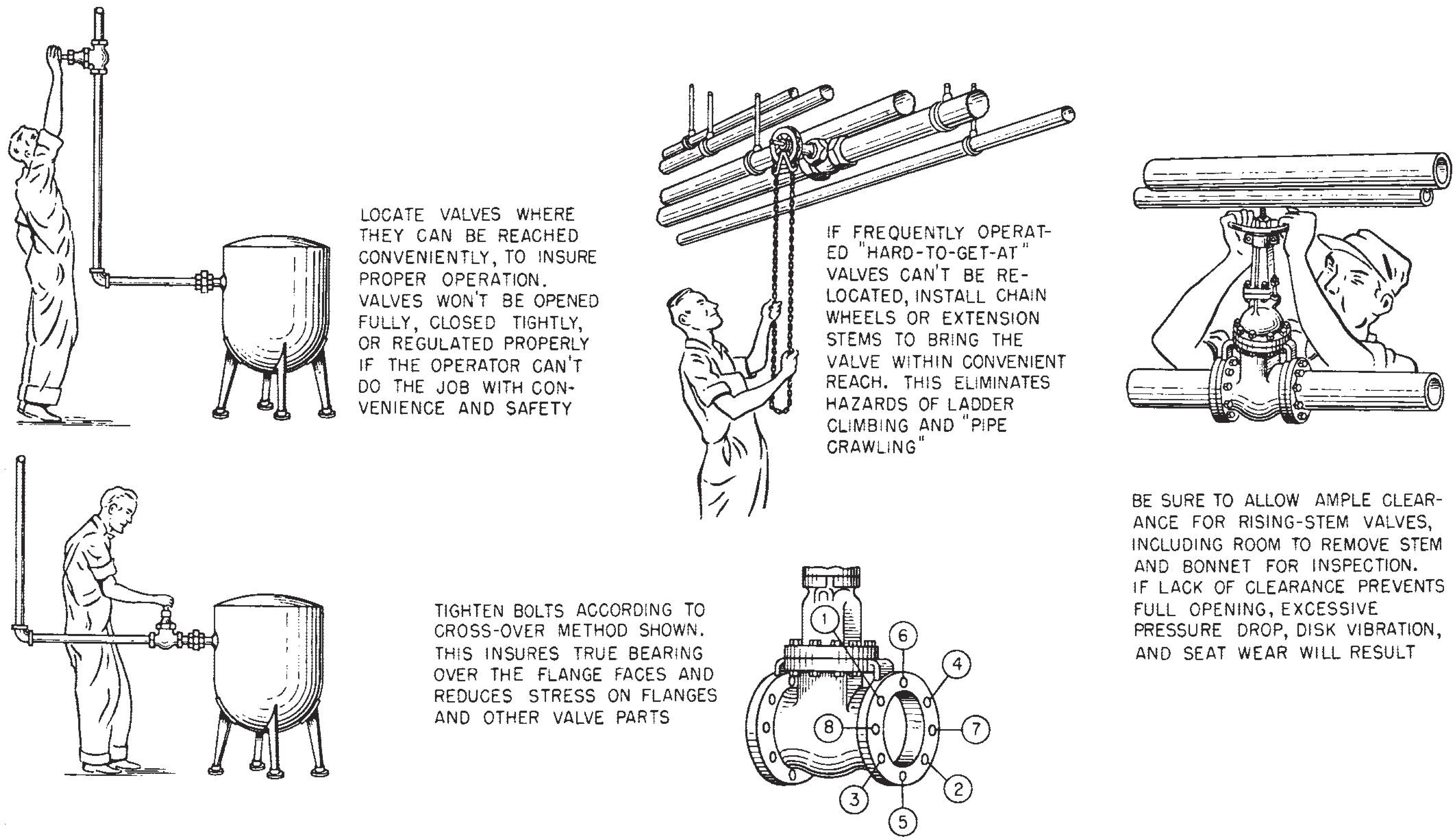


FIGURE 9.9 Long valve life begins with proper installation

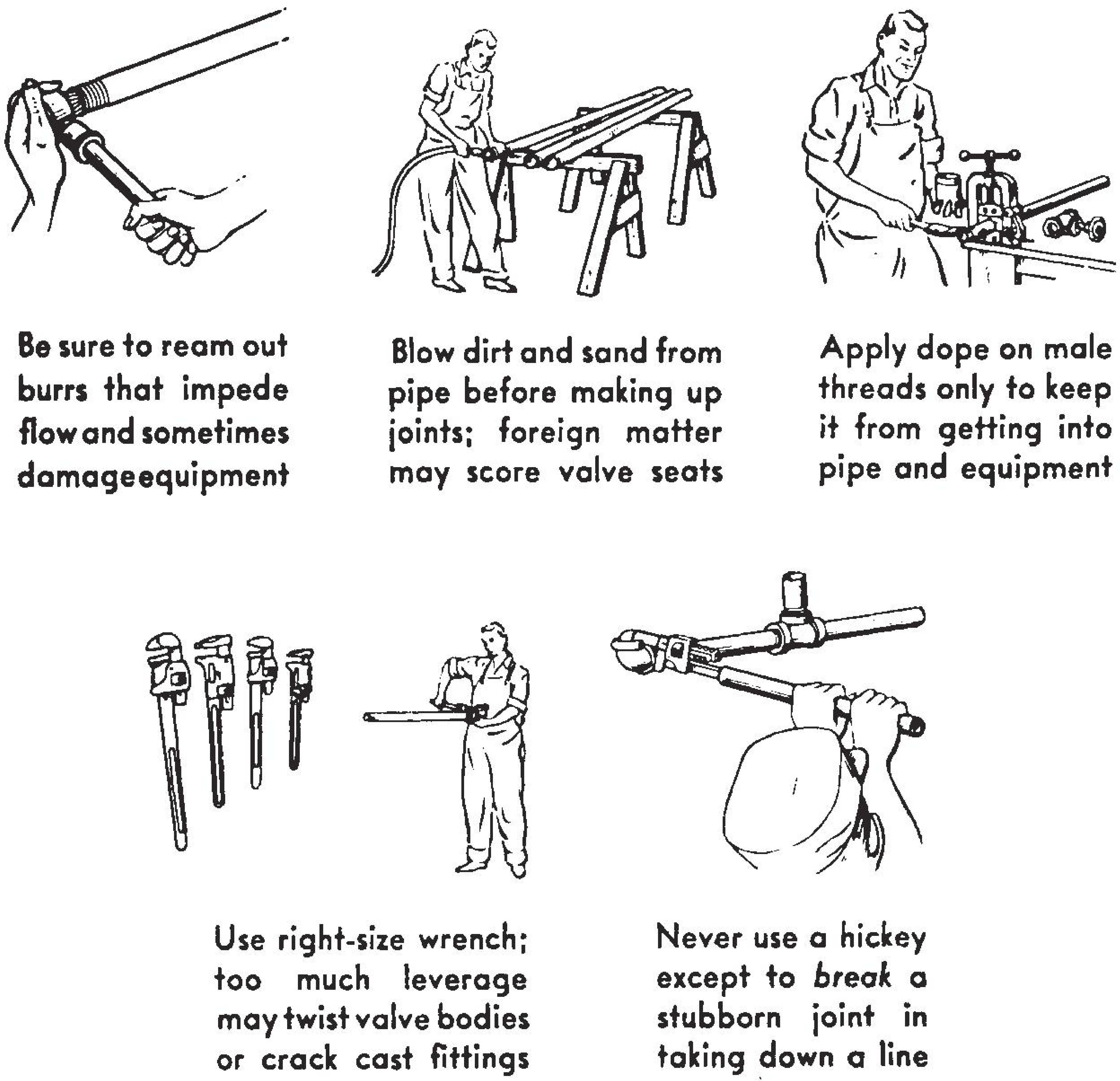


FIGURE 9.10 Suggestions for pipe fitters.

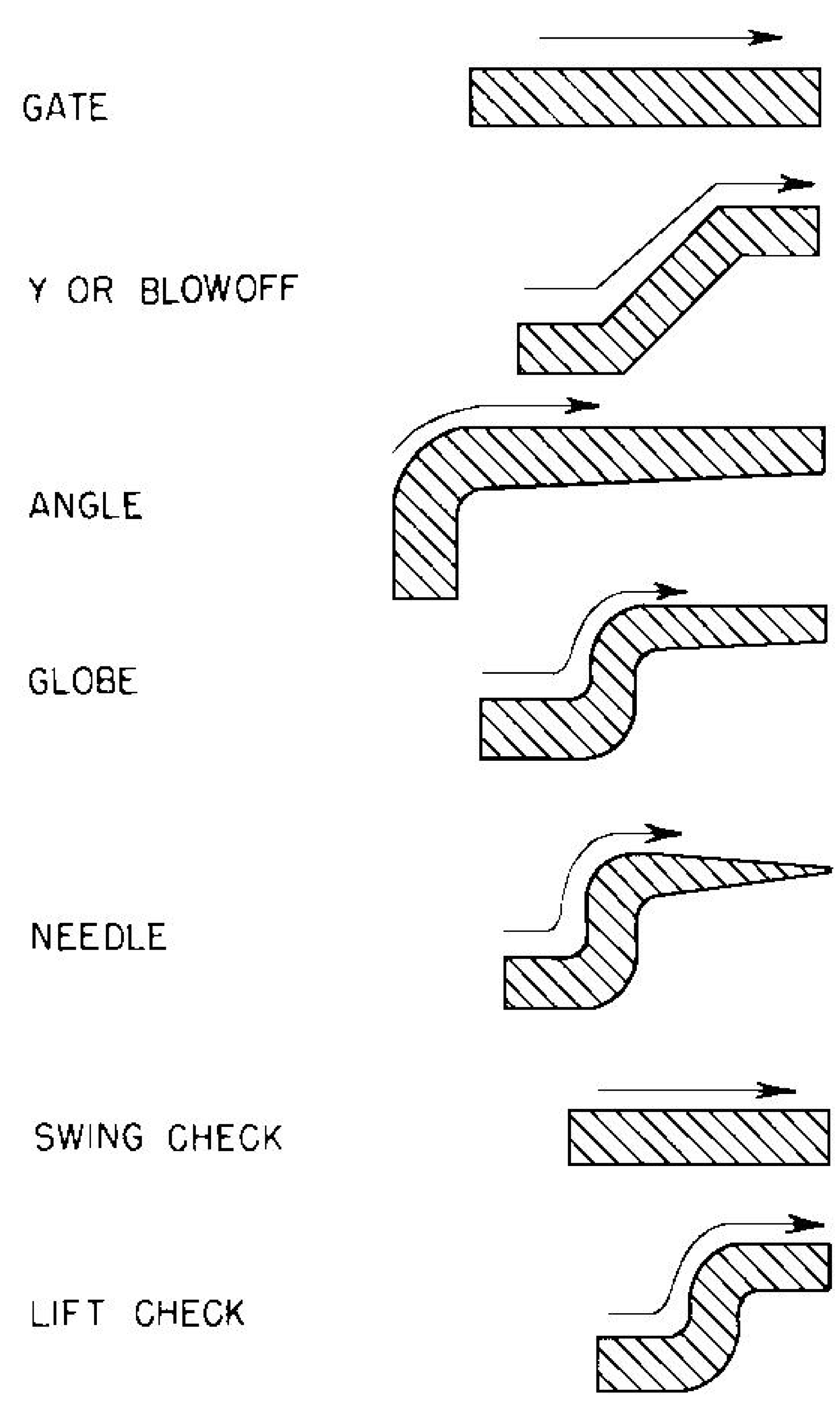


FIGURE 9.11 Valve flow chart.

Beware of valves just within fingertip reach of a normal man; rather than hunt for something to stand on, he'll stretch just enough to reach the handwheel. The valve cannot be closed tight, and leakage will develop. Always allow sufficient clearance for rising stems and for removing the bonnet and stem. It is much easier to leave the valve in the line and remove only internal parts for inspection and cleaning. Providing easy access to all valves represents the first step to correct operation, regular inspection, and careful maintenance. Good check-valve service depends to a large extent on meeting special needs. Correct position is most important; install swing and tilt checks with the pin horizontal and so that gravity will close the disk; place lift checks so that the lift is vertical.

Diaphragm-operated control valves present problems connected with the diaphragm as well as with the inner valve. Inverted installation on oil or chemical lines exposes the diaphragm to leakage through the stem packing. Heat from adjacent steam lines may cause early deterioration of the rubber. On steam-operated valves, protect the diaphragm against steam contact by installing a water leg or accumulator and filling it with water before admitting steam. When using the regulated pressure as the actuating medium, connect the pilot line, controlled by a lock-shield globe valve, on the downstream side of the valve. Do not make the connection in an elbow or pipe bend, because erratic pressures occur at these points. Select as a connection point a straight section of pipe at least 10 ft from the valve. The globe valve permits throttling the pilot supply to smooth out pressure pulsations, and the lock shield protects against tampering.

Regulating-valve bodies, marked as to flow direction, must be inserted in the line to conform with the marking (Fig. 9.12). A correctly chosen valve will not necessarily match the pipe size. For the connections, reducing flanges or bushings save space, but bell or venturi reducers give better flow conditions. Install a strainer ahead of the valve to remove foreign matter and protect the seats. Impingement noises can be minimized by installing the valve in a straight pipe run. Do not install an elbow or bend immediately downstream of the valve. To facilitate the regular inspection regulating valves deserve, connect a throttling globe valve as a bypass.

Check every valve installation for ways to simplify maintenance. When applying pipe insulation, end the permanent covering at bonnet bolts. Use a removable section to cover the bonnet; otherwise internal inspection may be neglected because of hesitation in breaking a perfect-looking insulated joint. After the system has been heated to operating temperature, tighten all body and bonnet bolts and adjust the stem packing.

Valve Maintenance. Neglecting a valve until it must be replaced or fitted with new parts wastes expensive materials; frequent and regular inspections uncover leaks and reveal other conditions, such as corrosion, incrustation, and wiredrawing, that mean future trouble unless corrected. Caught early, these defects can be repaired without major difficulty or expense; allowed to go unattended, they may require expensive parts and materials and can cause production shutdowns. A good maintenance program includes correct operation, regular and systematic inspection, proper lubrication of all rotating or sliding parts, replacement of stem packing when leakage or excessive friction develops, and refacing leaking seats and disks. Valve parts such as stem threads, thrust washers, and disk-spacing wedges or cams must be kept free of corrosion, incrustation, or foreign materials and must be adequately lubricated as recommended by the manufacturers. Plug cocks, with their large metal surfaces, require frequent lubrication to prevent galling and seizing of the sliding parts.

Packing. Proper maintenance of packing and correct adjustment of packing nuts are essential to satisfactory stem life and good valve performance. New packing, impregnated with graphite, lubricates the stem; after this lubricant disappears, friction between the stem and packing increases. If the packing nut must be tightened to a point where it is difficult to turn the stem, the packing has become dry and hard or is otherwise unsuitable for the service. In either case it should be discarded; it imposes an additional burden that will rapidly shorten stem-thread life. Excessive packing compression causes uncertainty as to whether or not the valve is fully seated. As a result, operators frequently seat valves tighter than they need be. This excessive closing effort, often applied with a wrench, injures the stem threads.

Stem packing that has been subjected to high-temperature steam and then allowed to cool often leaks a small amount when the line goes into operation again. Expansion and contraction of bonnet, stem, packing, gland, and packing nut cause this condition. It does not necessarily call for adjustment of the packing nut or gland; as soon as the valve becomes hot it will, in most cases, stop leaking.

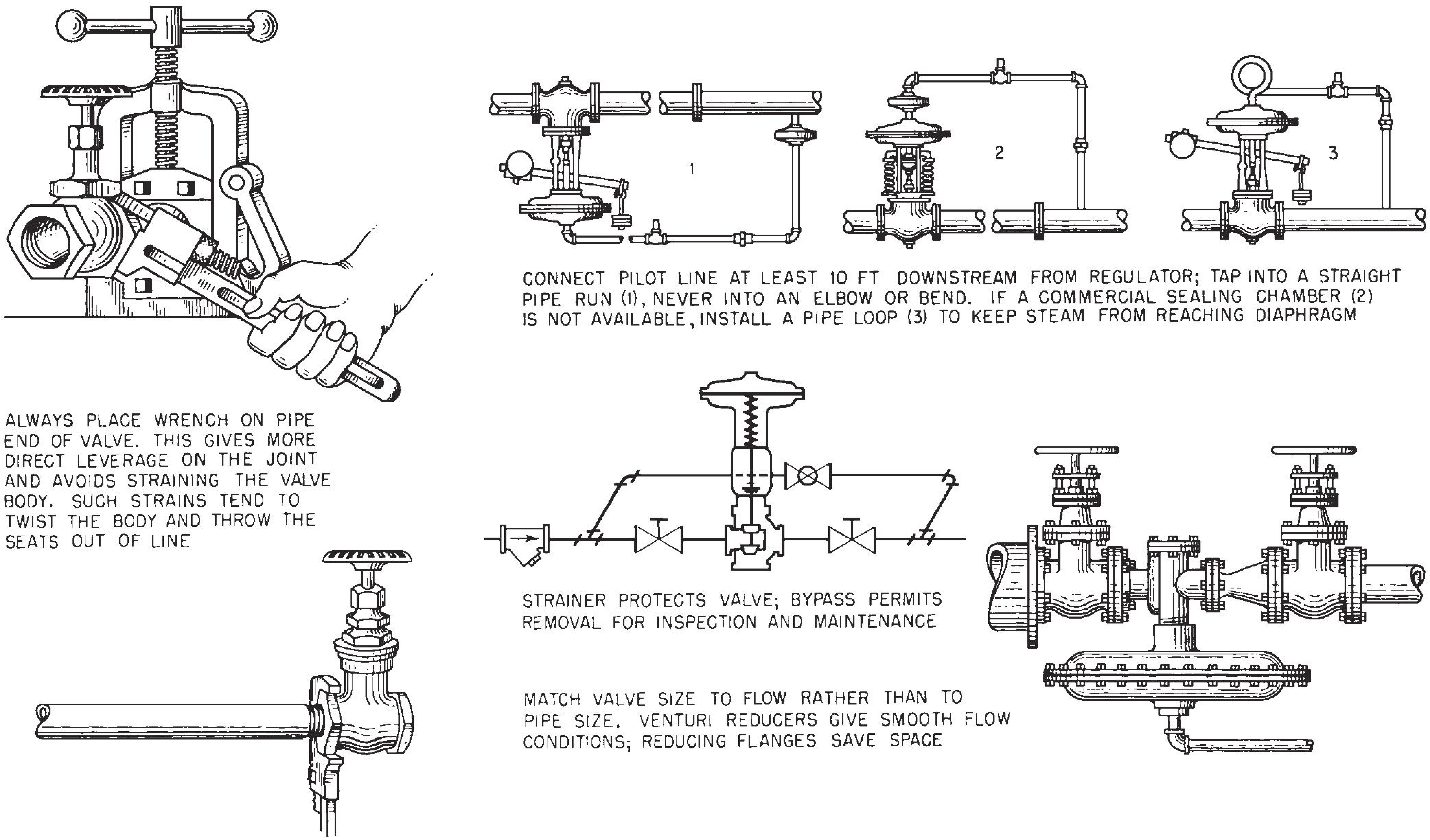


FIGURE 9.12 Good regulator hookups.

Packing maintenance is particularly important on automatically operated valves; excessive friction causes erratic movement. When lubricators are provided, keep the stem and packing adequately lubricated. Use stem packing that fits; preformed rings enter easily and draw up evenly. When winding coil packing spirally around the valve stem, force it to the outer edge of the stuffing box instead of wrapping it tightly to the stem. After adding the maximum number of rings, draw up the gland evenly, with a wrench, until the packing is forced into a snug position. Then slack off on the gland and make the nut finger-tight. A good valve stem, sufficient packing of the right kind, and a finger-tight gland will hold all moderate pressures. Higher pressures require a tighter gland to prevent the applied pressure from getting in under the packing.

Many valves contain a back seat closing off the packing gland against pressure, when the valve is open. This arrangement permits repacking the stem with the valve under pressure. Be certain a valve has this feature before attempting to repack under pressure. Maintenance and inspection involve frequent dismantling and reassembling of valves. Knowing the best way to do these operations simplifies the job and avoids damage to valves from improper handling. Before starting to remove a bonnet, open the valve so that no bending stress is placed on the stem during removal. Likewise, put the stem in the "open" position before replacing a bonnet. U-clamp gate-valve bodies have been split by tightening the bonnet joint with the wedge in its extreme closed position; union-bonnet gate-valve seats have been sprung apart by wedging action in the same manner.

Bolt Tension. On valves designed for high-temperature high-pressure service, bonnet bolts are usually tightened until a known tension is imposed on the bolt. Before loosening nuts, clean the bolt ends and measure the bolt length with a micrometer. Keep a record of individual bolt lengths, and elongate them the same amount when reassembling the valve, making sure the valve and bolt temperatures are the same as before. Always draw up body bolts evenly until the bonnet is true and square with the body. Most actual maintenance and repair operations are concerned with keeping seats and disks in leakproof condition. The specific methods used depend on the valve construction, condition of seats and disks, and equipment available. Modern methods of building up metallic surfaces (hard soldering, brazing, or welding) now offer means of salvaging valves with badly eroded seats and disks. Such building up makes repair possible without removal of the parent metal, greatly extending valve life.

Build up bronze seats with hard solder or bronze rod; alloy-steel seats for temperatures below 750°F can be repaired by brazing, which, although not as resistant as the original metal, will give good service. Building up alloy trim with supposedly identical metals can easily lead to trouble unless complete information is available as to the composition and hardness of the parent metal. Before building up disks on automatic valves for service on high pressures and temperatures, consult the manufacturer, because any major change in seat or disk contour may cause serious operating difficulties. If facilities for building up special trim metal are not available, the valve can be returned to the factory, although this practice may interfere with plant production. When it is absolutely necessary to buy new parts, buy seats and disks in pairs so that only a minimum amount of grinding is needed. Salvage all good parts of damaged valves for use as spare parts to rebuild other valves when they become damaged.

Seats and disks can be ground or refaced in many ways. The procedure for grinding seats and disks in union-bonnet globe valves is, perhaps, the simplest (Fig. 9.13). It is necessary only to pin the stem and disk together and use the bonnet as a guide for lapping the disk against the seat. Screwed bonnets cannot be used as guides for grinding; this job requires a grinding kit or a drill press. If a drill press is available, remove the stem from the bonnet and insert it in the drill chuck. Clamp the valve body in the drill-press vise; level on the top edge of the body-bonnet joint which parallels the seat surfaces. Pin the disk and stem, apply compound, and grind with the drill press at low speed and light spindle pressure.

Check Valves. Check-valve disks can be lapped against the seats; the disk usually contains a slot for a screwdriver to apply the turning movement. When lapping stainless iron disks and seats against each other, mix white lead and oil with the grinding compound to provide lubrication; otherwise the metal will drag and ruin the surfaces. The use of a grinding compound with small grain size reduces the tendency to gall. Grind with light strokes, lifting the disk frequently to a new position and cleaning the surfaces often. Stellited seats and disks can be ground in the same manner as that recommended for stainless iron, except that in some cases it may be necessary to use silicon-carbide grit.

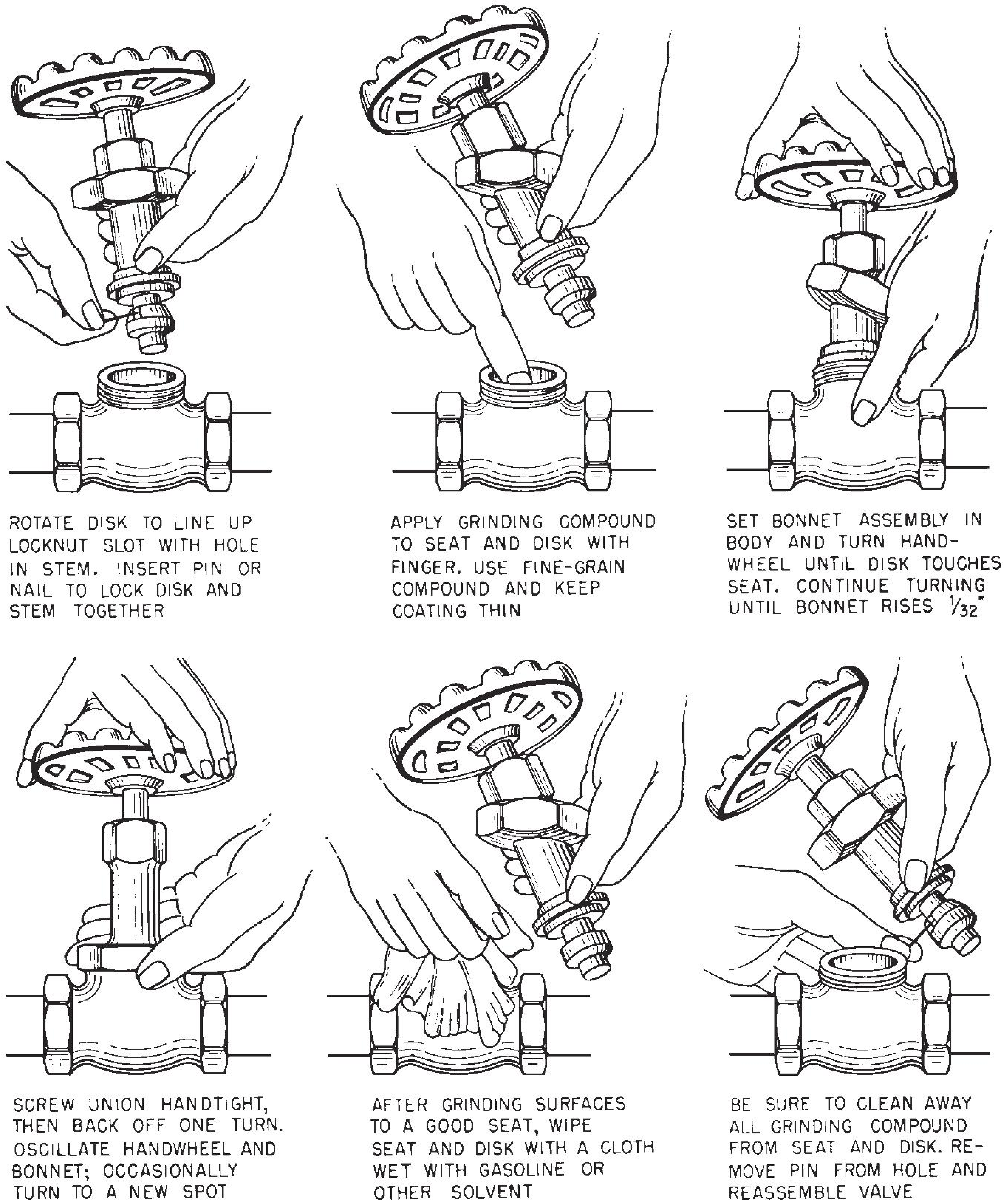


FIGURE 9.13 Regrinding globe-valve seats.

Balanced-pressure double-seat regulating valves require particular care in grinding. Both seats must be established at the same time. Watch the disks to see which seat touches first; put compound on this disk with just a trace on the other. Grind until both seats are established uniformly. To produce a true seat under operating conditions, provide a steam connection to heat the body and stem to operating temperature, before grinding.

Machining. On globe valves which are badly worn, or where eroded areas have been built up, machining eliminates excessive lapping. Seats can be machined in a lathe, or ground with an emery-cloth-covered metal disk mounted in a drill press. A similar grinding disk, mounted on a motor-driven flexible shaft, can also be used on small valves. Repeated machining or grinding may reduce seat thickness dangerously. On integral-seat globe valves too small to insert soldering or brazing equipment, the seat opening can be reamed and threaded to take a renewable seat ring, which can be made up from material available in the plant shop. While it is desirable to replace the seat with the same

metal, a different one can be used in an emergency. It is not advisable to use stainless iron of the same hardness for both seat and disk unless the valve is to handle oil.

Seat Rings. Repeated refacing of shoulder-design seat rings reduces the shoulder thickness to a point where metal contact pressure against the underside causes concaving of the outer or seating surface. The remedy lies in replacing the ring or increasing its thickness by welded overlays. Leakage past seat-ring threads must be repaired immediately; it can be done by welding and rethreading or by reaming and threading to take an oversize ring. If the damage is too great for these remedies, weld the ring solidly in the valve body.

Good service from composition globe-valve disks depends on care. When one side of such a disk is eroded, either machine it to a new face or reverse it and use the opposite side. Disk stocks can be stretched by using substitutes such as leather, scrap rubber belting, or lead machined smooth. Although not ideal, these materials will give satisfactory service until new disks arrive.

Refacing. Refacing of gate-valve seats and disks usually requires machining in a lathe or grinding in a drill press, although disks with surfaces not too badly worn can be refaced with a sanding wheel or by hand grinding, and seats can be ground with a hand brace. Lathe machining offers no major difficulties for parallel-seat gate bodies, but wedge-gate bodies require cumbersome holders. The drill-press grinding method is convenient.

Parallel-seat disks may be chucked readily for lathe refacing; a taper block to fit the faceplate helps with wedge disks. The drill-press grinding method can be used on wedges as well as seats. Special holding jigs, milled out to hold the wedge snug and level and keep it from turning, offer maximum convenience, but a jig is required for each valve size. One flat tapered plate, with clamps, will serve for many sizes and can be used on either a drill press or a lathe. Wedge disks can also be refaced by holding the surface against a motor-driven sanding or grinding disk. Keep the wedge centered on the disk and hold it with uniform pressure, or uneven grinding will ruin the taper. Hand grinding on an emery-cloth-covered flat surface proves satisfactory where only light scratches or machine marks need be removed.

Reseating Kits. Most of the methods described so far require that the valve be removed from the line. Valve reseating kits are available for refacing seat rings without removing the body from the line. This eliminates the need for breaking pipe joints, reduces possibilities of leaks, and saves time and labor.

Valves subjected to corrosive conditions soon become covered with "barnacles" which build up around seat and disk rings. If allowed to increase to any great extent, the barnacles soon creep over the seat edges and prevent the valve from closing tight. Cleaning the valve with a sandblast and applying a good paint or metal-spray coating on the areas around the seats greatly retards this growth. Timely repairs make valves last longer and save metal.

Hard-facing. This is a useful maintenance tool for protecting steel valve parts against severe abrasive wear and wiredrawing action. Cobalt-chromium-tungsten alloys (stellite) retain their hardness at red heat, making them particularly suited to surfaces in friction and to parts exposed to high temperature. Before applying hard-facing alloy, prepare the part by grooving the surface to a depth of from 3/32 to 1/8 in. (Fig. 9.14), leaving a ridge on each side. Round off all sharp corners, because sharp edges melt easily and interalloying between the base metal and the hard-facing alloy might occur. Such "dilution" results in decreased wear resistance and frequently causes blowholes. If the seat or part is too narrow for grooving, machine it flat and round off the edges. After grooving, clean the surface thoroughly, removing all dirt, scale, and grease.

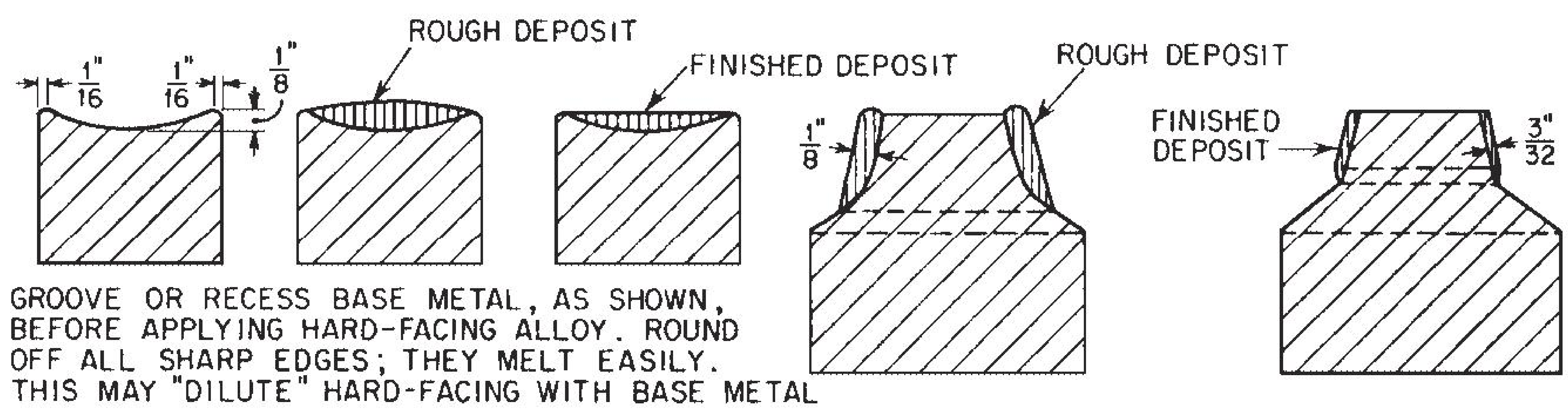


FIGURE 9.14 Correct preparation means good hard-facing.

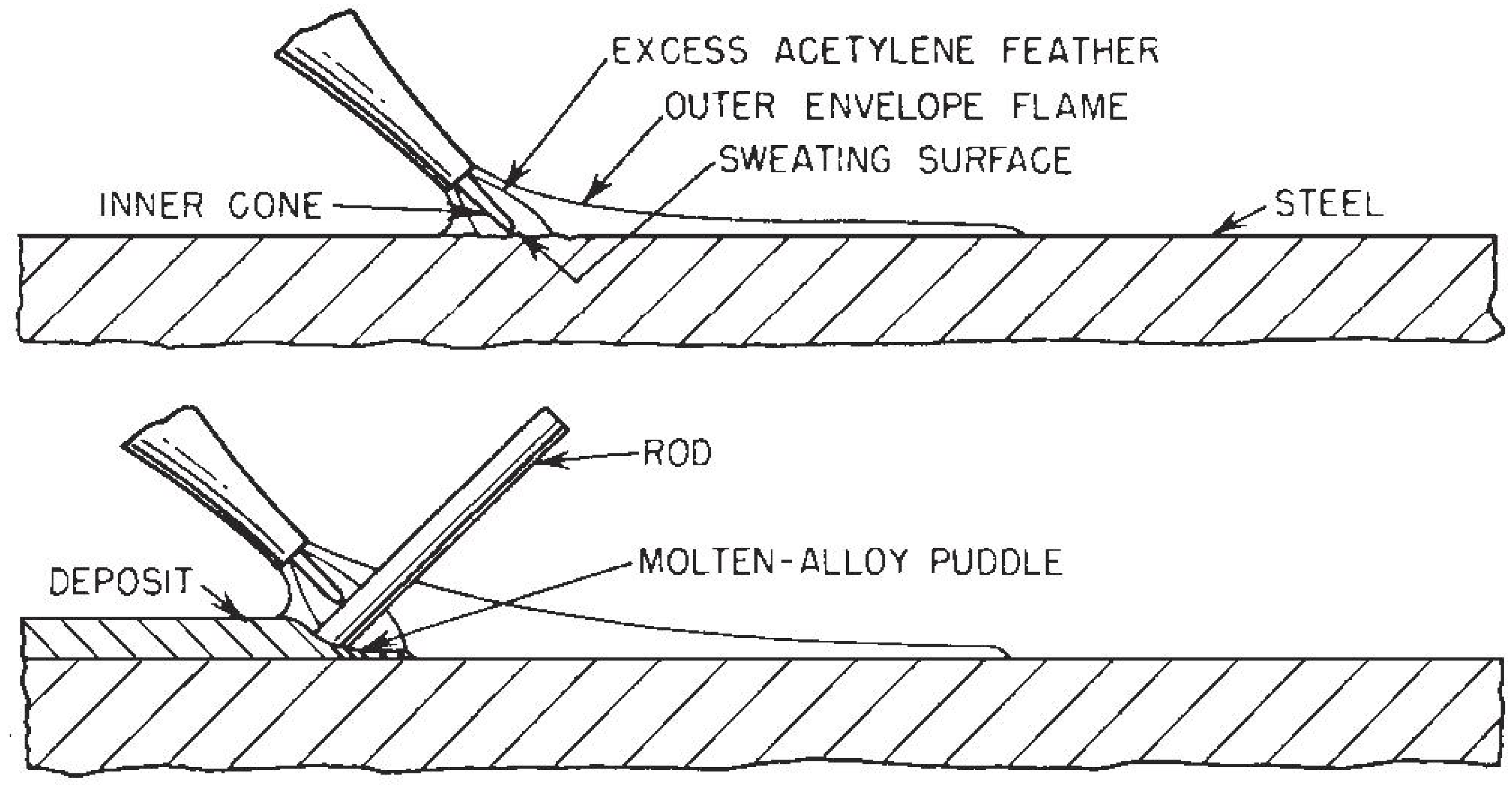


FIGURE 9.15 Sweating work surface with a torch.

If the part is small, say under 3 in. in diameter, and the welding flame is large enough to keep it red hot during welding, no furnace preheating is necessary; preliminary heating can be done with the torch. Larger parts require preheating with several torches or in a temporary furnace. Raise temperature slowly to 800 to 1200°F, or just under the point where steel begins to scale. This is a faint red heat, just visible in a dark room.

Maintain Heat. If possible deposit the alloy while the valve part is in the furnace; when this is impractical, reheat whenever the part cools to 600°F. Fixtures can be purchased or made that will rotate the work during the hard-facing. Such a device is not absolutely necessary; a helper can turn the part.

Hard-facing requires an excess acetylene flame (Fig. 9.15). Adjust the acetylene feather length, measured from the blowpipe tip, to three times the inner-cone length, measured from the welding-tip end. This flame prepares steel by melting an extremely thin surface layer, giving the steel a watery, glazed appearance called "sweating." This sweating, produced only by an excess acetylene flame, is necessary for successful hard-facing on steel.

Torch Angle. Hold the torch to direct the flame at a 30 to 60° angle to the surface, with the inner-cone tip about $^1/_8$ in. from the steel. Keep this position until the steel under the flame suddenly glazes. The extent of sweating area varies with size of welding tip, but for a medium tip, steel will sweat about $^1/_4$ in. around flame. Withdraw the torch slightly, and bring the welding rod between the inner cone of flame and the steel surface. The inner-cone tip should almost touch the rod, and the rod should lightly touch the sweating area. The melting rod forms a puddle on the steel. If the first few drops foam or bubble, or do not spread evenly, the steel is too cold and should be brought to the recommended temperature.

Some steels foam slightly when brought to sweating heat. When this occurs, do not deposit metal until foaming stops. If deposit is started before foaming is noticed, direct the torch at the foaming spot and agitate the molten metal with flame until foaming stops. Depositing metal during foaming causes blowholes and poor results. To spread molten alloy over the area, remove the rod from the flame and direct the flame into the puddle. Return the rod and melt off more alloy as required. Now direct the flame so that it plays partly on the edge of the puddle and partly on the adjoining steel surface. As steel approaches sweating heat, a puddle of hard-facing alloys spreads. As it spreads, bring the rod quickly into the flame again to add more metal as needed. If any dirt or scale appears on the steel or in the puddle, float it to the surface with the flame or dislodge it with the end of the welding rod.

With a little practice the right amount of alloy can be added to make the desired thickness. It is better to do this in one operation than to go back over the entire job to add another layer. During the operation, move the flame back to melt a thin surface layer of the deposit, to smooth out high spots as the work progresses. Do this quickly, without letting the front edge of the puddle solidify and without interrupting the steady forward travel of the work. After completing the deposit, use the flame to smooth out remaining rough surfaces. On this second pass, take care to melt only the hard-facing surface and not the base metal. This avoids bringing iron from the base metal to the hard-facing deposit.

Prevent Cracks. When the deposit reaches desired size and thickness, remove the flame slowly to prevent formation of shrinkage cracks and blowholes. If these occur, remelt the deposit and remove particles of scale from the pool. If holes still show, grind the alloy deposit down to steel, heat the area with flame, gradually, and deposit additional metal. Make sure that no slag, dirt, or scale is covered or embedded in the deposit to cause pinholes.

Slow cooling is absolutely essential to produce a deposit free from cracks and internal stresses. Parts showing a strong tendency to crack, such as large gate-valve wedges and seat rings, or parts on which the deposit is circular or large in area, should be returned to the preheating furnace while still hot from welding. Bring them slowly to a low red heat; then let them cool in the furnace. If a furnace is not available, place the part in dry powdered lime, ashes, or other insulating material, so that at least 2 in. of material covers and protects every point of the part.

Some alloy steels used for valve trim require heat treatment in order to maintain corrosion resistance. When this is necessary, follow the steel manufacturer's instructions, with but one exception: never cool hard-faced parts by quenching in water or in an air blast. This will set up strains and cause cracks in hard-facing. If quenching is considered necessary, use only oil. After the hard-faced part has cooled, excess metal must be removed. This can be done by grinding or by machining with a tungsten-carbide tool.

Traps. With these, too, good operation and low-cost maintenance starts with proper installation. For example, hard-to-get-at traps will be neglected; easy access encourages regular inspection. If a trap is exposed to low temperatures, protect it against freezing. Install impulse or thermostatic drains in the trap or piping inlet to release all water when pressure is shut off and the condensate temperature falls.

Trap location directly affects operation. Wherever physically possible, install traps below equipment to be drained, so condensate can flow by gravity. Avoid U bends or water seals; they obstruct free flow and cause "steam binding." A slug of water flowing toward the trap immediately after discharge lies in the pocket until steam, remaining beyond the pocket and in the trap chamber, condenses. Where traps must be installed above equipment to be drained, install a check valve and water seal or U bend in the connective piping. The check prevents backflow and loss of prime and also prevents back drainage from a common-return header, or entrance of air when the unit is shut off. The water seal, although acting as an obstruction, serves as a sump for condensate collection, allowing it to be carried to the trap in slugs. Without the seal it would be possible for the bottom coil to remain partly filled with condensate.

Where traps discharge into common lines and connections do not require a check value in the inlet line, a check in the discharge prevents backflow from other units or drainage back to an idle unit when the trap has an individual overhead discharge line. Even when condensate flows vertically downward to a trap, a heavy rush of condensate chokes the line, prevents backward escape of trapped steam, and requires time for steam to escape or condense before water enters the trap. Although obstructing bends installed on steam lines only delay trap action, more serious trouble occurs on air lines unless the trap vents back to the vessel being drained. Such vent lines can be used only when the trap stands below the drained equipment.

If allowed to enter traps, pipe scale and sediment prevent tight seating and cause blow-through. Cleaning the pipe before installation fails to protect fully because temperature changes and flow loosen other particles. Install a strainer ahead of the trap, or if this is not possible, fabricate a "dirt leg" from pipe, to act as a catch pocket (Fig. 9.16). Uniform piping connections help in removing or exchanging traps for inspection and repair. Test valves and a tee in the discharge from each trap facilitate checking trap action. Installing a bypass around the trap permits adequate drainage and removal for repair when no spares are available.

Many traps include a valve, an operating device (bucket, float, bellows, etc.) and, if necessary, a linkage and bearings between valve and operator. Trap maintenance usually involves cleaning to remove foreign matter that might interfere with valve or linkage action, reseating valves when necessary, removing lost motion from linkage, and renewing the body gaskets. Moving parts located inside the body, in contact with moisture, offer lubrication difficulties and may show considerable wear. Excessive wear and lost motion, if allowed to continue, prevent positive operation and may reach the stage where the valve no longer seats.

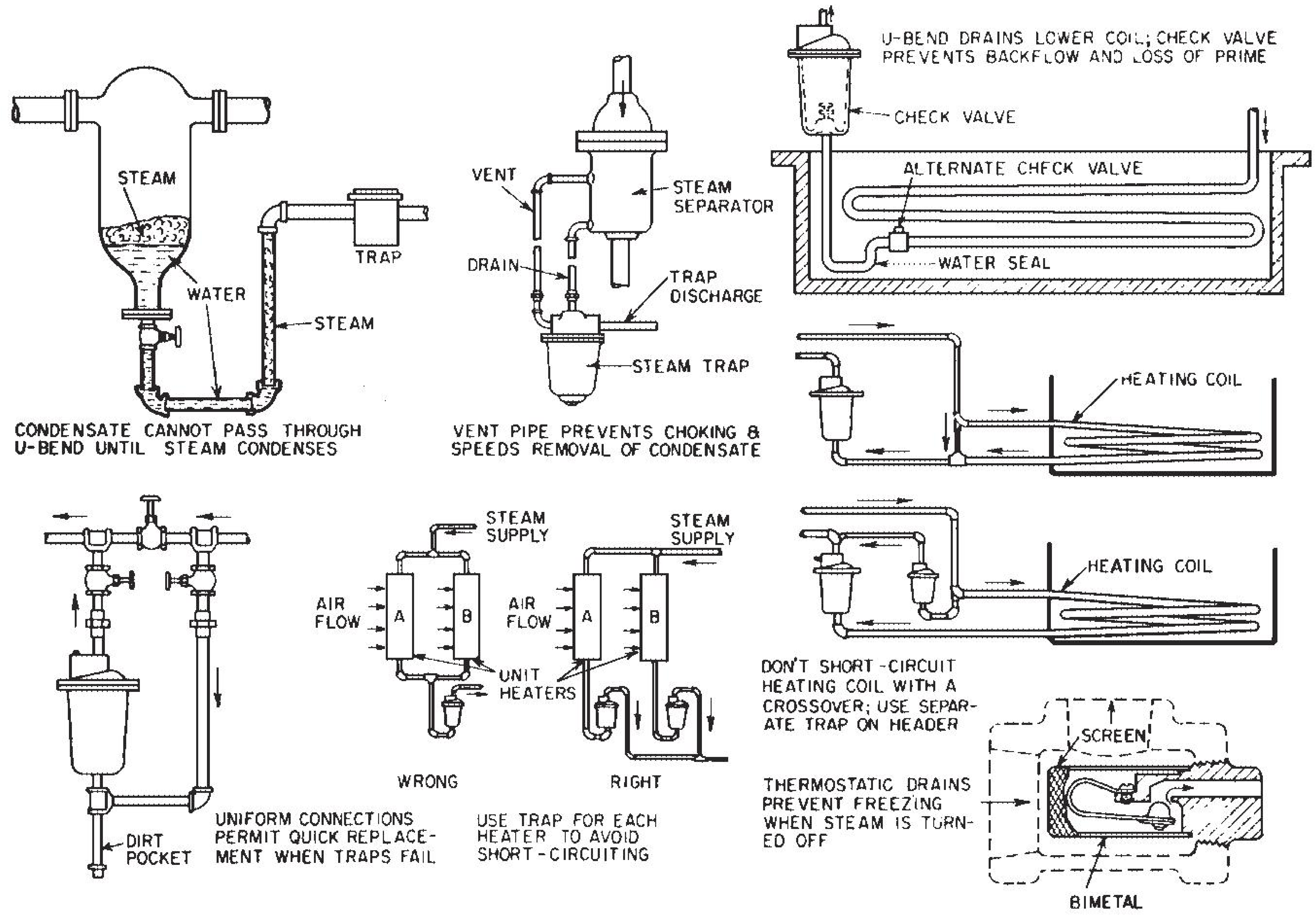


FIGURE 9.16 Pointers for trap installation.

Normal wear of some valves gradually enlarges the seat area at the point of disk contact. This increases the surface acted on by differential pressure across the valve, making it more difficult to operate. The area may increase to such an extent that the operating device is no longer powerful enough to move the valve. When this happens, the valve must be refaced to regain the original area. Whenever a trap fails to operate, and the reason is not readily apparent, observe the trap discharge by opening a test cock and breaking the discharge connection. Live steam usually indicates a leaking valve; it may be caused by the trap losing prime. Failure to discharge can be caused by a leaking bypass valve, inlet piping or trap obstructed by sediment and pipe scale, return line too small, or an obstructed outlet. Figure 9.17 shows a number of steps in trap care.

Valve leakage represents a common cause of trouble; worn seats are not so much to blame as are particles that prevent tight closure. Other trap troubles include rusting and sticking of the mechanism, lost motion in linkage, gasket and connection leaks, float leaks, and bent lever arms. Wear in levers and pins of a continuous-discharge trap causes intermittent operation. For most of these difficulties, simple mechanical remedies suffice, once the cause of the trouble is spotted. When valves do not seat properly yet appear to be in good condition, check linkage and stem length. Repeated regrinding may have shortened the stem. Make adjustments of stem length at or near full operating temperature.

There are numerous ways of checking trap operation. A slight temperature difference between inlet and outlet indicates a working trap; no difference indicates a leaking trap; a large change in temperature indicates no condensate passing. Intermittent-discharge traps produce a light clicking sound at each operation; constant-discharge traps can be checked with a listening rod or a stethoscope applied to the trap body. When visible discharge is not satisfactory evidence, passing the discharge into a vessel of water forms a positive test. Weigh the original quantity of water and check its temperature. After discharge, weigh the water again and check its temperature. Heat given up by trap discharge in falling to final temperature equals heat gained by original water quantity rising to same temperature. The chart in Fig. 9.18 simplifies computations, eliminating the use of steam tables. Table 9.2 gives trap troubleshooting data.

Pipe Supports. An ideal piping system would float like a layer of logs on a smooth pond, each part self-supported and imposing no stress on any other part. All elements in the system would hold their

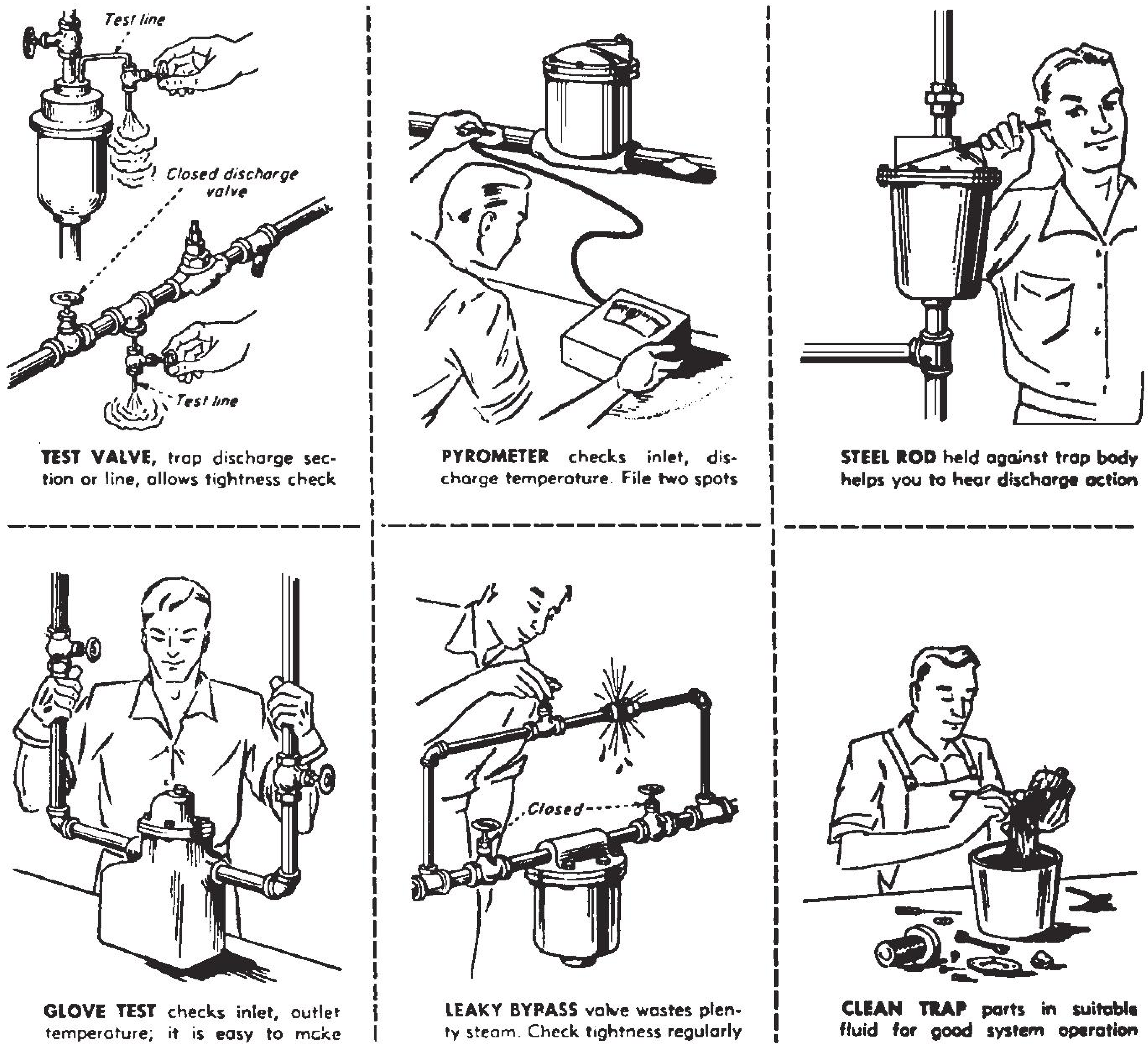


FIGURE 9.17 Trap maintenance tests.

correct relative positions and alignment despite thermal expansion and contraction. Actual well-built systems approach this theoretical ideal by the intelligent use of anchors to fix certain points of the system, and expansion joints, supports, and guides to combine support with free movement for all the rest of the piping in the system. Without being an expert in the mathematical design of piping systems, the maintenance engineer should understand the duty to be expected of each part of the system under his care so he can check whether that job is actually being performed as it should.

First take the system as it stands, cold or at some fixed temperature. Anchors should securely lock the anchored points in the piping to heavy steelwork or other dependable footing. Between each pair of anchors should be an expansion joint or bend designed to absorb all possible movement from temperature differences. Pipe supports should be spaced closely enough to prevent undue sag in the span. All steam and air lines should be properly pitched for condensate drainage and checked to make certain that sag does not bring the center point of any span below its lower support, thus forming a condensation pocket leading to water hammer and other troubles.

Check each hanger or other support to make sure that it carries its share of the load; that pipes track truly on fitted rollers or other guides; and that supports and their attachments are amply strong for the load and set to carry the weight, yet permit free pipe movement in the direction of expansion. To avoid trouble these conditions should be met whether the line is hot or cold. To make a single set of adjustments serve for both extremes of temperature is a most difficult job for both the designer and the maintenance engineer. One requirement is an understanding of thermal expansion, and here are the main points:

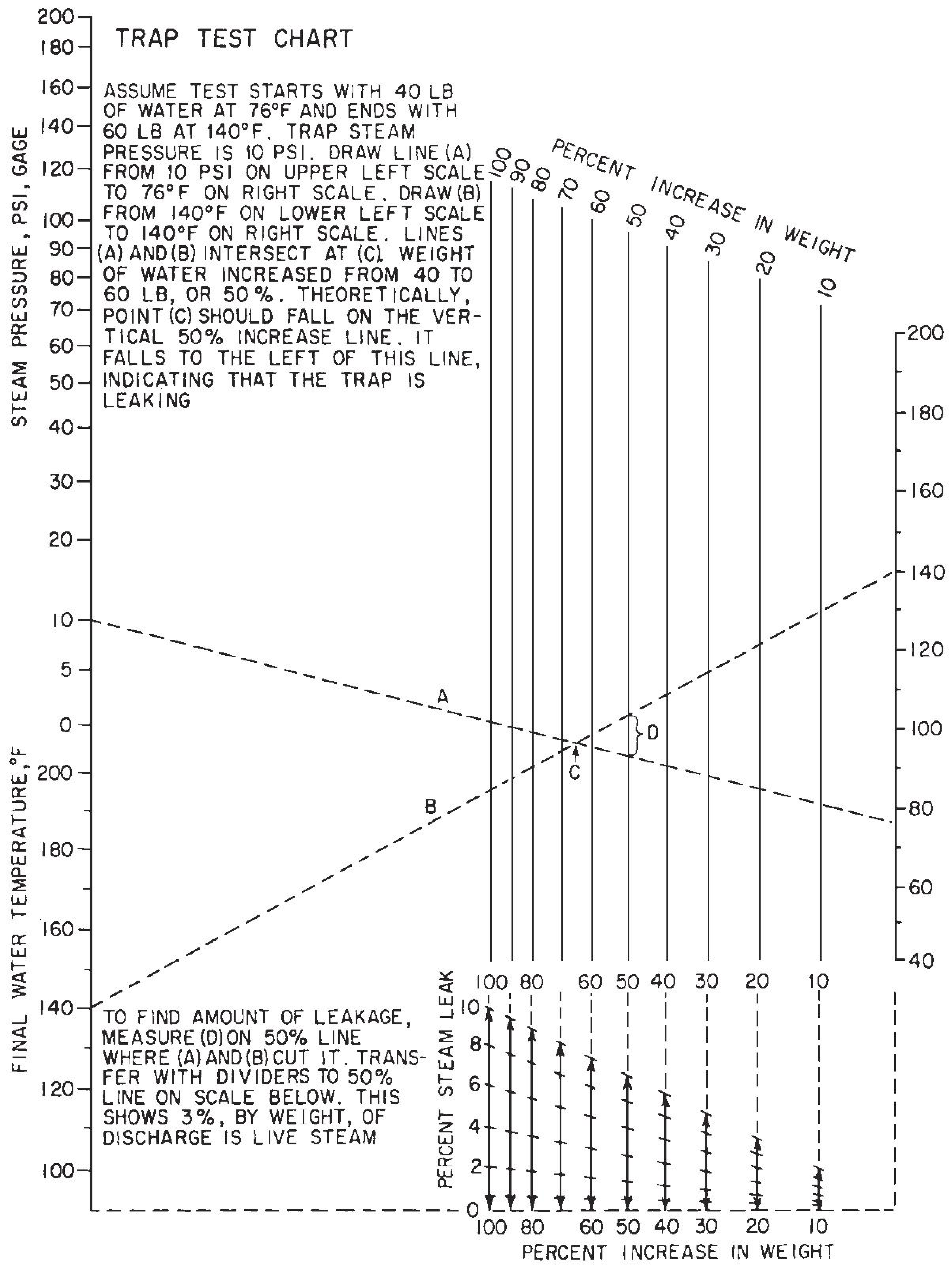


FIGURE 9.18 Trap test chart.

When an unrestrained steel body is uniformly heated, no forces whatever are set up, either internal or external. The body expands gently and proportionately in all directions. Thus, if a piping system is completely unrestrained except that the weight is carried at all points by fully "floating" supports, heating to a higher temperature will produce no forces whatever, nor any change in the proportions of the layout. All dimensions will be slightly increased, as if the new layout were a slightly enlarged photograph of the original. The coefficient of expansion for any grade of iron or steel is approximately 0.000007 per degree F. This simply means that the expansion is 7 parts per million per degree. For example, if the temperature of a steel pipe is raised by 400°F, its length will increase by about $400 \times 7 = 2800$ parts per million, or about 0.28 in. per 100 in. The coefficient of expansion varies somewhat with the actual temperature; so expansions should be taken directly from Table 9.3 or similar data if accurate results are desired.

With either corrugated-metal or well-lubricated sliding expansion joints, the forces to be handled by the anchors are substantial, but much less than with the expansion pipe bends commonly used for

TABLE 9.2 Troubleshooting Chart for Steam Traps

Trouble	Possible cause and cure
Trap doesn't discharge	1. Steam pressure too high, pressure regulating out of order, boiler pressure gage reads low, steam pressure raised without altering or adjusting trap. On the last item consult trap maker. He can supply parts for higher pressure or tell you how to adjust trap. 2. Plugged strainer, valve, or fitting ahead of trap; clean. 3. Internal parts of trap plugged with dirt or scale; take trap apart and clean. Fit strainer ahead of trap. 4. Bypass open or leaking; close or repair. 5. Internal parts damaged or broken; dismantle trap, repair.
Trap won't shut off	1. Trap too small for load; figure condensate quantity to be handled and put in correct-size trap. 2. Defective mechanism holds trap open; repair. 3. Larger condensate load from (*a*) boiler foaming or priming, leaky steam coils, kettles or other units, or (*b*) greater process load; find cause of increased condensate flow and cure, or install larger trap. *Note:* Traps made to discharge continuously won't show these symptoms. Instead, the condensate line to trap overloads; water backs up.
Trap blows steam	1. Open or leaky bypass valve; close or repair. 2. Trap has lost prime; check for sudden or frequent drops in steam pressure. 3. Dirt or scale in trap; take apart and clean. 4. Inverted bucket trap too large, blows out seal; use smaller orifice or replace with smaller trap.
Trap capacity suddenly falls	1. Inlet pressure too low; raise to trap rating, fit larger trap, change pressure parts or setting. 2. Back pressure too high; look for plugged return line, traps blowing steam into return, open bypass or plugged vent in return line. 3. Back pressure too low; raise
Condensate won't drain from system	1. System is air-bound; fit suitable vent or trap with larger air capacity to get rid of the air. 2. Steam pressure low; raise to the right value. 3. Condensate short-circuits; use a trap for each unit.
Not enough steam heat	1. Defective thermostatic elements in radiator traps; remove, test, and replace damaged elements. 2. Boiler priming; reduce boiler-water level. If boiler foams, check fires and feed with fresh water while blowing down boiler at quarter-minute intervals. 3. Scored or out-of-round valve seat in trap; grind seat or replace old trap body with new one. 4. Vacuum pump runs continuously; look for a cracked radiator, split return main, cracked pipe fitting, or a loose union connection. Or pump shaft's packing may leak. 5. Too much water hammer in system; check drip-trap size. Undersized drip traps can't handle all condensate formed during warm-up so hammering results. Fit larger trap if drip lines are clean and scale-free. Size for warm-up load, not for load with mains hot. 6. System run-down; older heating plants are sometimes trouble-some because a large number of trap elements are defective. Easiest cure is replacement of all thermostatic elements in the radiators. This is low-cost, sure.

TABLE 9.2 Troubleshooting Chart for Steam Traps (*continued*)

Trouble	Possible cause and cure
Traps freeze in winter	1. Discharge line has long horizontal run where water collects; make discharge line as short as possible and pitch away from trap. 2. Trap and piping not insulated; fit insulation to outdoor traps and piping connected to them.
Back flow in return line	1. Trap below return main doesn't have right fittings; use check valve and a water seal, or both, depending on what the trap maker recommends. 2. High-pressure traps discharge into a low-pressure return; flashing may cause high back pressure. Change piping to prevent return pressure from exceeding trap rating. 3. No cooling leg ahead of a thermostatic trap that drips a main; condensate may be too hot to allow trap to open right. Use a 4- to 6-ft cooling leg ahead of thermostatic traps on this service. Fit strainer in cooling leg to keep solids out of trap.

Courtesy of *Power* magazine.

TABLE 9.3 Thermal Expansion of Steam Pipe

Temp, °F	Cast iron pipe	Steel pipe	Wrought iron pipe	Copper pipe
−20	0	0	0	0
0	0.127	0.145	0.152	0.204
100	0.787	0.898	0.939	1.338
200	1.495	1.691	1.778	2.500
300	2.233	2.519	2.630	3.665
400	3.008	3.375	3.521	4.870
500	3.847	4.296	4.477	6.110
600	4.725	5.247	5.455	7.388
700	5.629	6.229	6.481	8.676
800	6.587	7.250	7.508	9.992
900	7.579	8.313	8.639	11.360
1000	8.617	9.421	9.776	12.741

Condensed from Crocker, "Piping Handbook," 5th ed., McGraw-Hill Book Company, New York, 1967.

the higher pressures. Figuring such bends is a rather mathematical branch of design engineering, but the maintenance engineer may be able to learn from the designer how much force the bend exerts for each inch of compression as the connected piping expands.

Expansion. When piping is heated, it moves straight out from the anchor by an amount proportional to the temperature rise and to the distance from the anchor. In an elaborate piping system many complications can arise. If a long horizontal run without an expansion joint ends in a riser, expansion will push the riser out of plumb unless the top end is so suspended that it can move out too. If the weight of a riser, when cold, is equally distributed over several rigid hangers, one above the other, heating the line will expand the pipe and thereby unload all upper hangers, shifting the entire load to the bottom hanger. Many other similar effects will be observed, and generally can be sized up on the spot by the exercise of commonsense mechanics.

Here are some practical points to remember: Supports for a horizontal pipe (Fig. 9.19) far from an anchor must allow ample roll, slide, or swing in the direction of the pipe expansion. If temperature changes cause any piping (whether horizontal or vertical) to rise and fall, such sections must be carried on properly designed spring supports. For large movements in a vertical direction, the springs

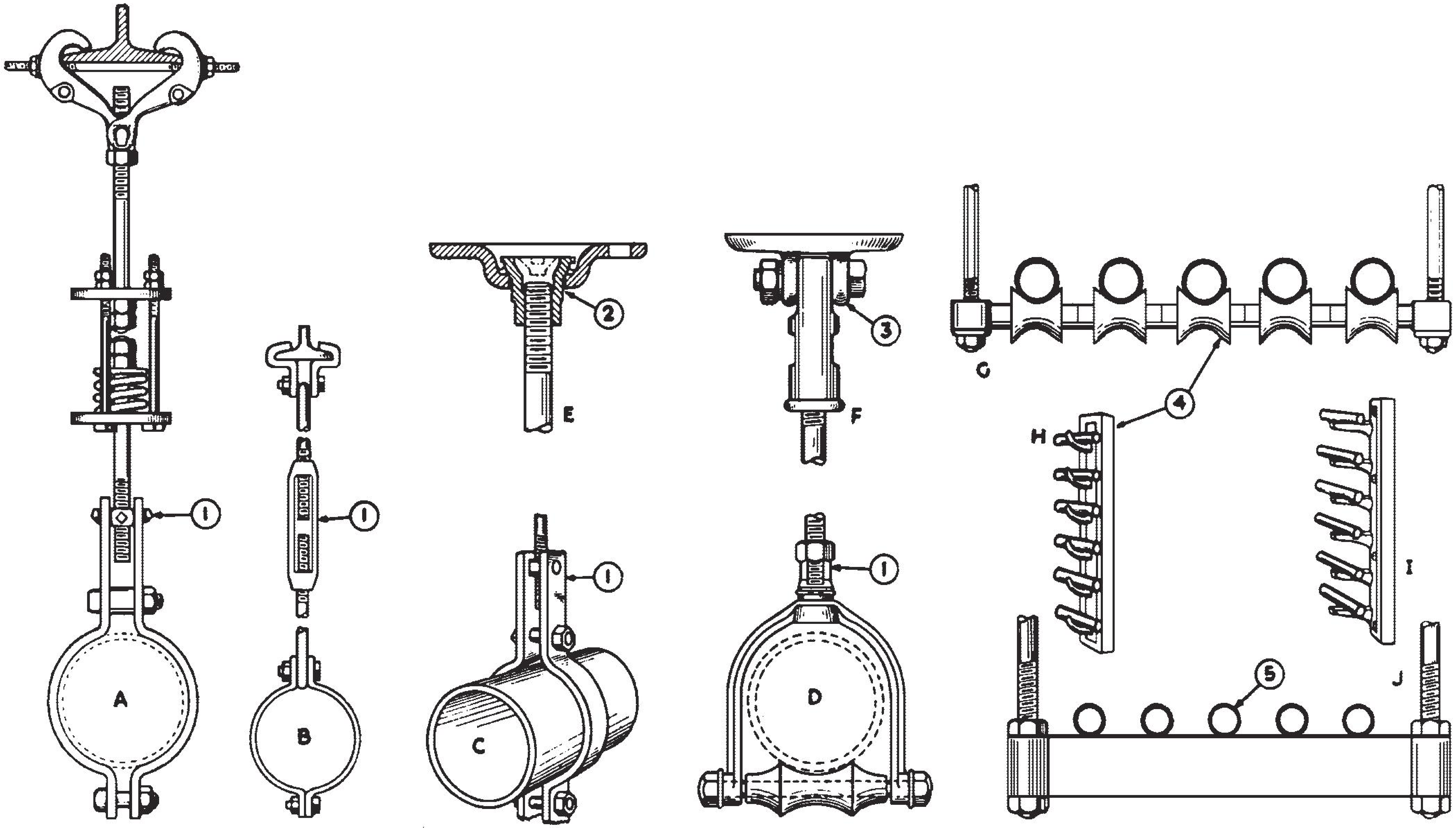


FIGURE 9.19 Typical pipe supports. Hangers generally permit vertical adjustment (1) to permit piping alignment to be maintained and to allow for proper division of load among supports. Pivoted hangers may permit universal movement (2) or may permit one-way movement as shown at (3). Multiple supports for banks of small pipe may be grooved for axial movement only (4) or may have a flat surface (5) to allow a certain amount of sidewise movement.

must be "soft." Such springs permit substantial up and down movement with only a moderate change in the support delivered to the pipe.

Maintenance of a piping system will naturally start with obvious ills—leaks, water hammer, swaying, and vibration. Leaks may be caused by joints improperly designed or made up, by expansion forces, or by improper support. Piping should always be supported on both sides of every large valve. Where leaks persist or recur despite good joint technique, check the alignment and condition of neighboring anchors, supports, and expansion joints to make sure the leak is not caused by external forces. Other common causes of leaks are pipe swaying and water hammer. Water hammer may result from reciprocating pumps or the too quick closing of valves. A common cause, traceable to poor maintenance, is undrained condensate caught in low points of the piping, back of globe valves, etc. Check the system to make sure that low points are raised and all other pockets drained. Sags caused by misalignment can usually be cured by simple hanger adjustments.

When all such obvious ills have been cured, the job shifts to preventive maintenance—regular routine inspection to make sure that anchors are holding and showing no signs of breaking or slipping; that walls and footings near anchors are not showing distress cracks; that sliding expansion joints are not leaking or sticking; that supports everywhere are in line with pipe and tracking true; that supporting rolls turn freely and support their share of the load whether the line is hot or cold; and that bolts, turnbuckles, and other stressed members give no sign of distress or possible early failure.

Pipe Insulation. A good maintenance program keeps insulation in perfect condition because necessary inspections and repairs save many times their cost in fuel. Here are practical pointers on maintaining and repairing covering of piping, valves, and fittings—chiefly those containing steam or hot water:

A good job of hot-pipe insulation will have the following characteristics: (1) efficient insulating material applied to economic thickness, (2) material able to stand ordinary handling, (3) inner layer able to stand pipe temperature, (4) insulation bound securely to pipe, (5) joints closely fitted and staggered (if double layer), (6) insulation well covered and painted, if necessary, and (7) complete waterproofing for outdoor or underground lines.

Materials. Widely used heat-insulating materials for plant piping include calcium silicate, laminated asbestos-felt, and various forms of mineral wool (both molded and felted), including glass wool, as well as other materials. The calcium covering is molded in sections or blocks. Both calcium and laminated-asbestos coverings may be used safely up to 600°F. Mineral-wool insulation can withstand temperatures higher than 1000°F.

For pipes above the temperature limit of calcium and asbestos, double-layer insulation is common. An outer layer of calcium or asbestos is protected from overheating by an inner layer of molded covering composed of calcined diatomaceous silica, asbestos fiber, and cementing materials. This high-temperature covering looks like calcium and has lower insulating efficiency and the ability to resist temperatures well above 1000°F. Both mineral wool and laminated asbestos are particularly suited for points subjected to heavy vibration or shock. Table 9.4 gives the insulation thickness commonly recommended for laminated-asbestos and calcium coverings. Note that the correct thickness increases with both pipe size and temperature.

Figure 9.20 shows standard methods of covering piping and fittings. Such applications are fairly simple for any mechanic. Calcium and other molded coverings can be easily sawed to trim length and beveled with a knife at the ends that face the flanges. Coverings on bends, fittings, and other irregular surfaces can be built up by wiring on odds and ends of calcium blocks or pipe coverings and filling the remaining spaces with calcium cement. Since this cement is of exactly the same composition as the solid pieces, the mass sets as a homogeneous whole.

Insulation for cold lines may range all the way from a simple antisweat jacket for cold water to elaborate built-up, thick insulation for low-temperature refrigerating lines. Materials used include hair felt, cork, and mineral-wool felt. An essential characteristic of low-temperature insulation is complete sealing against the penetration of moisture that would otherwise destroy the insulation. Check periodically to make certain that such coverings remain hermetically sealed and completely free of internal water or ice. The application of refrigeration insulation is a specialty for experts, as is most heat insulation.

Maintenance. Routine maintenance of warm-pipe insulation should include prompt repair of damaged surfaces, repainting and waterproofing, tightening bands and wires, and repairing torn canvas jackets. Look out for shrinkage, loosening, and the effect of moisture, fumes, and vibration. Make sure that steam temperatures have not been raised above the safe limit for the material used. Check carefully for steam and water leaks concealed by insulation.

Often a casual inspection will reveal bare flanges—either originally bare or left so by a recent replacement of the gasket. A single large bare flange can waste a ton of coal per year; so all such should be covered—preferably with replaceable covers.

Flanges. The preservation of flange covering is a major maintenance problem. Even supposedly removable covers are often broken in removal, particularly when an emergency requires quick access to the flange. Replacement of the insulation is a nuisance, too often delayed or omitted because the plant can run without it. It should be a standard rule to protect flange covers as far as possible and replace them promptly.

Silicate of soda (water glass) is a convenient and powerful adhesive for cementing tears in asbestos laminations or canvas jackets. Where calcium covering is broken, a monolithic repair can

TABLE 9.4 Recommended Thicknesses of Pipe Covering

Plus inner layer of HT for high temperature

	Temperature of hot surface, °F														
	170	270	370	470	570	670		770		870		970		1070	
	Temperature difference, °F														
Pipe size, in.	100	200	300	400	500	600		700		800		900		1000	
	Calcium					HT	Cal	HT	Cal	HT	Cal	HT	Cal	HT	Cal
1	S	S	1½	2	2	2		2		2		2½		2½	
2	S	S	1½	2	2	1½	1½	1½	2	2	2	2½	2	2½	2½
3	S	S	1½	2	2	1½	1½	1½	2	2	2	2½	2	2½	2½
4	S	S	1½	2	2	1½	1½	1½	2	2	2	2½	2	2½	2½
5	S	S	2	DS	DS	1½	2	1½	2	2	2	2½	2	2½	2½
6	S	S	2	DS	DS	1½	2	1½	2	2	2	2½	2	2½	2½
8	S	S	2	DS	DS	1½	2	2	2	2	2½	2½	2½	3	2
10	S	S	2	DS	DS	1½	2	2	2	2	2½	2½	2½	3	2
12	S	S	2	DS	DS	1½	2	2	2	2	2½	2½	2½	3	2
14	S	S	2	DS	DS	1½	2	2	2	2	2½	2½	2½	3	2
16	S	S	2	DS	DS	1½	2	2	2	2	2½	2½	2½	3	2
18	S	S	2	DS	DS	1½	2	2	2	2	2½	2½	2½	3	2
20	S	S	2	DS	DS	1½	2	2	2	2	2½	2½	2½	3	2
Flat	1½	1½	2	2½	3	1½	2	2	2	2	2½	2½	2½	3	2

Note: HT = high-temperature covering, S = standard thick, DS = double standard.

Laminated Asbestos

Temperature of heated surface, °F	Pipe size, in.		
	Under 2	2 to 4	4½ and up
Up to 300	1	1½	1½
301 to 400	1½	1½	2
401 to 500	2	2	2½
501 to 600	2	2½	3

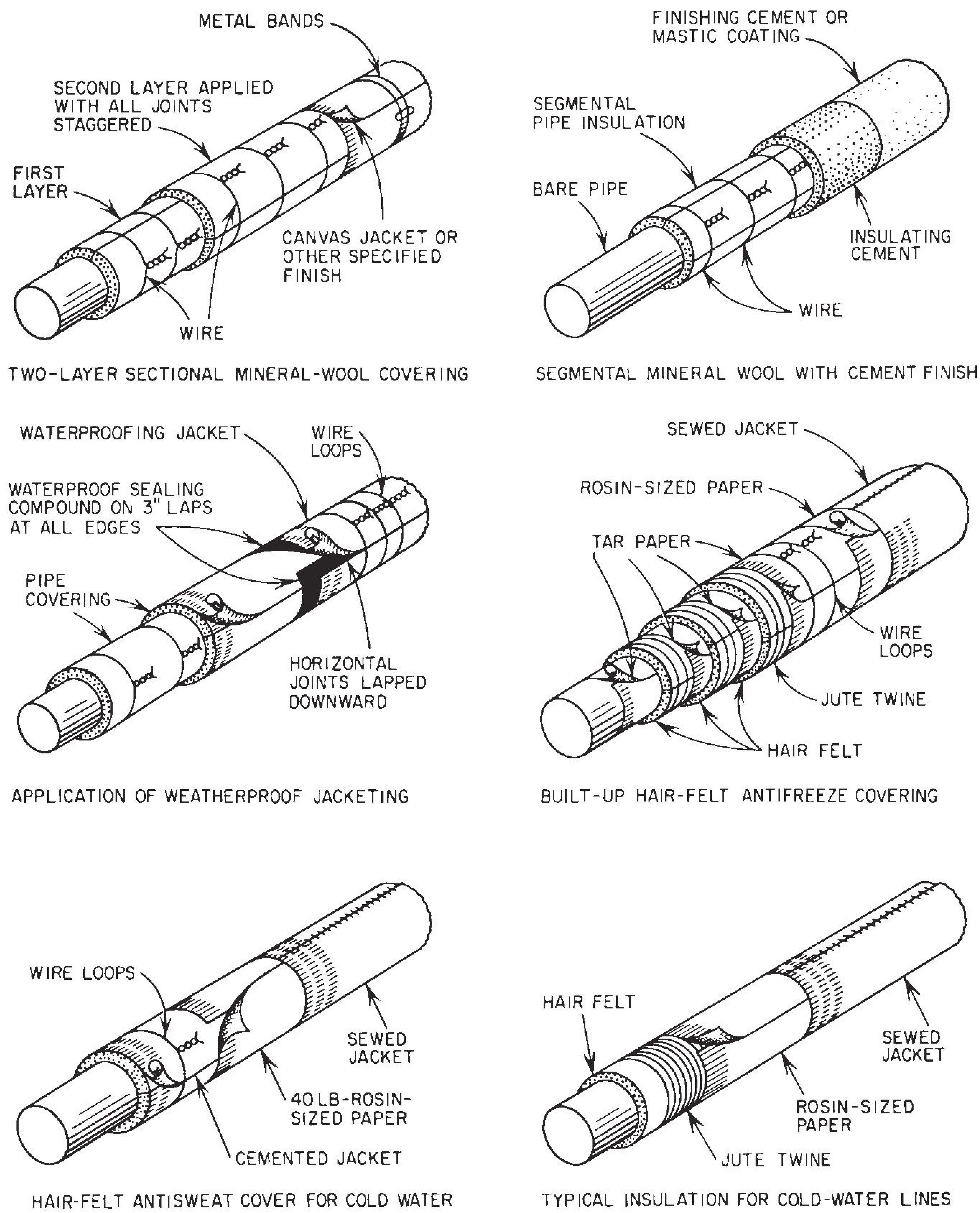


FIGURE 9.20 Methods for applying some typical pipe insulations.

be made by wiring in a calcium "Dutchman" and filling voids with cement, which has the same composition as the block. See Fig. 9.21 for pointers on insulation maintenance.

Maintenance Welding. Each element of piping maintenance has two aspects: (1) how to select and install to reduce maintenance, and (2) how to maintain. In the case of welded joints the second is practically eliminated. If the joint is rightly selected and made, it should need no maintenance for the life of the plant. In general, maximum use of welding means minimum maintenance except where joints must be broken from time to time. Flange connections should be used at such points.

Most engineers are familiar with the standard lines of welding fittings—ells, tees, crosses, flange necks, etc.—also values with welding necks. No attempt will be made here to show how to make pipe welds. That information is available in concise booklets issued by the manufacturers and in

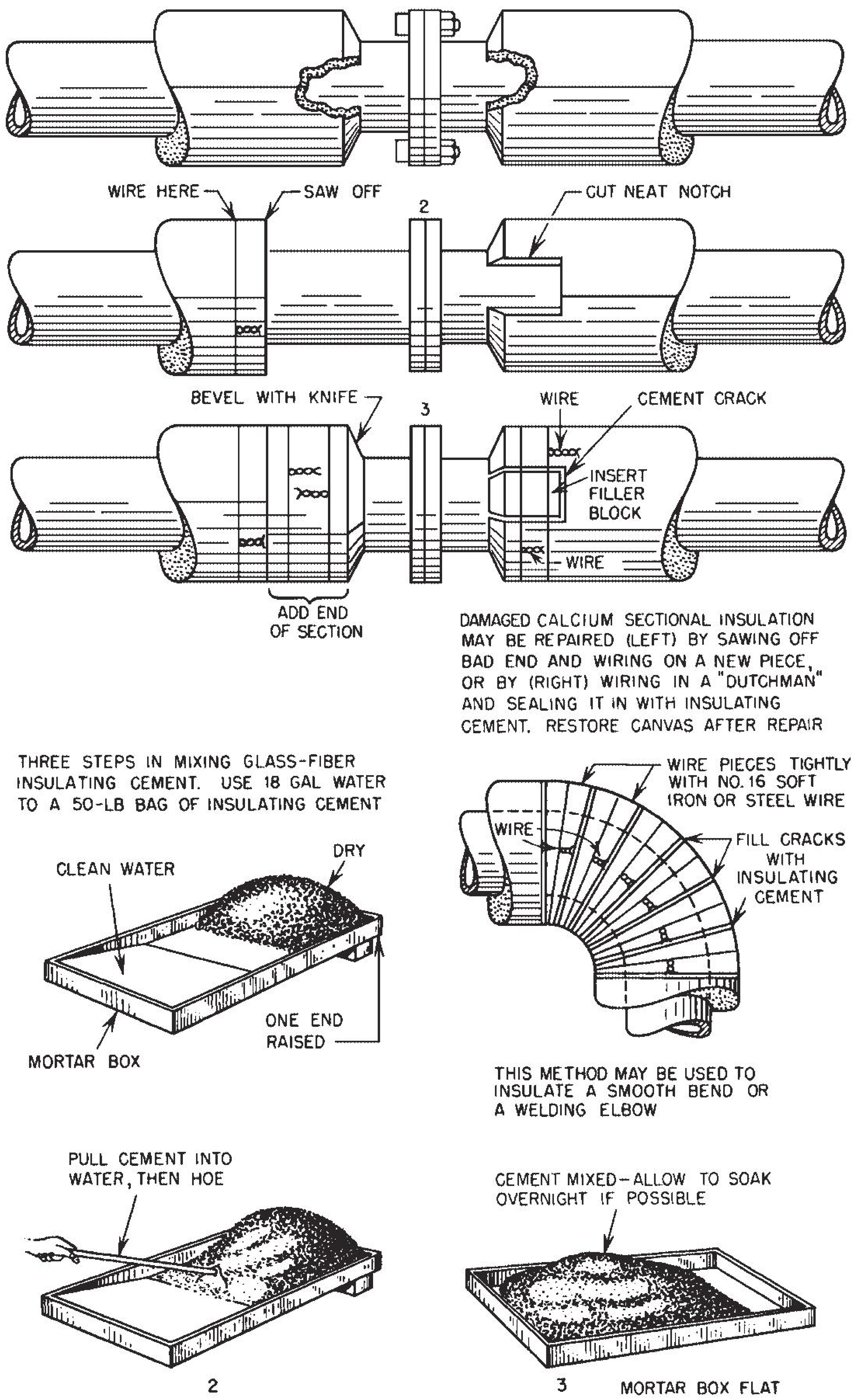


FIGURE 9.21 Pointers on insulation maintenance.

standard codes for the making and testing of welds and for the qualification of welders. The ANSI code "Pressure Piping" covers welded joints from the specification aspect.

The elaborate sleeves, patches, and reinforcements of the early days of welding reflected the user's lack of confidence in the process. Modern practice favors the plain butt weld, made by officially qualified welders following standard welding and testing procedures. Results have been so good that welding is preferred today for the highest pressures and temperatures. In such lines flanges are used only where joints must be breakable.

Joints. In low-pressure work, welding has the disadvantage that cast-iron valves cannot be welded in, so one must either use the more expensive steel valves or install the cast-iron valves with flanged joints. For pipe wall up to $^3/_4$ in. thick, the sides of the standard butt joint are beveled $37^1/_2°$, with a $^1/_{16}$-in. land. For thicker pipe the bevel is U-shaped to avoid the need for excessive welding. Chill rings ensure full penetration of the weld without the formation of dangerous "icicles" in the pipe. Except for high-velocity steam, the extra cost of a flush chill (Fig. 9.22) is rarely warranted.

Flanges can be installed in a welded system in the three ways shown in Fig. 9.22. The lap-joint stub end, welded to the pipe and backed by a slip-on flange, gives the highest type of lap joint. A quick field connection can be improvised by joining a slip-on flange to the pipe end by two fillet welds as shown. Welding may be used also to seal the thread where a flange is screwed onto a pipe.

In high-pressure work it is best to build up welds in $^1/_8$-in. layers, cleaning and inspecting after each layer. This catches defects before they are buried. Moreover, the heat of each layer improves the grain structure of the underlying layer.

It is considered unsafe to weld "carbon-moly" steel without preheating; so induction and resistance electric heaters have been designed to keep the pipe at 300 to 600°F throughout the welding. Similar heaters are used to stress-relieve high-strength welds at 1200°F.

Corrosion-Resistant Piping. In recent years a number of nonmetallic materials have been introduced for corrosion-resistant service—plastic, glass, etc. Older corrosion-resistant materials include transite, stainless steel, cast iron, and coated piping. The coating may be plastic, bitumastic, rubber, etc. The general maintenance procedures for piping and fittings made of special materials resemble those for iron and steel piping, except for differences resulting from the materials involved. For specific procedures in maintaining special piping materials, consult the manufacturer. There are many variations in procedures and methods which must be carefully observed. The corrosion questionnaire, Table 9.5, can be helpful as a reminder of factors to be considered.

Pipe Materials. For a comprehensive list of piping materials see the ANSI code "Pressure Piping." For maximum safety it is advisable to design all industrial piping in accordance with code requirements, even though local regulations may not make this mandatory. Correct design, with adequate provisions for maintenance, is the key to long trouble-free operation with minimum attention. Routine inspection of piping systems and their associated equipment ensures finding minor defects before they become major ones.

Flange Bolting. Table 9.6 gives data determined by the Crane Company in a number of extensive surveys of field-erected flanged joints. Experience shows these stresses are satisfactory for American Standard steel flanges. It is recommended that the initial bolt stress be about 45,000 psi. Figure 9.23 shows a micrometer for measuring bolt-stud elongation.

This company also recommends that the bolting in all flanged joints operating at temperatures over 500°F be pulled up after the first shutdown. At high temperatures, where creep may be expected to occur, it is recommended that bolting be pulled up at least once during the first 200 hr of service, regardless of whether the line has been shut down or not. Check bolt stress periodically during the life of the installation, as part of the routine maintenance program.

As a general rule, it is only necessary to check the elongation in two or three diametrically opposite bolt studs, using the average of the values obtained as the elongation for the remainder. If the bolt elongation is to be determined while the joint is in service, one bolt should be checked and pulled up before another is loosened. Measurements should be taken immediately after releasing the load before the bolt temperature decreases. The following procedure should be used in checking bolt elongation: (1) Determine the length of the bolt in the assembled joint. (2) Release the load on the bolt by loosening the nut, and remeasure the length. (3) Subtract the second reading from the first. (4) Divide this value by the effective length of the bolt. (Effective length equals the distance from the center of one nut to the center of the other.) If the residual elongation is less than 70 percent of the values given in Table 9.7, the bolts should be pulled up so that the final elongation approximates the figures shown.

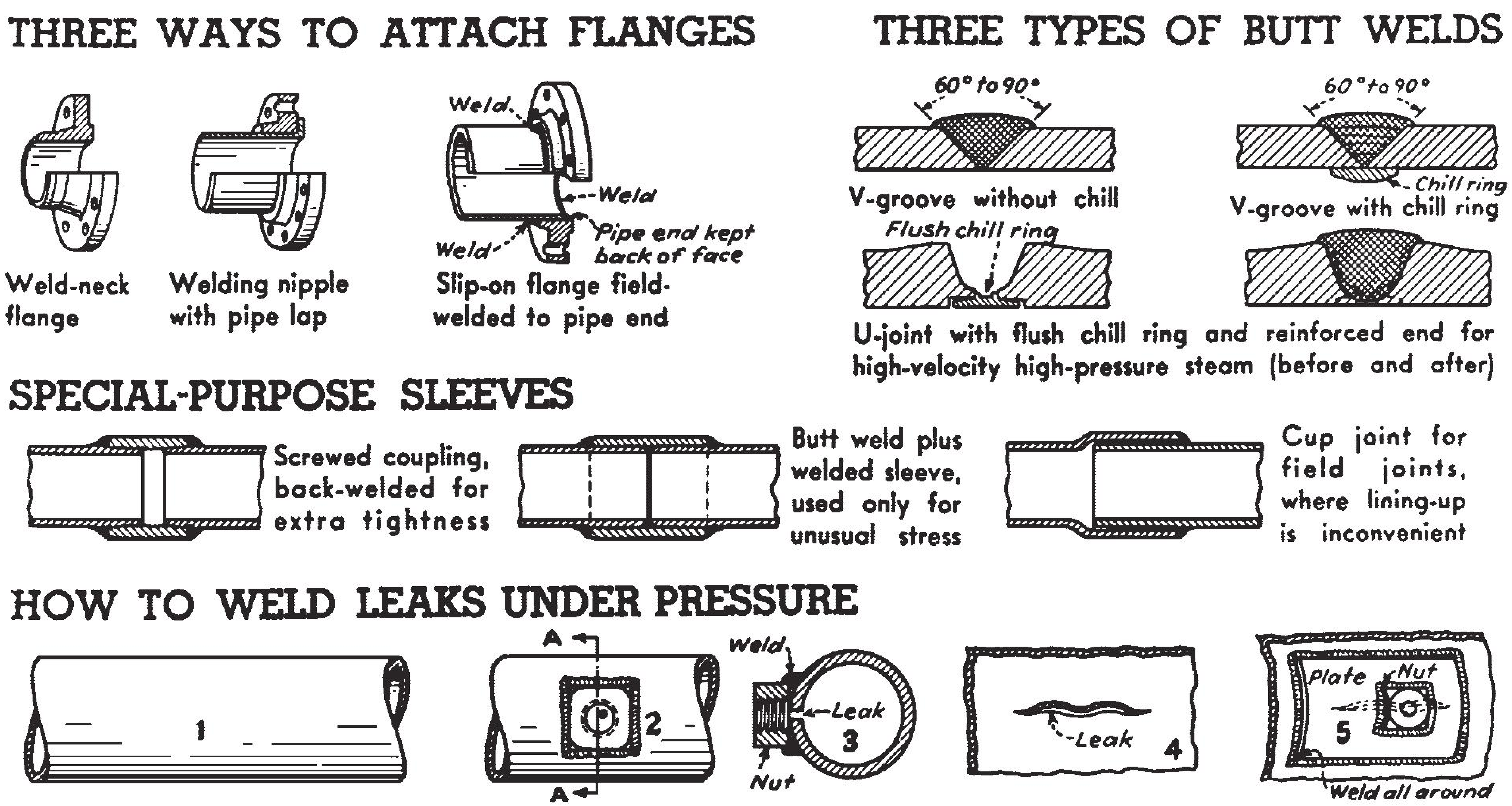


FIGURE 9.22 Use of welding in piping maintenance. Electric-arc welding is a quick way to stop leaks in pipe under pressure. To close pinhole leak (1), weld an ordinary square-head machine nut around leak (2, 3). This permits water, etc., to escape through the hole in the nut during welding, making it possible to complete the weld. To stop the leak, screw a bolt into the nut with sealing compound. A cracked or split pipe (4) can be closed in the same way. This kind of leak usually is too large to be covered by a nut; so a piece of plate stock is applied as a patch. Drill a small hole in the plate and shape the plate to the pipe. Next, weld a nut over the hole and weld the plate to the pipe (5). Then screw in a bolt or plug, using sealing compound.

TABLE 9.5 Corrosion Questionnaire

1. What is the name or composition of fluid to be handled?
2. What is the concentration (percentage strength, specific gravity, pH value, etc.)?
3. What is the operating temperature and pressure in pipelines?
4. If the fluid is not a water solution but a gas, organic fluid, etc., is water or water vapor apt to be present at any time or at any particular location? If so, explain.
5. If the fluid is a water solution, are substances other than water (abrasive solids, oil, etc.) present at any time or at any particular location? If so, explain.
6. Is there much opportunity for air-in leakage?
7. Is the flow through the system intermittent or continuous?
8. If intermittent, is the system ever drained entirely of fluid and allowed to dry?
9. Is the piping system washed or rinsed at regular intervals, and if so, with what materials?
10. Is a slight amount of corrosion objectionable from the standpoint of contamination or discoloration of product?
11. If so, what metals are particularly objectionable?
12. What materials are being used or proposed for piping, tanks, etc.?
13. Has any specific trouble been experienced with these materials?
14. What materials have been used for valves and fittings?
15. In general, what is the comparative life of the different materials which have been used?
16. What packings give the best service?

Courtesy of Crane Company.

TABLE 9.6 Flange-bolting Data

Size of alloy-steel bolt-stud*	Average stress applied manually, psi†	Approximate torque to obtain stress, ft-lb‡	Elongation, in. per in. of effective length§
3/4	52,000	175	0.00173
7/8	48,000	255	0.00160
1	45,000	370	0.00150
1 1/8	42,500	500	0.00142
1 1/4	40,000	665	0.00133
1 3/8	38,000	860	0.00127
1 1/2	36,500	975	0.00122
1 5/8	35,000	1,285	0.00117
1 3/4	34,000	1,700	0.00113
1 7/8	33,000	2,200	0.00110
2	32,000	2,350	0.00107

*Coarse thread series, 1 in., and smaller; 8-pitch thread series, 1 1/8 in., and larger.
†Average stress applied by maintenance men in assembly, using a lever and wrench or by sledging.
‡Based upon well-lubricated threads.
§Based on a modulus of elasticity of 30,000,000. The effective length of bolt-stud equals the distance from center of one nut to center of the other.

Courtesy of Crane Company.

Liquid Velocity. Table 9.7 gives condensed data from the experience of two hydraulic organizations with liquid velocities for pipes and ditches of many different types. It is useful in choosing or changing velocities to reduce pipe maintenance and deterioration.

Leak Detection. Where leaks are difficult or impossible to detect visually, radioactive isotopes are widely used to determine the general location of the opening in the pipe. The isotope is injected into the liquid stream and its flow traced by a Geiger counter. A leak is detected by a change in the

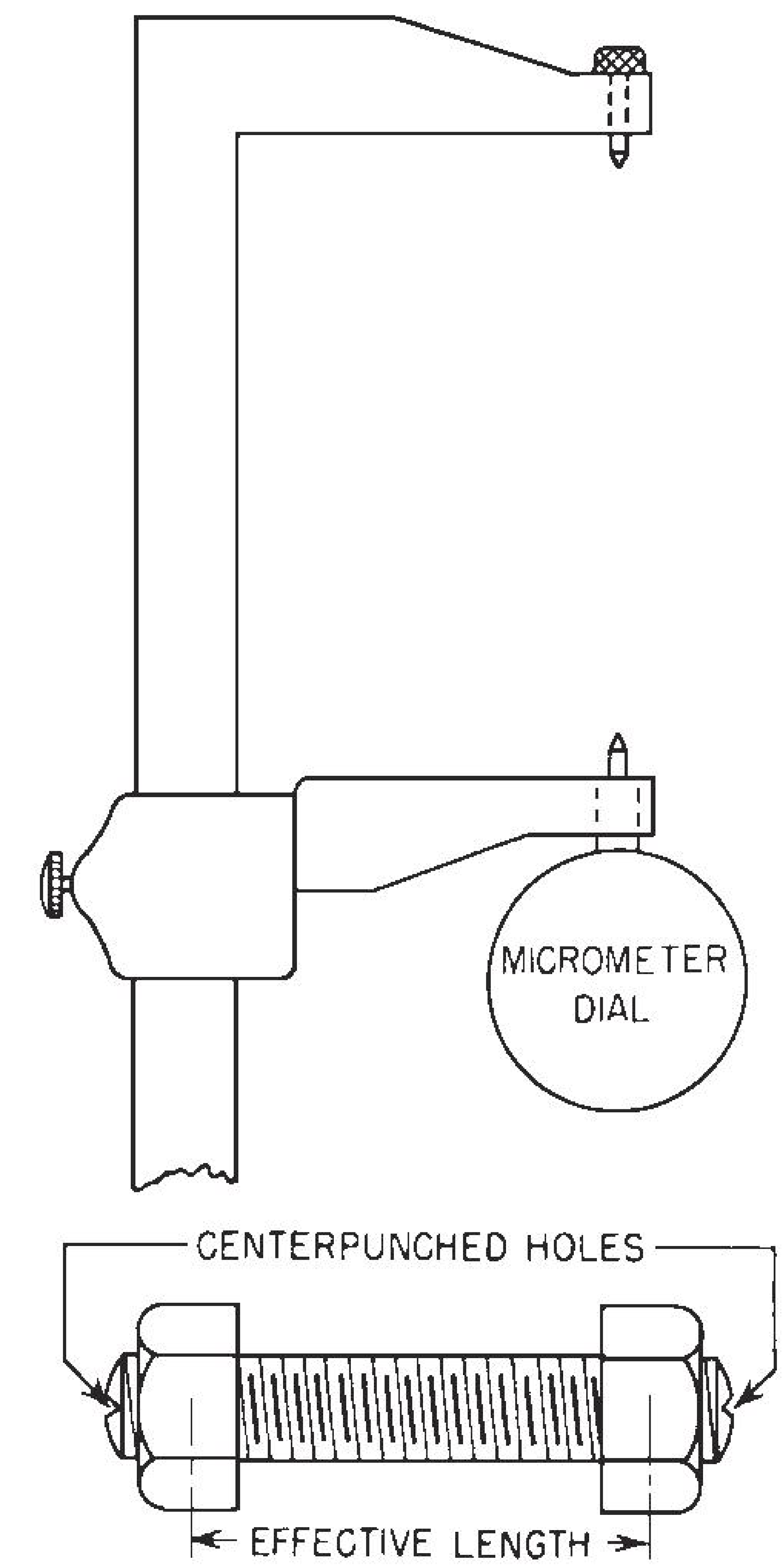


FIGURE 9.23 Micrometer for measuring bolt elongation.

radioactivity level. Since this method exposes personnel to some radioactivity and requires special equipment, it is usual practice to have a firm specializing in this type of testing do the work.

Acid Cleaning. Piping, valves, heat exchangers, process vessels, etc., are often cleaned today by means of acid. This is pumped through the piping and vessels under controlled conditions of concentration, velocity, temperature, and time. After the acid is removed from the system, a neutralizing agent is generally applied, followed by a flushing with clean water. Since a rather specialized knowledge and equipment setup is required for acid cleaning, it is usual practice to have this work done by a firm having the required experience and machinery. Since portable equipment is suitable, acid cleaning can be done in the plant during a routine shutdown. Correctly applied acid cleaning can often reduce piping-system maintenance costs considerably.

Pipe Identification. Colored bands in accordance with the ANSI "Scheme for Identification of Piping Systems" are valuable in plant maintenance operations. They permit more ready identification of the piping, eliminating errors.

Relief Devices. Safety and relief valves must be kept in good working order at all times to ensure safe operation and prevent the loss of valuable liquids. Routine tests of relieving capacity should be part of the piping maintenance program. Safety heads (Fig. 9.24*A*) must be replaced immediately after function (Fig. 9.24*B*).

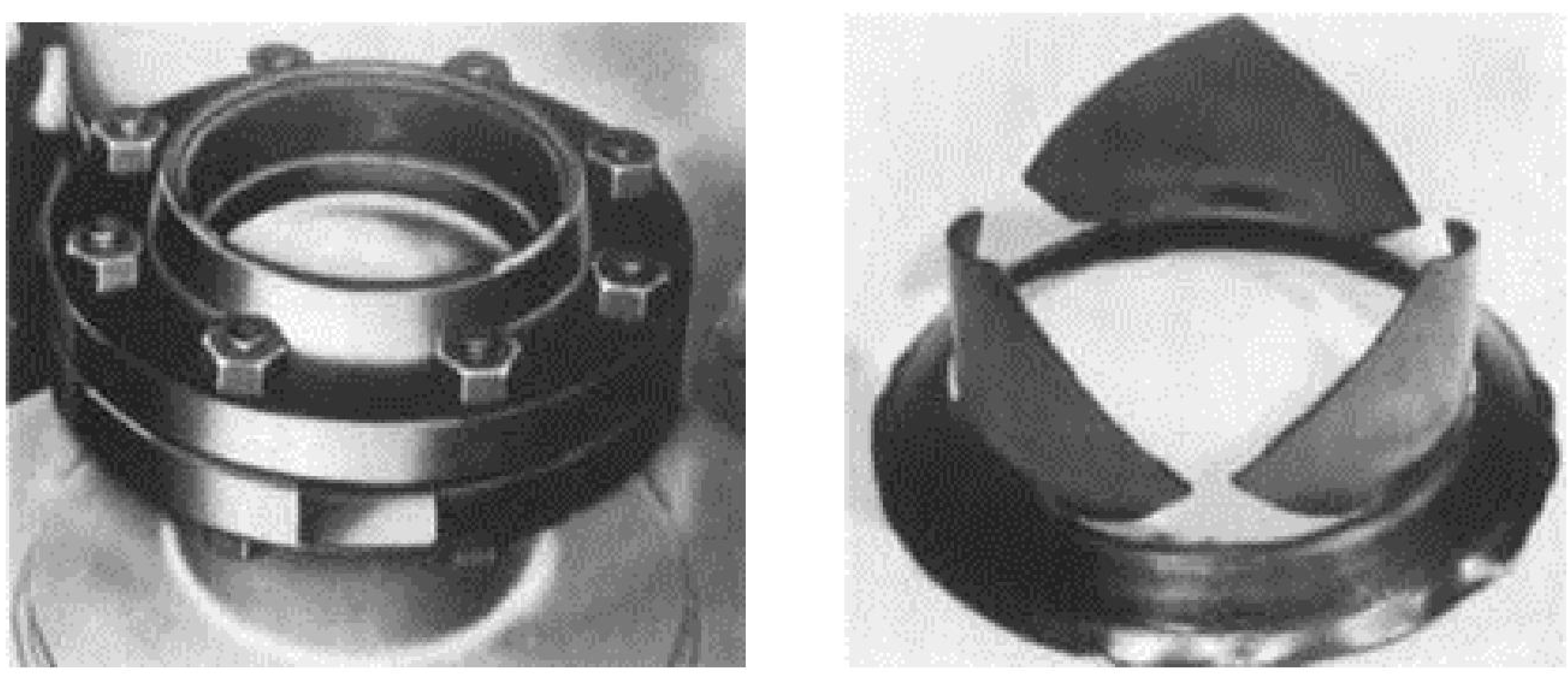
Plastic Piping. Metal piping is being replaced in many applications by plastic piping of various types. Plastic piping has internal, external, and electrolyte corrosion resistance. It is easy to install, is not subject to caking of the fluid on the walls, and cannot be pitted by tuberculation. Since plastic pipe does not corrode, it is ideal for high-purity systems.

Plastic pipe, however, has both temperature and pressure limitations. The usual maximum allowable operating temperature is in the 200°F range and depends on the material of which the pipe is

TABLE 9.7 Water Velocities to Reduce Piping Maintenance

Conditions to be prevented	Type of flow	Pipe or ditch material	Velocity limits, ft per sec
Deposits of silt and mud	Vertical upward	All types of pipes and ditches	24 min
	45° upward		13 min
	9° upward		5 min
	3° upward		4 min
	Horizontal		3.3 min
	3° downward		2.6 min
	9° downward		Almost zero
Rust formation	All types	All corrosive pipe materials	26 min
Deterioration of pipe walls	All types	Concrete pipe, carrying pure water	20 max
		Concrete pipe, carrying sand-laden water	10 max
		Steel and cast-iron pipe	50 max
		Wood-stave pipe	40 max
Deterioration of ditch walls	All types	Fine-grained sand	0.6 max
		Coarse sand	1.2 max
		Small stones	2.4 max
		Coarse stones	4.0 max
		Rock	25 max
		Concrete carrying sandy water	10 max
		Concrete ditch, carrying pure water	20 max
		Sandy loam, 40% clay	1.8 max
		Loamy soil, 65% clay	3.0 max
		Clay loam, 85% clay	4.8 max
		Soil, 95% clay	6.2 max
		Clay	7.3 max
Formation of ice in ditch or race	All types	All types of ditches or races	5.0 min

These velocities realize special conditions.



(*A*) (*B*)

FIGURE 9.24 *A.* Safety heads on compressed-air bottles. *B.* Ruptured safety head.

made. Certain glass-fiber-reinforced pipe can withstand temperatures up to about 250°F; asbestos-reinforced pipe can operate at temperatures in the 400°F range. New plastic materials are rapidly being introduced—some allow operation at over 500°F. In general, though, plastic piping is usually confined to operating temperatures less than 200°F.

Allowable working pressure of plastic pipe decreases as operating temperature increases. Most plastic-pipe manufacturers recommended that the working pressure not exceed 20 percent of the

bursting pressure. Typical bursting pressures range from a high of about 1500 psi at 70°F for smaller-diameter pipe to a low of about 100 psi for 6-in. pipe at 70°F. Plastic pipe is more expensive than galvanized metal pipe, but the many advantages of plastic often make the extra investment worthwhile.

Valves, elbows, tees, flanges, and other pipe fittings are also made of plastic. Figure 9.25 shows a typical molded plastic valve. These parts have the same advantages as plastic piping and are subject to the same general pressure and temperature limitations.

Maintenance of Plastic Piping. Relatively little maintenance is required for plastic piping. Since the external surface of plastic piping resists corrosion, painting is never required for protection, though the surface may be painted for purposes of appearance.

Inspect plastic piping regularly for leaks, sagging, or out-of-roundness. Any of these conditions can be caused by excessive operating temperatures or pressures. Repair leaks using the solvent cement recommended by the pipe manufacturer. Be certain to drain pipe and dry it thoroughly before applying cement. Brush the cement carefully over the entire surface being repaired. Apply liberal amounts of the cement to ensure complete repair of the leaking joint. At least a 10-hr drying time is required before the pipe can be subjected to operating pressure and temperature. Where possible, allow a 48-hr drying time for development of full strength in the pipe.

Leaks caused by sagging or out-of-round pipes cannot be repaired without replacing the affected section of pipe, unless the damage is only minor. Sagging is caused by too few supports. Figure 9.26 shows the usual support spacing for polyvinyl chloride plastic piping recommended by one pipe manufacturer. Note that the recommended spacing is a function of flow temperature, pipe thickness (schedule number), and pipe diameter. Where pipe sagging causes leaks, check the distances between supports, the flow temperature, and the ambient temperature at the pipe. Fit more support if the distance between the existing supports exceeds that recommended in Fig. 9.26. Continuous supports are sometimes used for short spans in place of spaced supports. All supports should allow the pipe to expand axially without damage to the exterior surface of the pipe. Therefore, the support must not clamp the pipe tightly.

Where sagging is caused by excessive flow or ambient temperature, reduce the fluid or air temperature. Excessive temperature can cause sagging even when the pipe has properly spaced supports.

Out-of-roundness usually results from excessive operating temperature. Reduce the fluid or air temperature before installing replacement sections for the out-of-round pipe.

Since plastic piping is particularly sensitive to shock resulting from being struck by a hard object, protect the piping in areas of heavy traffic. Use a guard railing or low wall to protect the piping from fork-lift trucks and other vehicles used in industry. Plastic pipe used in outdoor service is often buried under ground to protect it from moving vehicles and the ultraviolet rays of sunlight. Direct sunlight will shorten the life of plastic piping made of certain materials.

FIGURE 9.25 Polyvinyl chloride molded plastic Y-type globe valve for use in plastic piping systems.

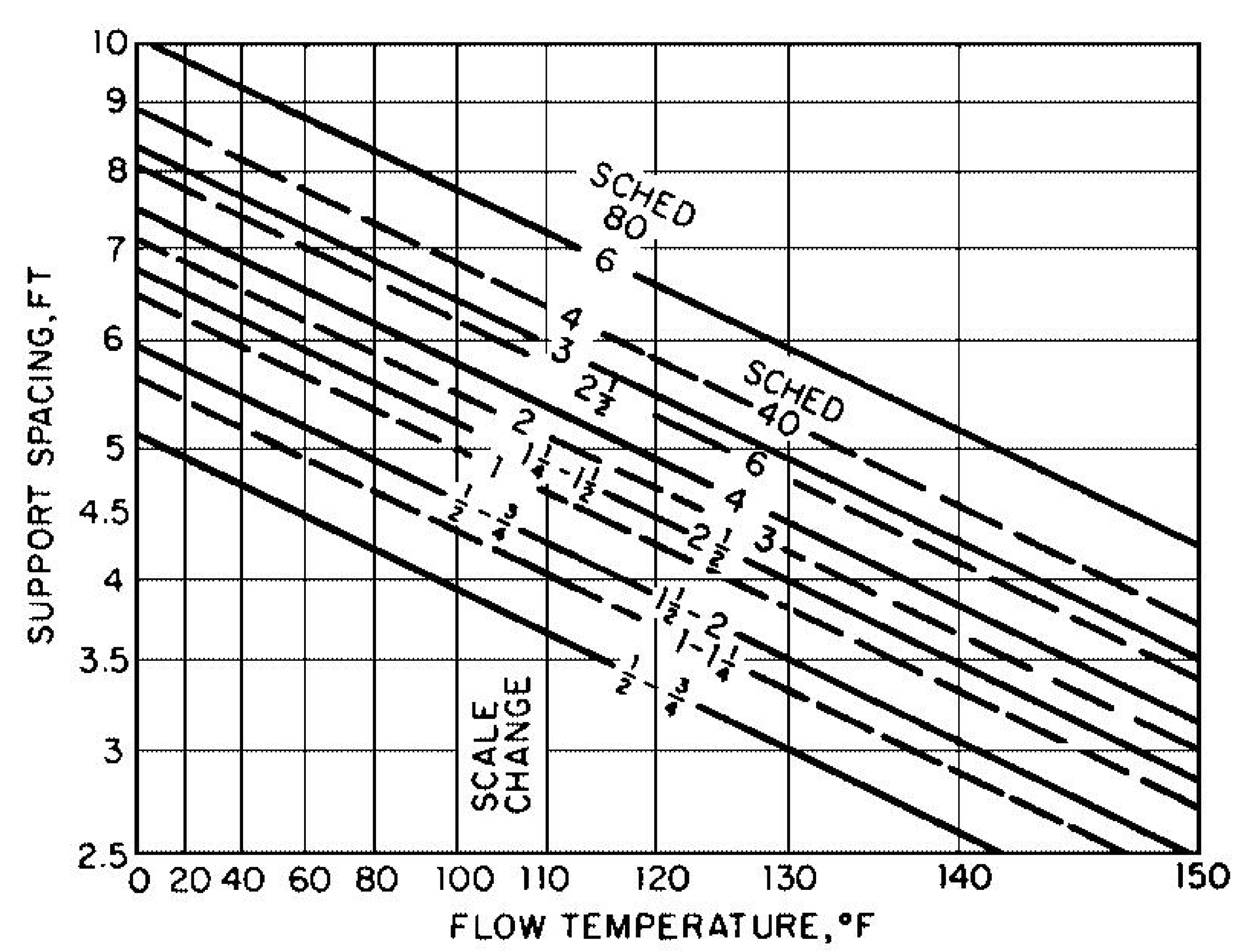


FIGURE 9.26 Recommended support spacing for uninsulated polyvinyl chloride piping. Chart is for plastic pipe carrying fluids of up to 1.35 gravity. For insulated piping reduce spans by 30 percent.